REPORT SUMMARY

SUBJECTS	Reliability, operations, maintenance, and human factors / Major components
TOPICS	Equipment qualification Mechanical equipment Pumps Valves Surveillance Maintenance
AUDIENCE	R&D and safety engineers

Qualification of Active Mechanical Equipment for Nuclear Plants

Proving that nuclear plant mechanical equipment can perform safety functions under extreme conditions is costly when each type of equipment must be tested individually. The less-rigorous qualification methods described in these guidelines are effective for rugged mechanical components.

BACKGROUND Equipment that performs safety-related functions in nuclear plants must be qualified to operate under all specified service conditions. Because electrical equipment is generally not designed to function in hostile environments, rigorous testing is required to qualify it for operation under postulated accident conditions. In contrast, mechanical equipment designed according to the ASME code for operation under adverse process conditions is inherently rugged. This project explores ways to qualify such mechanical equipment in nuclear plants without the need for expensive individual testing.

OBJECTIVES To provide guidelines, methods, and technical bases for qualifying active mechanical equipment in a cost-effective manner.

APPROACH Researchers first identified the types of mechanical equipment needing qualification and the typical environments in which such equipment must operate. They then compiled information relevant to qualifying a variety of mechanical devices: pumps, valves, diesel engines, drive turbines, fans, and snubbers. The information included manufacturers' design data, data documenting temperature and radiation effects on nonmetallic materials, industry standards, failure modes, surveillance and maintenance practices, and earthquake experience. With this information, the project team prepared a flowchart for use as a checklist in utility qualification programs.

RESULTS The study describes several procedures that can be used to qualify mechanical equipment in nuclear plants. Two of the most important are (1) a review of nonmetallic parts for radiation, thermal, and wear degradation; and (2) a thorough program of surveillance testing and preventive maintenance. Another recommended procedure, the use of radiation thresholds for the degradation of nonmetallic parts such as seals and gaskets, is appropriate

because radiation is the only hostile element in the nuclear plant environment that is new to this otherwise proven equipment. Seismic qualification can usually be accomplished on the basis of analysis of the equipment or experience with similar equipment in past earthquakes; plant-specific seismic testing should be conducted only as a last resort. The study further recommends that utilities document the technical bases for qualifying equipment as an integral part of the qualification program.

EPRI PERSPECTIVE This study complements and extends research on electrical equipment qualification carried out under EPRI project RP1707. That project has already assisted nuclear plant owners in an extensive NRC evaluation of qualification methods for electrical equipment. Plant owners can use the methods described in this report to demonstrate an adequate level of qualification for mechanical equipment. Properly applied, these methods qualify such equipment without the need for expensive piece-by-piece testing under simulated accident conditions. The guidelines in this report will be useful to engineers and managers responsible for mechanical equipment qualification in both operating nuclear plants and those being designed or constructed.

PROJECT RP2189-1
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Qualification of Active Mechanical Equipment for Nuclear Plants

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NP-3877
Research Project 2189-1

Final Report, March 1985

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ABSTRACT

This report describes a methodology which can be used for qualifying active mechanical equipment for safety-related service in nuclear power plants in a cost effective manner. The emphasis in this report is to develop guidance on technically sound methods that may be used to qualify mechanical equipment without resorting to piece-by-piece testing. This report identifies the types of mechanical equipment which may need to be qualified and discusses the considerations which are important in the determination of qualification loading conditions. In particular, the traditional practice of seismic qualification of all safety related equipment is reviewed to provide a well-founded technical basis for decoupling environmental loads from seismic (and dynamic) loads. Detailed examination of design, function and potential failure modes of valves, pumps, diesel engines, drive turbines, fans and snubbers provides the technical foundation for the appropriate qualification methods to be used for environmental, seismic and dynamic qualification. These appropriate qualification methods include: design, operating and earthquake experience; analysis and testing; evaluation of the effects of radiation on non-metallic parts; consideration of thermal and wear aging; surveillance and maintenance; and proper documentation.

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SUMMARY

This report describes a methodology that can be used by utilities, vendors, architect-engineers, and consultants to demonstrate qualification of active mechanical equipment which performs a safety related function in nuclear power plants. This report is intended primarily for those engineers and managers who have been assigned responsibility for mechanical equipment qualification. It is recognized that, as a group, these engineers and managers will represent a variety of technical disciplines and backgrounds. Some will be electrical engineers with experience in electrical equipment qualification but with less experience in the operation of mechanical equipment. Some will be mechanical engineers who are experienced in the operation of mechanical equipment but have little or no experience in equipment qualification. Others may have experience in both areas. The report, therefore, seeks to guide a relatively diverse group which may be responsible for mechanical equipment qualification.

The mechanical equipment qualification guidance in this report emphasizes appropriate qualification methods which can be used by operating plants. It is suggested that NTOL plants which have not submitted their mechanical qualification packages may wish to follow the report recommendations in lieu of the rigorous methodology being considered for codes and standards currently under preparation (e.g., IEEE/ASME-STD-627 and ANSI/ASME 551 draft codes).

The overall objective of the study is to compile the technical bases for qualifying mechanical equipment for safety-related service in nuclear power plants in a cost effective manner. The specific objectives are:

1. Identify types of mechanical equipment in domestic nuclear plants which may need to be qualified.

2. Recommend appropriate methods (historical, documentation, analysis and testing, other) that may be used to qualify mechanical equipment.

3. Develop guidance which the nuclear industry could follow to demonstrate qualification of mechanical equipment.

The guidance is based on the assumption that the mechanical equipment has been properly designed and proof tested at the appropriate ambient and elevated process conditions including operational dynamic loads. The purpose of qualification is then to show that the effects of aging, LOCA, and seismic loads do not negate that proven ability. The guidance emphasizes appropriate qualification methods which can be used by operating plants and plants under construction as an alternative to the rigorous methodology being considered for codes and standards currently under preparation.

This report concludes that mechanical equipment qualification is fundamentally different from electrical equipment. The basis for this is:

<u>Mechanical Equipment</u>	<u>Electrical Equipment</u>
Rugged Design - High normal operating loads - Process conditions	Less Rugged Design - Low mechanical loads - Ambient conditions
Ageable Parts Replaceable	Ageable Parts Frequently Integral
Failure Modes Predictable - Generally structural - Aging effects manifested by degradation in mechanical properties	Failure Modes Subtle - Generally non-structural - Aging effects often inferred from degradation in mechanical properties

The review includes the following considerations:

- Equipment Design/Operating Experience
- Equipment Seismic Experience
- Effects of Radiation on Non-Metallic Materials
- Mechanical Failure Modes
- Surveillance and Maintenance

- Seismic Analysis and Testing Methods
- In-Situ Resonance Search Techniques

The types of mechanical equipment used in safety-related service which are discussed in this report are:

- Gate Valves
- Butterfly Valves
- Check Valves - Vacuum Breakers
- Globe Valves
- Safety Relief Valves
- Main Steam Isolation Valves
- Centrifugal Pumps
    - Vertical In-line
    - Vertical Deep Draft
    - Horizontal End Section
    - Horizontal Multistage
- Diesel Engines
- Drive Turbines
- Fans
- Snubbers

The report describes the above equipment in sufficient detail to provide a basis for identifying potential failure modes which can be addressed by proper consideration of environmental, seismic and dynamic loading conditions.

This report provides guidance in determining which specific mechanical components are safety-related and which features of the mechanical components deserve the most attention to assure its active function.

The specification of environmental, seismic and dynamic loading conditions which apply to mechanical equipment for any particular plant are not within the scope of this report. However, typical loading conditions have been used to guide utilities in assessing the capability of their equipment.

The potential for exposure of mechanical equipment to high radiation is a unique feature of nuclear power plants. This report shows that qualification for radiation effects is satisfied where:

a. The total integrated dose does not exceed 10^5R (except where Teflon is used and the limit is 10^4R), or

b. The only non-metallic material is oil or grease and the integrated dose does not exceed 10^7R.

c. Graphite and asbestos materials are used; waive further review if the only non-metallic materials used in the equipment are graphite or asbestos based.

In radiation environments which are more severe, radiation qualification of non-metallic subcomponents should be included in the basis for the preventive maintenance and surveillance program. Even in these cases, radiation exposure is not necessarily the controlling or limiting factor in the environmental qualification of mechanical equipment. The parts in mechanical equipment which are subject to environmental degradation are non-metallic but are replaceable during maintenance. The important point about replacement parts in mechanical equipment is that environmental factors other than radiation may be more significant -- even in nuclear plants -- in determining the replacement intervals. These other environmental factors include thermal and wear aging.

For mechanical equipment, environmental qualification should, therefore, be based primarily on two factors:

- A well-established program of preventive maintenance and surveillance testing

- A review of the non-metallic parts which may be degraded by thermal, radiation, or wear aging

Typical surveillance testing requirements are stipulated in plant technical specifications. These requirements, translated into specific measurements in the plant test procedure, allow aging effects such as wear, corrosion and running clearances to be monitored and tracked throughout the life of the equipment. The Technical Specifications issued for all operating plants require that all active safety-related components, including diesel engines, mechanical drive turbines, pumps and valves undergo periodic surveillance test.

The predictability of mechanical failure modes resulting from thermal, radiation and wear aging is based on experience, good maintenance practices and the availability of data on the radiation effects on the mechanical properties in non-metallic materials. Consequently, environmental testing need not play a significant role in the overall qualification program for mechanical equipment at nuclear power plants.

For mechanical equipment, seismic and dynamic qualification can be based primarily on experience and analysis. Experience has always played a key role in assuring operability of mechanical equipment. The aspects of mechanical equipment operability which can be evaluated by seismic qualification methodologies are dominated by bearing load capability and adequate running clearances which are based on extensive design and operating experience. A recent study by the Seismic Qualification Utilities Group (SQUG) investigated the use of seismic experience from non-nuclear facilities where earthquakes have occurred. The SQUG study has received the endorsement of an independent panel called the Senior Seismic Review and Advisory Panel (SSRAP). The SQUG study provides seismic experience data which can be used for some pumps and valves as complete seismic qualification by direct comparison of equipment and seismic motion bounding spectra in the data base. Additional data might be compiled and evaluated to extend this approach to generic seismic qualification to include a broader range of component sizes and types as well as seismic motion bounding spectra.

For components not covered by generic experience data, analytical methods are well developed and well suited to the prediction of mechanical failure modes. Qualification by analysis involves the choice of static or dynamic analysis. A static analysis may employ either the static method or static equivalent method. The static method is the simplest form that applies to rigid equipment and is an analysis based on Zero Period Acceleration (ZPA) loads being applied to the equipment centers of mass. The static equivalent method is based on the equipment response being unknown and the equipment being subject to peak acceleration loads multiplied by a coefficient of 1.5 to account for unknown closely coupled dynamic modes. Dynamic analysis involves modelling the equipment by masses and structural elements and determining the response of the model to a spectrum of dynamic seismic loads. Dynamic analysis almost always involves the use of computer programs designed to solve the time dependent, finite element equations.

In some cases, qualification may employ a combination of analysis and testing. Economic and technical considerations can play a role in the decision to use testing as part of the qualification program. The common types of qualification testing are:

- In-situ Testing (resonance search techniques) for determining equipment natural frequencies to be used in the seismic analysis.
- Static Testing, such as the application of a static horizontal load to a valve stem during cycling of the valve.
- Shake Table Testing, primarily of smaller components or ancillary equipment used on such equipment as mechanical drive turbines and diesel engines.

It is recommended that qualification testing plans be carefully thought out to ensure the need for the test, to establish the test objectives, and to justify how the test results will support the qualification of the mechanical equipment.

In summary, the guidance in this report for the qualification of active safety-related mechanical equipment involves the following topics:

- Typical safety-related systems which contain mechanical equipment in both Pressurized Water Reactors and Boiling Water Reactors
- Definition of what is meant by active safety-related mechanical equipment
- Refinement in the definition of safety-related to distinguish which equipment is provided to function in the case of an accident and which equipment is provided to function in the case of an earthquake
- Mechanical failure modes of mechanical equipment
- Use of all types of historical experience data (design, operating, earthquake) to support the qualification of mechanical equipment

- A review of ageable materials
- Preventive maintenance and surveillance

Section 1
INTRODUCTION

BACKGROUND

The process of demonstrating that safety-related nuclear power plant equipment can function as required to perform its safety-related function during all specified service conditions is referred to as equipment qualification. Qualification is accomplished by the generation, documentation, and maintenance of evidence adequate to demonstrate that there is no fundamental deficiency of the design, performance, materials, construction, maintenance and testing that will prevent the equipment from performing its safety-related function. The purpose of equipment qualification is to eliminate potential common mode failures which could incapacitate essential safety functions. Random failures are minimized by surveillance. Additional assurance of system safety functions is provided by redundancy in components and features.

The general requirement for equipment qualification is contained in the General Design Criteria (GDC) 1, 2 and 4 of Title 10, Code of Federal Regulations, Part 50 (10CFR50), Appendix A and in Appendix B. For mechanical systems and equipment, General Design Criteria 14, 15 and 30 through 46 contain requirements for quality, in-service inspection, periodic pressure testing, and functional testing during the life of the nuclear power plant. In addition, GDC 34, 35, 38, 41 and 44 require suitable redundancy in components and features to assure that the following system safety functions can be met, assuming a single failure:

Residual Heat Removal (GDC 34)
Emergency Core Cooling (GDC 35)
Containment Heat Removal (GDC 38)
Containment Atmosphere Cleanup (GDC 41)
Cooling Water (GDC 44)
Containment Isolation (GDC 34, 35, 38, 41, and 44)

For other systems, GDC 55 and 56 require similar considerations for containment isolation valves. Similarly, GDC 22, 24, 26 and 29 require that the possibility of systematic, nonrandom, concurrent failure of redundant elements be avoided in the design of protection systems (electrical) and reactivity control systems (electrical and mechanical).

Industry-wide design standards for mechanical equipment have been in use for over fifty years. The design rules used by mechanical equipment manufacturers for mechanical equipment found in nuclear power plants are based on extensive design experience in conventional power plant, petrochemical plant and marine vessel applications. These mechanical design rules are often supplemented by well developed manufacturer's standards. The ASME Boiler and Pressure Vessel Codes and Piping Codes and associated ANSI Standards have been used for the design of primary pressure boundary components. In addition, well established quality assurance procedures were usually followed during the design and manufacture of mechanical equipment intended for use in nuclear power plants. These mechanical design rules have been endorsed and augmented by NRC Regulatory Guides and Standard Review Plans.

Construction details reveal fundamental differences between electrical and mechanical equipment. Electrical equipment makes much more use of non-metallic materials than does mechanical equipment. The non-metallic materials in electrical equipment are frequently an integral portion of the equipment, whereas, in mechanical equipment the non-metallic materials can usually be readily replaced during maintenance. The failure modes of electrical equipment are often subtle, difficult to analyze and may be more directly affected by aging processes, e.g., a short circuit caused by swelling of a material. On the other hand, since failures of mechanical equipment are generally structural in nature, they can often be characterized by mechanical analyses and are not usually related to aging. Even in the cases where aging may affect operability of mechanical equipment, the effects of aging are more predictable for mechanical equipment than for electrical equipment. Aging degradation in electrical properties of materials is often inferred from data showing degradation in such mechanical properties as elongation, compression set and viscosity. These known degradations in mechanical properties are directly related to degradations in the performance of non-metallic materials as used in mechanical equipment. The operability of mechanical equipment is dominated by metallic parts that can be affected by wear, thermal distortion, alignment and friction between moving parts. The dominant active role of non-metallic materials used in mechanical equipment is to minimize wear and provide lubrication. Most other roles are passive, that is, preventing leakage of the process fluid.

Safety-related electrical equipment has been subjected to a comprehensive qualification review following the issuance of NRC IE Bulletin 79-01B, Environmental Qualification of Class IE Equipment, January 14, 1980 and other related NRC NUREGs and rules. Much of this work has focused on defining the service conditions such as temperature, humidity and radiation and determining their effect on electrical equipment operability. Mechanical equipment is expected to be more resistant to these or similar service conditions because mechanical equipment is generally of rugged construction. In addition, the fundamental differences between mechanical and electrical equipment design and manufacture, as noted above, indicate that the qualification of mechanical equipment might be accomplished in a different fashion from the methods applied to electrical equipment.

In summary, the qualification process for mechanical equipment used in safety-related service in nuclear power plants should take into account the fundamental differences between electrical and mechanical equipment:

1. Most mechanical equipment found in nuclear plants is similar in design and construction to equipment found in use in conventional power plants, petrochemical plants, marine, and many other applications. These applications often have more severe temperature, humidity, pressure and dynamic environments than nuclear applications. Design evolution has been gradual and controlled by industry-wide design standards.

2. Although non-metallic parts are subjected to radiation not experienced in non-nuclear applications, the

ageable parts of mechanical equipment are almost always replaceable. Thus, preventive maintenance and parts replacement play a dominant role in qualification.

3. Mechanical equipment is designed for normal operating loads which can be much greater than superimposed seismic loads. Bearings and other internal load-bearing components routinely handle the stresses and deflections resulting from these significant normal operating loads. By comparison the only mechanical loads applied to electrical equipment occur for almost all cases during seismic events.

In addition, operating experience of mechanical equipment has an important role in qualification.

1. Most equipment undergoes some functional testing by the vendor and additional testing in the as-installed condition prior to plant startup (pre-operational tests).

2. The equipment is often installed in nuclear plants in configurations that allow periodic surveillance testing at operating conditions similar to those which would exist if it was required to perform its safety-related function.

3. Because of the similarities between nuclear and non-nuclear mechanical equipment, a great amount of historical data and experience exists on the operation of this equipment in various environments. Mechanical equipment manufacturers have factored this experience into their design improvements and material selections.

In general, mechanical equipment design and operating experience coupled with surveillance testing experience provides _prima facie_ evidence that existing mechanical equipment qualification practices are fundamentally adequate.

OBJECTIVES

The overall objective of this report is to compile the technical bases for qualifying mechanical equipment for safety-related service in nuclear power plants in a cost effective manner. The specific objectives are:

1. Identify types of mechanical equipment in domestic nuclear plants which may need to be qualified.

2. Recommend appropriate methods (historical, documentation, analysis and testing, other) that may be used to qualify mechanical equipment.

3. Develop guidance which the nuclear industry could follow to demonstrate qualification of mechanical equipment.

The guidance is based on the assumption that the mechanical equipment has been properly specified and proof tested at the appropriate ambient and elevated process conditions including operational dynamic loads. The purpose of qualification is then to show that the effects of aging, LOCA and seismic loads do not negate that proven ability.

APPROACH

The approach taken in this project is to compile and review information on the qualification of the types of mechanical equipment which are included in the scope. The sources of this information include published articles, reports and books, such as:

- Manufacturer's data on materials application
- Data on the effects of temperature and radiation on non-metallic materials
- Industry standards on mechanical equipment
- Final Safety Analysis Reports (FSARs) on determination of safety related active mechanical equipment
- Technical Specifications on frequency and acceptance criteria for surveillance testing
- USNRC Guidance on Equipment Qualification
- Earthquake Engineering Research Institute (EERI) reports on earthquake damage
- Seismic Qualification Utilities Group (SQUG) reports on equipment seismic capability
- The independent assessment of SQUG data by the Senior Seismic Review and Advisory Panel (SSRAP)

In addition, discussions with equipment qualification engineers at nuclear utilities and at mechanical equipment manufacturers provided useful insights for the achievement of the project objectives.

SCOPE

The scope of the report includes discussions of the general state-of-the-art methodology for environmental, dynamic and seismic qualification of active mechanical equipment. Guidance is presented for the qualification of valves, pumps, diesel engines, mechanical drive turbines, fans and snubbers, which constitute a large fraction of active safety related mechanical equipment used in nuclear power plants. Additional information on the related topics of seismic testing facility capability, seismic operating experience data and bearing capacity determination is presented.

Section 2 elaborates the report scope in a summary fashion to highlight the general elements of mechanical equipment qualification.

Section 2
SUMMARY GUIDE TO QUALIFICATION METHODS

Mechanical equipment qualification is the responsibility of the nuclear plant owner. The owner must provide assurance that the equipment is capable of performing all required safety functions during all design basis accidents, including earthquakes. Alternative methods of achieving this assurance are discussed extensively in this report. This section gives a summary guide of the elements of mechanical equipment qualification and refers the reader to the sections with more detailed coverage.

The general elements of mechanical equipment qualification are:

1. Identification of the equipment to be qualified
2. Description of safety functions and performance criteria
3. Identification of service conditions (both operational and accident)
4. Description of potential failure modes
5. Environmental qualification (including aging)
6. Seismic/dynamic qualification
7. Specification of maintenance/surveillance requirements to preserve qualification
8. Documentation

Methods of carrying out these elements are discussed briefly in the remainder of this section.

2.1 IDENTIFICATION OF EQUIPMENT

Mechanical equipment to be qualified is identified by determining the plant-specific components that play an essential role in systems with the following safety functions:

- Emergency Reactor Shutdown
- Residual Heat Removal
- Reactor Core Cooling
- Containment Isolation
- Containment Heat Removal
- Radioactivity Retention

Typical systems and the mechanical components usually found in these systems are given in Section 3.1 for both a PWR and BWR. In addition to the above components, all active mechanical equipment required to provide auxiliary support such as lubrication, cooling water, cooled air and diesel fuel oil to the mechanical equipment which perform safety functions should also be included. Although Section 3.1 can be used as a guide, the qualification list is unique for a specific plant.

The types of mechanical equipment addressed in this report perform an active motion to accomplish a safety function. In other words, operability rather than structural integrity is the primary focus of mechanical equipment qualification. Of course, the approach taken in the qualification of gaskets and seals could be extended to equipment which does not perform an active function. Section 3 presents a discussion of the definition of active, safety-related as used in this report and the scope of active, safety-related mechanical equipment installed in nuclear plants.

2.2 DESCRIPTION OF EQUIPMENT SAFETY FUNCTIONS AND PERFORMANCE CRITERIA

Equipment safety functions and performance criteria are derived from the system performance requirements, the plant licensing basis and the plant design basis events. This information is incorporated into the equipment specifications and the plant technical specifications.

It is important to explicitly identify what the equipment is expected to do and when its function is required. This should include whether it must function during or after any design basis event. The specific design basis event should be identified so that realistic, and not overly conservative service conditions can be established.

2.3 IDENTIFICATION OF SERVICE CONDITIONS

This report does not address the calculation of service conditions for specific plants. However, typical values are included in the report. More important to the qualification of mechanical equipment is a discussion of the important factors to consider in the determination of which qualification loads - environmental, seismic and dynamic - to impose on mechanical equipment. In particular, the report presents a basis for evaluating environmental loads separately from seismic and dynamic loads (see Section 3). Some of the equipment will be required to perform a safety function only when it is subjected to environmental loads. Other equipment will be required to perform a safety function only when it is subjected to seismic or dynamic loads. Equipment which performs safety functions in both cases can generally be shown to be subjected to either environmental or seismic/dynamic loads for a given function but not to both types of loads at the same time. An overall summary of the determination of appropriate qualification loads is presented in Section 3.4.

In addition to design basis loads, mechanical equipment experiences operational and process conditions during plant operation and equipment testing. With the exception of radiation, these conditions are comparable to those experienced by similar equipment in other industrial applications. Thermal and wear aging should be considered since they can be the limiting factor in establishing surveillance intervals and replacement frequencies for wearing parts.

2.4 POTENTIAL FAILURE MODES

The qualification of active mechanical equipment to perform safety functions should be based on a review of potential failure modes which could result from environmental, seismic or dynamic loads. For major types of active mechanical equipment, potential failure modes and appropriate qualification methods that address them are presented and discussed in Section 7, as follows:

Valves	Sections 7.1 to 7.7

Centrifugal Pumps	Table 7-6
Diesel Engines	Tables 7-8 and 7-9
Mechanical Drive Turbines	Table 7-11
Fans	Table 7-13
Snubbers	Table 7-14

2.5 ENVIRONMENTAL QUALIFICATION

Environmental qualification of active mechanical equipment is concerned primarily with non-metallic parts. Typically, in mechanical equipment, these non-metallic parts are replaceable. Preventive maintenance and surveillance are key elements in establishing the frequency of replacement for these non-metallic parts. The purpose of environmental qualification is to show that the effects of aging and LOCA do not negate the ability of the non-metallic parts to perform as demonstrated by performance and surveillance testing.

The environmental qualification of non-metallic materials should be based on determination of the material's capability to withstand the effects of appropriate environmental and process conditions without loss of capability to perform the required safety function. The qualification criteria for a specific material or component should be determined based upon the actual environment to which the material will be exposed during its required lifetime and the required safety function of the material.

The determination of qualification can utilize published data and manufacturer's experience with the non-metallic material in its application when exposed to the effects of thermal, wear and radiation aging. A material is considered qualified for its application if it satisfies one of the following criteria:

- The exposure conditions are less severe than the material capability.
- The non-metallic part can still perform its function as installed, until such time as it can be replaced (e.g., O-rings may still seal effectively although they could disintegrate during equipment disassembly).

- The non-metallic part can no longer be shown to be able to perform its function, but the mechanical equipment within which it is installed can still perform its safety function (e.g., a gasket may leak, but the pump still achieves acceptable head-flow characteristics).

If a non-metallic part does not satisfy one of the above criteria, alternate qualified materials are generally available for replacement use.

Environmental Evaluation Methodology

Guidance for environmental qualification of active mechanical equipment is discussed in Section 4, including the effects of environmental loads on the typical materials used in mechanical equipment and the importance of periodic replacement of parts in controlling the aging of components and limiting radiation exposure. Figure 4-1 is a flow diagram illustrating a methodology that can be applied to the qualification process. The methodology emphasizes the role of preventive maintenance and surveillance in the qualification process through a series of questions that address:

- Periodic preventive maintenance and disposition of parts susceptible to aging.
- Surveillance tests and the degree to which accident ambient and equipment temperatures are present in the surveillance test.
- The extent that exposure to chemical solutions or high radiation should be considered in the qualification process.

In general, a materials review should be conducted for equipment exposed to acid or caustic solutions. The following guidelines are suggested for materials review of equipment exposed to normal and post-accident radiation doses:

1. Oils and greases: Conduct a design review only if the integrated radiation dose exceeds 10^7R.
2. Organic materials: Conduct a review, analysis or test only if the radiation dose exceeds 10^4 R or if the equipment contains no teflon and the radiation dose exceeds 10^5R.

3. <u>Graphite and asbestos materials</u>: Waive further design review if the only non-metallic materials used in the equipment are graphite or asbestos based.

Figure 4-2 is a flow diagram illustrating a methodology that can be applied to the qualification of non-metallic parts used in applications where the accumulated exposure exceeds 10^5 R.

Examples of the use of the environmental qualification guidance presented in this report are included in Section 8.

2.6 SEISMIC AND DYNAMIC QUALIFICATION

Seismic and dynamic qualification of active mechanical equipment is concerned with assuring the operability of the equipment during or after it is subjected to seismic or dynamic loads (or both). It is important to note "during or after" since some equipment is not required to perform a safety function during an earthquake or if it is called upon, can satisfactorily perform the function when earthquake motion ceases. Performance testing and surveillance testing are key elements in establishing the basic operability of the equipment. The purpose of seismic and dynamic qualification is to show that the effects of earthquakes and other loads do not negate the ability of the equipment to perform its safety function.

Mechanical equipment is basically of rugged design because of the requirements to withstand high normal operating loads and process conditions which are comparable to typical LOCA environments. In addition, failure modes of mechanical equipment are predictable because, unlike electrical functional failure, they are almost always structural (fracture or deformation) or quasi-structural (interference of motion). Examples include:

- Shaft deflection
- Bearing overload
- Loss of running clearance
- Support overload
- Nozzle overload

Equipment manufacturers rely on experience and analytical predictions in the design of shafts, selection of bearings and determination of appropriate running clearances. Supports and nozzles are designed on the basis of ASME code requirements, which are also derived from experience and analytical techniques.

The timing of seismic/dynamic loading relative to when the active functioning of the equipment is required is an important consideration in seismic and dynamic qualification of active mechanical equipment. In many cases it can be shown that active equipment is not active and is not required to be active during seismic loading. A good example is the containment isolation valves. Many of these valves are closed normally and are only required to remain closed during an accident. Other containment isolation valves are open to allow water or steam flow during normal plant operation. Some of these valves may close as a consequence of main turbine trip which could result during an earthquake; these valves may require seismic qualification unless their failure to close during a seismic event can be shown to be inconsequential to plant safety. Dynamic qualification - considering only fluid blowdown loads - for isolation valves whose safety function is line break isolation is required for some valves. However, the superposition of seismic loads simultaneous with the blowdown loads may not be realistic for purposes of qualification.

The plant specific seismic/dynamic qualification criteria for active mechanical equipment should be developed for each type of mechanical equipment on the basis of its safety function, performance criteria, and potential failure modes as discussed in Sections 3 and 7.

Seismic/Dynamic Evaluation Methodology

The appropriate seismic/dynamic evaluation methodology for mechanical equipment can involve experience, analysis, testing or a combination of these. The most cost-effective method should be determined on a plant specific basis. Based on both technical and economic considerations, a general order of precedence for seismic/dynamic qualification is:

Method	Basis
• Qualify by Experience	Equipment similar to equipment already qualified or covered by earthquake experience data
• Qualify by Analysis	Failure modes are structural or quasi-structural
Static Method	Equipment is rigid
Static Equivalent	Conservative approach for non-rigid equipment
Dynamic Method	Used when static methods too conservative or the seismic load effects are complex
• In-situ Testing in Support of Experience or Analysis	Used to determine the natural frequency range of installed equipment
• Qualify by Testing - Shake Table	Functional failure modes difficult to address by analysis

Guidance on dynamic loading is included in Section 5. Details on the various evaluation methods are given in Section 6. In particular, the general process for seismic qualification is shown in Figure 6-4.

Examples of the use of the seismic/dynamic qualification guidance presented in the report are included in Section 8.

2.7 MAINTENANCE/SURVEILLANCE REQUIREMENTS

The role of surveillance and maintenance is important to the qualification of mechanical equipment. This is particularly true in the case of environmental qualification and also in the case of equipment operability aspects of seismic/dynamic qualification. Section 4 presents a discussion of the elements of surveillance and maintenance programs for environmental qualification. Section 7 provides guidance on the use of surveillance and maintenance programs for seismic and dynamic qualification.

2.8 DOCUMENTATION

Since proper documentation is a critical part of qualification, an equipment qualification file should be established for active mechanical equipment included in the program scope. The certificate of conformance should play a significant role in mechanical equipment qualification because of the greater reliance on experience data and analysis as compared with testing of specific equipment. Section 9 provides a discussion of qualification documentation.

Section 3
EQUIPMENT AND LOADS IDENTIFICATION

This section of the report discusses and provides guidance for identifying the mechanical equipment that is active and safety-related and guidance for determining the environmental, dynamic and seismic loads placed on this equipment during normal, accident and seismic events.

3.1 SAFETY-RELATED FUNCTIONS OF ACTIVE MECHANICAL EQUIPMENT

Tables 3-1 and 3-2 list the typical BWR and PWR safety-related systems with mechanical equipment and indicate the types of active mechanical equipment in each system. Additional columns indicate whether it is appropriate to qualify the mechanical equipment for earthquakes (SSE), accidents (LOCA), or both. It is recognized that traditional practice has been to seismically qualify all safety-related equipment. This practice is particularly important in the design of pressure containing components such as vessels, tanks, heat exchangers and piping. It has also been applied to the pressure parts of active components, such as valve bodies and pump casings. The mechanical design rules for pressure components are well-established and codified in the ASME Boiler and Pressure Vessel Codes and Piping Codes. These rules require that seismic loads be included in the component design. Because of the excellent experience with these code rules, further consideration of passive components is excluded from the scope of this report. In the case of active mechanical equipment, the timing and duration of equipment operation relative to the occurrence of postulated accident or earthquake is important. For example, some active mechanical equipment is only required to be capable of active functioning after an earthquake occurs but not during the earthquake. Other active mechanical equipment is intended to function only to mitigate the consequences of a loss of coolant accident (LOCA). The duration of operation is

TABLE 3-1

TYPICAL BWR SAFETY-RELATED SYSTEMS WITH MECHANICAL EQUIPMENT

	SAFETY/RELIEF VALVES	OTHER VALVES	PUMPS	MECHANICAL DRIVE TURBINES	FANS	PIPE SNUBBERS	CONTROL ROD DRIVES	DIESEL ENGINES	QUALIFY FOR			NOTES
									SSE	LOCA	BOTH	
Nuclear Boiler	x	x				x			x			(1)
Control Rod		x				x	x		x			(2)
Standby Liquid Control		x	x			x			x			
Emergency Core Cooling		x	x	x		x				x		
Residual Heat Removal		x	x						x			(1)
Standby Gas Treatment					x	x				x		
Essential Service Water		x	x			x					x	
Control Room HVAC					x					x		
Containment Isolation		x								x		
Combustible Gas Control		x			x					x		
Containment Spray			x							x		
Equipment Cubicle Cooling		x			x	x					x	
Reactor Core Isolation Cooling		x		x		x			x			
Isolation Condensers		x							x			
Emergency Power (diesels)			x					x			x	(4)
Suppression Pool Cooling		x				x					x	(5)

NOTES:
(1) Recirculation pumps excluded (see text).
(2) Control rod drives excluded (see text).
(3) Sometimes the RHR components are used for low pressure safety injection and residual heat removal. If so, qualify to both LOCA & SSE.
(4) Both LOCA and SSE because LOCA followed by main turbine trip and SSE by itself may cause loss of off-site power.
(5) Safe Shutdown may require safety relief valve discharge to the suppression pool.

TABLE 3-2

TYPICAL PWR SAFETY-RELATED SYSTEMS WITH MECHANICAL EQUIPMENT

	SAFETY/RELIEF VALVES	OTHER VALVES	PUMPS	MECHANICAL DRIVE TURBINES	FANS	PIPE SNUBBERS	CONTROL ROD DRIVES	DIESEL ENGINES	QUALIFY FOR			NOTES
									SSE	LOCA	BOTH	
Reactor Coolant	x										x	(1)
Control Rod							x				x	(2)
Auxiliary Feedwater		x	x	x		x					x	(3)
Component Cooling		x	x			x					x	
Chemical & Volume Control		x	x							x		
Emergency Core Cooling		x	x	x						x		
Hydrogen Recombiner		x			x					x		
Essential Service Water		x	x			x					x	
Containment Spray		x	x			x				x		
Containment Isolation		x			x					x		
Containment Fan Cooling		x			x						x	
Equipment Cubicle Cooling		x			x						x	
Control Room Cooling		x			x						x	
Residual Heat Removal		x	x								x	(4)
Emergency Power (diesels)			x					x			x	(5)

NOTES:
(1) Reactor coolant pumps excluded (see text).
(2) Control rod drives excluded (see text).
(3) Both LOCA and SSE because system is used to mitigate consequences of small break LOCA in addition to emergency reactor shutdown.
(4) Sometimes the RHR components are used for low pressure safety injection and residual heat removal. If so, qualify to both LOCA and SSE.
(5) Both LOCA and SSE because LOCA followed by main turbine trip and SSE by itself may cause loss of off-site power.

also important to the qualification process. For example, the equipment may be required to perform a very brief active function such as a valve closing or the equipment may be required to operate continuously such as a pump rotating and delivering flow for an extended period following a LOCA.

One of the purposes of addressing LOCA/SSE in Tables 3-1 and 3-2 is to emphasize that the specific safety function of the active mechanical equpment should be stated to guide the determination of appropriate qualification loads - environmental, seismic and dynamic - to apply to the equipment. These safety functions can be grouped in the following manner.

- Emergency Reactor Shutdown
- Residual Heat Removal
- Reactor Core Cooling
- Containment Isolation
- Containment Heat Removal
- Radioactivity Retention
- Support Systems

Emergency Reactor Shutdown. This is the capability to quickly stop the fission process and reduce the amount of heat generated in the core following LOCA and also following non-LOCA emergency events such as:

- Main steam isolation
- Loss of off-site power
- Loss of main condenser heat sink
- Loss of feedwater

Residual Heat Removal. This is the long term capability to remove core decay heat and maintain primary coolant temperature and pressure at or below normal operating conditions following normal or emergency reactor shutdown.

Reactor Core Cooling. This is the capability to cool the reactor core following a loss of coolant accident. The maximum LOCA is defined as a primary coolant system pipe break equivalent in size up to the double-ended rupture of the largest pipe in the reactor primary coolant system.

Containment Isolation. This is the capability to close all of the lines and ducts leading from the reactor containment to other buildings in the plant in order to prevent release of radioactivity following a LOCA.

Containment Heat Removal. This is the long term capability to remove heat from the primary containment. This is done by containment spray and by containment cooling (PWR containment fan coolers or BWR pressure suppression pool cooling). The PWR containment requires heat removal capability primarily following a LOCA, but also to a limited extent following overpressure transients which lead to emergency reactor shutdown. The BWR also requires containment heat removal following LOCA and during emergency reactor shutdown when there may be steam blowdown into the suppression pool for long periods of time.

Radioactivity Retention. This is the long term capability to remove fission products from the containment atmosphere and to minimize the need to vent and purge the containment atmosphere. This function is required only as a result of LOCA resulting in core damage. Another aspect is the retention of stored radioactive liquids and gases outside of containment which should be ensured during earthquakes, strong winds and floods.

Support Systems. This is the long term capability to provide auxiliary support such as cooling water, cooled air, diesel fuel oil, and emergency power to the equipment providing the safety functions previously discussed.

These safety functions are accomplished by a combination of mechanical and electrical equipment. In this report, the emphasis for qualification purposes is on active mechanical equipment. What is meant by the word "active"? It means that some part of the equipment must slide, rotate, reciprocate or otherwise move in order for the equipment to achieve its safety function. The equipment can be either automatically or manually actuated. The method of actuation can be an important consideration in the timing of the equipment operation relative to the actual incidence of an accident or earthquake.

Safety Related Systems

The safety related system designations in Tables 3-1 and 3-2 are reasonably self-explanatory, but the following should be noted since definitions are not necessarily consistent from plant to plant.

- The term "appurtenances" used with diesel generators includes turbochargers, fuel and lubricating oil pumps, any cooling water pumps dedicated to the cooling water requirements of the diesel engine, and any other on-engine or off-engine components provided for engine operation.

- Essential Service Water is that system or systems which provide cooling water to heat exchangers used for residual heat removal and containment cooling.

- Component Cooling System is that system or systems used to cool the active components of safety related equipment either directly such as by engine jackets or indirectly such as by area fan coolers and oil coolers. Component cooling may be performed by the Essential Service Water System.

- Control room cooling refers to the systems provided to make the reactor control room habitable following design basis events.

Active Safety-Related Mechanical Equipment

Tables 3-1 and 3-2 include the basic types of active mechanical equipment that can be the subject of environmental, seismic and dynamic qualification. These tables reflect information from a limited sample of plants, developed by reviewing FSAR's and applying the guidance contained in this section. Although the information is only typical, it can be used as a guide and is considered adequate for the purposes of this generic review of mechanical equipment qualification criteria and methods.

Valves are listed in Tables 3-1 and 3-2 as Safety Relief and other valves. To obtain a clearer idea of the types of other valves used in safety applications, a review was made of a typical BWR. This review encompassed the Emergency Core Cooling System and the Standby Liquid Control System but did not include containment isolation valves in other systems or valves in support systems such as cooling water. A tabulation of the quantity of each type of valve with a nominal size of at least four inches is given below:

NUMBER OF VALVES BY TYPE

SYSTEM	GATE	GLOBE	RELIEF	CHECK	EXPLOSIVE	TOTAL
HPCS	4	4	2	1		11
LPCS	2	1	2	1		6
SLC		2	2		2	6
RCIC	11	8	3	2		24
RHR	33	27	13	2		75
TOTALS	50	42	22	6	2	122

In a separate review of active containment isolation valve types for a PWR, the following tabulation was obtained:

TYPE OF VALVE	QUANTITY
Gate	12
Globe	48
Check	37
Butterfly	6
TOTAL	103

These two tabulations show that gate and globe valves are the most common types to be considered for qualification. Another common type, particularly in containment isolation applications, is check valves.

Equipment Excluded From This Report

For the purposes of this report, certain mechanical equipment has been excluded from detailed consideration. For example, mechanical equipment whose sole purpose is to perform a passive safety-related function is excluded:

- Pressure vessels
- Heat exchangers
- Tanks
- Piping
- Pipe supports

Also excluded is electrical equipment that supports the operation of mechanical equipment, for example:

- Electric motors
- Electric valve operators
- Electric solenoid valves

These are typically addressed in electrical equipment qualification programs but are considered in mechanical programs to the extent that they have mechanical properties (such as weight) affecting the qualification of the associated mechanical equipment.

In addition, primary coolant pumps, air compressors, hydraulic pumps, control rod drives and small pumps which make up normal water leakage from ECCS piping have been excluded. These are discussed below:

- Primary coolant pumps (PWR) and recirculation pumps (BWR). Operation of these pumps is not required after a design basis event and they are not driven by safety-related power sources.

- Even though they are used in safety-related systems, air compressors and hydraulic pumps are not safety-related if the systems are provided with fluid accumulators or receivers that continously hold sufficient fluid at required pressure to accomplish the system's safety function.

- Control rod drives are not discussed in detail in this report since they are addressed in detail in other programs.

- The small pumps provided to prevent draining of safety system injection lines and subsequent waterhammer on system initiation. These small pumps may be excluded from qualification if there is time for the operator to ensure that the main safety system pump - such as high pressure injection - starts to keep the lines filled. Protection against water hammer can be enhanced by providing backup emergency power to the small pumps rather than by imposing equipment qualification requirements on them.

3.2 PROBABILISTIC CONSIDERATIONS IN SSE/LOCA EVENTS

In addition to considerations of system safety functions, timing and duration of operation of mechanical equipment and passive/active distinctions, probabilistic considerations also support the view that SSE and LOCA should generally be treated separately in the qualification of mechanical equipment.

Studies (3, 4, 10) show that there is a very low probability of a LOCA being induced by a SSE event and that some systems and components, such as isolation system valves, are provided only to mitigate a LOCA. Other components are provided only to achieve safe shutdown following non-LOCA events such as an SSE that could result in loss of the main condenser. A considerable saving, as well as a realistic prioritizing of qualification requirements, would result if the equipment required only for LOCA were not required to be qualified for the SSE and equipment required only for non-LOCA events were qualified for SSE but not LOCA environments.

Tables 3-1 and 3-2 categorize systems according to whether they are required for LOCA or SSE or both. Note that qualification to "both" means separate (i.e. not concurrent) application of load types. The major impact of this method of classifying qualification requirements in this manner is to:

1. Eliminate the need to SSE qualify the active function of containment isolation, vent and purge valves and ECCS equipment.
2. Eliminate the need to qualify dedicated emergency makeup and RHR system components for LOCA harsh environments even if they are located in a harsh environment resulting from a LOCA. (Some RHR systems are used both for LOCA and to achieve shutdown.)
3. Eliminate the need for many seismic braces and snubbers on non-primary pressure boundary ECCS piping and equipment, thus providing better access for surveillance and maintenance.

In the event an SSE occurs, the containment isolation valves and ECCS equipment and piping - if qualified only for LOCA - would have to be post-SSE qualified for continued operation. But this would be on a plant specific basis after the SSE event as part of an overall requalification program.

10 CFR 100 Appendix A (1) defines as being subject to the criteria of Appendix A - Seismic and Geological Siting Criteria for Nuclear Power Plants - those structures, systems and components which ensure:

(a) The integrity of the reactor coolant pressure boundary, or

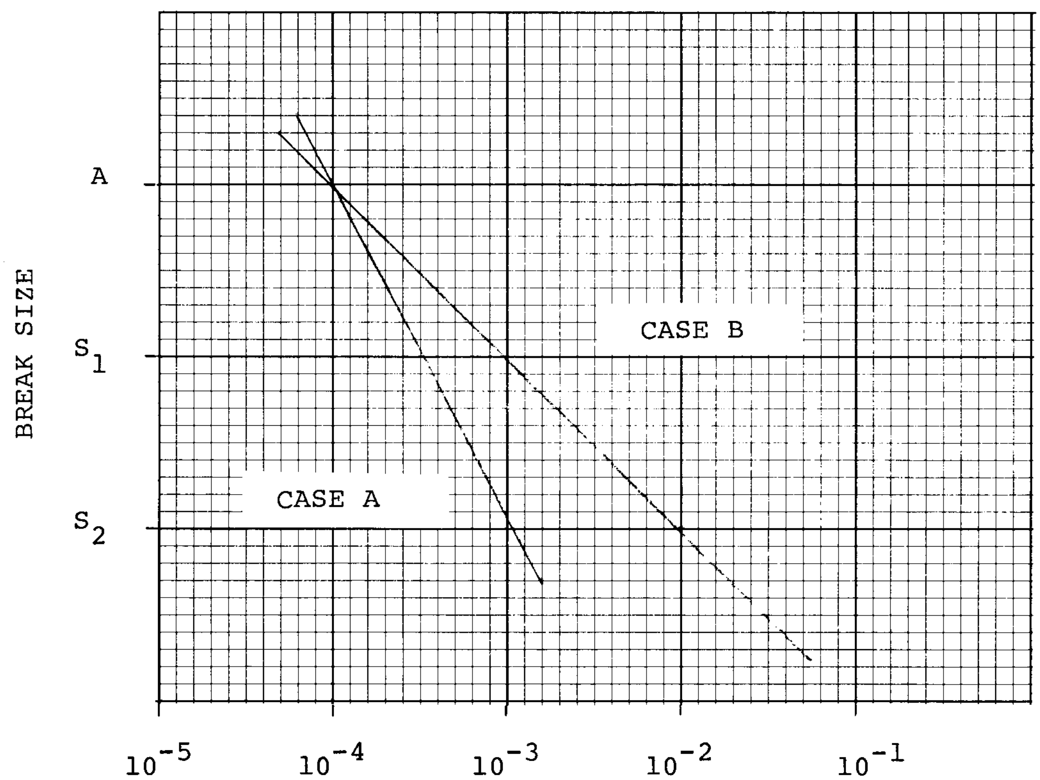
(b) The capability to shut down the reactor and maintain it in a safe shutdown condition, or

(c) The capability to prevent or mitigate the consequences of accidents which could result in potential offsite exposures comparable to the guideline exposures of 10 CFR 100.

The requirement for consideration of Loss-of-Coolant Accidents (LOCA) includes circumferential severance of the largest reactor coolant pipe as formalized in Appendix A to 10 CFR 50 (2). The occurrence of a pipe rupture is also a small probability event due to the stringent design criteria, the use of material which is highly resistant to fracture, and the comprehensive preservice and inservice inspection techniques employed. The low likelihood of occurrence of these events and the capability of the system to withstand the events assure that there is a low probability of compromising the structural integrity of the primary coolant system design.

Studies were referenced in the course of preparation of WASH-1400 (3) that assessed the probability of pipe failure. Included were data from nuclear sources, industrial sources, and a number of other sources. Various pipe sizes were considered in the evaluation, and the probabilities were assessed for different pipe diameter ranges.

S. H. Bush (6) assessed the reliability of piping in light-water reactors based upon several non-nuclear and nuclear piping and vessel failures. An annual probability of 10^{-5} to 10^{-6} was found to encompass catastrophic or disruptive type failures and 10^{-4} to 10^{-5} including such less severe failures as pipe leakage. S. H. Bush (7) suggests that the probability of cracking (though not necessarily a failure which should be considered a LOCA) is 1×10^{-2} per reactor year for piping susceptible to intergranular stress corrosion cracking (IGSCC). A more recent study prepared for the BWR Owners Group and EPRI (8) showed a pipe size dependency for piping susceptible to IGSCC. The probabilities summarized in Figure 3-1 show no increase for large breaks and only one order of magnitude increase for small breaks. Note that a large break means a rupture of a liquid line of diameter greater than six inches or of a steamline of diameter greater than eight inches.



CASE A: WASH 1400 BASE CASE
CASE B: WASH 1400 BASE CASE MODIFIED FOR POSSIBLE IGSCC

A = Large Break

6" Liquid Line
8" Steam Line

S_1 = Intermediate Break

2½-6" Liquid Line
4½-8" Steam Line

S_2 = Small Break

½-2½" Liquid Line
1-4½" Steam Line

FIGURE 3-1: MEDIAN LOCA INITIATING RUPTURE RATES PER PLANT YEAR (Reference 8)

A concern may exist that these events may be dependent, i.e., the occurrence of the SSE event may lead to a LOCA. Reference (4) demonstrated that for a typical plant, the probability of a LOCA resulting from an SSE is very small. These studies included consideration of undetected flaws in the pipe and concluded that, even with flaws much larger than those realistically expected in nuclear power plant piping, and in the worst location, the conditional probability of a LOCA resulting from large earthquake motions is on the order of 10^{-6} per SSE. It should be emphasized that this is a conditional probability and therefore does not include the probability of occurrence of the limiting earthquake itself. Based upon these probability values, postulation of an SSE induced LOCA does not constitute a reasonable design basis.

Reference (9) estimates the probability of the simultaneous occurrence of SSE ($P=10^{-4}$ events/year) and LOCA ($P=10^{-6}$) is estimated to be 10^{-10} per year. This probability is significantly below the probability level of 10^{-6} which is generally required for event evaluation. Even if the strength of the LOCA dependency could be several orders of magnitude greater than that calculated in Reference (4), and the probability of joint SSE and LOCA occurrence would still be below 10^{-6} per year. Reference (10) estimates that the probability of a leak and LOCA due to pipe cracks in a PWR is 10^{-6} and 10^{-12} per plant life (respectively) and states the probability of a seismic induced LOCA is even lower.

Since the probability of a seismic induced LOCA is less than the probability level requiring evaluation, and safe shutdown of a nuclear plant is required following an SSE, the guidelines for qualification of active mechanical equipment could be as follows:

1. Active mechanical components required only to mitigate the consequences of LOCA and other low probability accidents with low dependence on SSE shall be qualified to normal and LOCA dynamic loads and normal, abnormal and accident environmental conditions (not SSE).

2. Active mechanical components required only for safe shutdown shall be qualified for normal and abnormal dynamic loads and environmental conditions and SSE (not LOCA).

3. Active mechanical components required for both the functions of (1) and (2) above shall be qualified for normal, abnormal and LOCA dynamic loads and environmental conditions and SSE (all), but not concurrently.

3.3 DISCUSSION OF QUALIFICATION LOADS

The determination of appropriate qualification loads - the environmental, seismic and dynamic conditions to which safety-related mechanical equipment can be subjected - should be based on the factors discussed above:

- System safety function
- Timing/duration of equipment operation
- Passive/active distinctions
- Probabilistic considerations

The occurrence of a safe shutdown earthquake would not lead to the loss of pressure integrity of passive mechanical equipment, such as vessels, piping, tanks and heat exchangers which have been seismically designed. In addition, the pressure boundary of active mechanical equipment would also remain intact by virtue of its design to the ASME code. For this reason, the environmental load, resulting from a LOCA or HELB, is not required to be concurrently applied with seismic loads to assure the operability of mechanical equipment. This means that it is appropriate to decouple environmental loads from seismic (and dynamic) loads. The environmental loads should only be applied to non-metallic parts of active mechanical equipment. The seismic and dynamic loads should be applied to those parts of active mechanical equipment whose failure could prevent the active functioning of the equipment.

The calculation of the magnitude of the environmental, seismic and dynamic loads is beyond the scope of this report. However, some typical loads are indicated for purposes of comparison.

Table 3-3 includes the types of active, safety-related equipment found in nuclear plants and typical environmental conditions based on a limited sample of nuclear plants. Three general locations are shown:

TABLE 3-3

SUMMARY OF EQUIPMENT TYPES AND TYPICAL ENVIRONMENTAL LOADS BY GENERAL LOCATION

GENERAL LOCATION	TYPES OF EQUIPMENT	TYPICAL ENVIRONMENTAL LOADS
Inside of Containment	Isolation Valves Fans Snubbers Control Rod Drives Safety/Relief Valves Main Steam Isolation Valves	Temperature Normal 65-120^{o}F Accident 340^{o}F Peak Pressure 50 psig Relative Humidity <=100% Radiation Normal $3.5x10^{6}$R Accident $2x10^{8}$R
Outside of Containment in HELB or ECCS recirculation area	Pumps Valves Fans Snubbers	Temperature Normal 65-130^{o}F Accident 148-212^{o}F* Relative Humidity 95% Radiation Normal $3x10^{5}$R Accident $1x10^{7}$R
Other Areas not subject to HELB or ECCS recirculation	Pumps Valves Fans Snubbers Diesels Compressors	Temperature Normal 95-110^{o}F Accident 148^{o}F Relative Humidity <=100% Radiation Normal 10^{3}R Accident 10^{4}R

*The temperature due to a steam line or high temperature line break.

- Inside of Containment. This location is subject to the highest radiation, temperature and pressure, because it is subject to the full effects of a primary coolant loss-of-coolant accident (LOCA).

- Outside of Containment. This includes areas subject to high energy line breaks (HELB) and recirculation of radioactive fluid following a LOCA.

- Other areas. These are other areas not subject to a HELB or ECCS recirculation. The temperature and radiation levels are always relatively low.

In reference to electrical equipment qualification, the first two of these are called "harsh" environments while the third is called "mild."

Environmental Conditions Guidelines

- Accident bulk temperature inside containment is usually based on the isenthalpic expansion of primary coolant to containment pressure. This results in temperatures of approximately 340°F. The temperature may be as high as 430°F for steam line breaks inside containment of PWRs with once-through steam generators and superheated steam conditions.

- Accident bulk outlet temperatures outside containment may be as high as 212°F and are based on coolant losses from high energy line breaks being limited by rapid isolation and the coolant losses flashing to atmospheric pressure and rapidly coming to thermal equilibrium as saturated steam at atmospheric pressure.

- Direct impingement of a high energy line break need not be considered as an environmental condition on safety-related mechanical equipment if the broken or leaking line disables the safety function of the mechanical equipment, and the same safety function can be accomplished by other means, including redundant equipment which would not be disabled by the line break. For example, many plants have steam turbine driven auxiliary feedwater systems. These do not have to be qualified for the effect of a leak or break in a steam supply line if there is sufficient physical separation to prevent impingement from affecting redundant systems providing auxiliary feedwater.

- If safety-related mechanical equipment is redundant or located in separate cubicles or rooms each with redundant environmentally qualified safety-related room coolers then the abnormal or accident temperature can be based on operation of the room coolers.

- Radiation doses to equipment are usually based on the guidelines of TID 14844 (64) as endorsed by NUREG 0588 (65).

Dynamic Loads

Dynamic loads are those transient loads that are not seismic and are not encountered in normal plant operation and surveillance testing. Identification of such loads requires answers to the following questions.

- Are there unusually high flows through the equipment during design basis events that may affect the function of the equipment during or after the event?

- Are there unusually high vibratory loads transmitted to the equipment through building structures and connecting systems during the event?

Two important dynamic loads identified in this report are (1) two phase flow through isolation valves as they close following a break in its associated downstream high energy piping, and (2) BWR suppression pool hydrodynamic loads. These are discussed in Section 5.0.

Seismic Loads

Table 3-4 summarizes typical seismic conditions obtained from a sample of nuclear plants with a design basis horizontal ground acceleration of 0.2g. Vertical accelerations, as shown in Table 3-4, are equal to 2/3 of the horizontal acceleration at the same location within the building. This approach to vertical accelerations is common to older plants which were designed and built prior to the recommendations of Regulatory Guide 1.60 to model the vertical response of the building structure separately from the horizontal response. The extensive seismic reviews conducted as part of the Systematic Evaluation Program (SEP) confirmed the acceptability of using vertical accelerations as 2/3 of the horizontal acceleration. A recent recommendation, based on the extreme conservatism of the current definition of the vertical ground design response spectrum in Regulatory Guide 1.60, supports the SEP approach:

The vertical ground design response spectra values should be taken as 2/3 of the values specified for the horizontal ground design response spectra, across the entire frequency range (81).

Three different general locations have been selected as typical of the magnitudes of seismic loading which apply to active safety related mechanical equipment:

Basemat: This is the bottom slab or foundation of the reactor building, containment building or auxiliary building. This is the lowest seismic intensity location in the plant, and generally is the location of much of the large rotating equipment such as pumps and diesel engines.

Floor or Structure Mounted: This is the floors and structures above the basemat. Structural amplification results in higher seismic loads for floor or structure mounted equipment than for basemat mounted equipment. This is the location for smaller pumps and for area cooling equipment.

Pipe Mounted: These are equipment devices (namely valves) that are mounted into piping systems and that use the piping as the chief means of support. The seismic accelerations are imposed on the valves by the piping system. The results of piping system dynamic analysis with respect to seismic accelerations on pipe mounted equipment are often not represented as seismic response spectra. The results shown in Table 3-4 are conservatively estimated from the methodology discussed in Section 6. A note of caution on the use of the data for pipe mounted equipment shown in Table 3-4 and discussed in Section 6: The accelerations are those imposed on the equipment at the connection into the piping systems. These accelerations must be amplified at the valve extensions, such as the valve operator, unless the extension can be shown to be rigid.

TABLE 3-4

SUMMARY OF EQUIPMENT TYPES AND TYPICAL SEISMIC LOADS
BY GENERAL LOCATION
(0.2 g SSE Ground Acceleration)

GENERAL LOCATION	TYPES OF EQUIPMENT	TYPICAL SEISMIC LOADS
Basemat	Large Pumps Drive Turbines Diesel Engines Compressors Fans	Accelerations: Horizontal: Peak: 1.2 g ZPA: 0.3 g Vertical: Peak: 0.8 g ZPA: 0.24 g
Floors and Structures	Small Pumps Fans Valves*	Horizontal: Peak: 1.8 g ZPA: 0.5 g Vertical: Peak: 1.2 g ZPA: 0.3 g
Pipe Mounted	Valves	Horizontal: Peak: 4.5 g** Vertical: Peak: 4.5 g**

* Connected to structure such as containment by short, stiff piping sections.

** Acceleration at piping connection: This peak is the ZPA response of valve and its extensions if they are rigid.

Section 4

ENVIRONMENTAL QUALIFICATION OF MECHANICAL EQUIPMENT

This section of the report reviews and discusses environmental qualification of mechanical equipment used in nuclear power plants. Several topics are included.

- A discussion of the issue of aging in mechanical equipment and the important roles played by preventive maintenance, surveillance testing and the ability to replace or refurbish ageable components in mechanical equipment.

- A guide for an environmental qualification methodology for mechanical equipment. Fundamental to this methodology is the need for a preventive maintenance program recognizing the need for periodic inspection, refurbishment and replacement of ageable components.

- Data and evaluation of some non-metallic materials used in mechanical equipment with emphasis on exposure to radiation, a consideration unique to nuclear power plant applications.

Mechanical equipment, such as pumps, valves and engines used in nuclear power plants consist of an assemblage of metallic parts and non-metallic parts such as gaskets, seals, bearings, diaphragms and lubricants. The parts may be required to maintain an active function of the component, such as rotation, or a passive function, such as prevention of leakage. Both metallic and non-metallic parts must be considered and evaluated in the environmental qualification of a component.

Environmental qualification in its broadest sense is an evaluation of the combined effects of wear, temperature, humidity, chemicals, ambient vibration, pressure and radiation. The net effect of these environmental stresses is to cause changes in the properties of the parts affecting operability. Aging is considered to be the changes in properties resulting from accumulated exposure to plant normal operating conditions over the useful life of the part. Additional change may occur due to increased, shorter term environmental stresses caused by plant abnormal and accident conditions. The environmental qualification evaluation must necessarily consider that the component is aged by normal plant operating conditions at the time the abnormal or accident condition occurs. The issue of aging, the roles of surveillance and maintenance programs, and the replaceability of certain components in mechanical equipment play very important roles in qualification. These are discussed below as they relate to qualification of mechanical equipment.

REPLACEMENT OF AGEABLE PARTS

Mechanical equipment has been traditionally designed for replacement of parts that are subject to aging. The parts are usually those that are most subject to wear and to environmental degradation.

Parts subject to wear include bearings, piston rings, pump wear rings and special, close tolerance components where sliding action can take place, such as guides. Environmental degradation is most significant in the non-metallic materials such as lubricants, gaskets, hoses, packing, and diaphragms. In addition to wear, the important aging mechanisms for non-metallics are temperature, humidity, and radiation. An important aspect of the application of non-metallic materials is that they are usually located within the equipment where they are subject to the equipment conditions, not ambient environment, as the aging mechanism. For example, a diesel engine cylinder head gasket is subject to engine temperature when operating, not ambient temperature. On the other hand, the radiation dose may be considerably less than the external environment because the part may be totally enclosed and shielded from Beta radiation by metal.

OPERATING ENVIRONMENT

An important fact to note is that most of the safety-related mechanical equipment in nuclear plants is used in almost identical form in non-nuclear service such as fossil power plants and petrochemical facilities, where temperature, humidity and chemical corrosiveness can be more adverse than in nuclear plants. This may be an important consideration in the environmental qualification of the equipment for nuclear plant applications. Historical application of mechanical equipment in non-nuclear service may be a complete basis for qualifying its application in nuclear plants if the latter application involves

- Comparable temperature conditions, and
- "Mild" radiation environmental conditions, or
- "Harsh" radiation environmental conditions which are not the limiting factor in setting the frequency of replacement of ageable parts

This aspect is discussed in more detail later in this section of the report.

SURVEILLANCE AND MAINTENANCE

The role of surveillance and maintenance is important to qualification of mechanical equipment. Surveillance and maintenance programs encompass the following elements:

Preventive Maintenance: The role of preventive maintenance in qualification is to prevent equipment parts from degrading by normal aging processes to the extent that functional failure could occur under a plant abnormal transient or accident. Preventive maintenance is a scheduled program of regular maintenance activities such as replacement of oil, lubrication of bearings or replacement of a seal. Procedures for the preventive maintenance program are developed, taking into account equipment manufacturer's recommendations and operating experience. Wherever applicable, preventive maintenance procedures specify the nature of the specific post-maintenance testing requirements and acceptance criteria. These procedures are periodically reviewed and updated to incorporate new and additional information obtained from the manufacturers, experience in plant operations, licensee event reports, NRC bulletins and information notices, and from equipment qualification tests. The preventive maintenance activities are generally documented and include descriptions of abnormalities noted and corrective actions taken.

Surveillance Program: The role of surveillance is to monitor and record equipment performance and to schedule and perform corrective action, such as maintenance, when required. For active mechanical equipment this includes the scheduled functional testing of safety-related active mechanical equipment under conditions approximating conditions that may exist when the safety function is required and monitoring of important parameters such as pump head and flow, valve closing time, engine temperature and equipment vibration during the testing.

4.1 ENVIRONMENTAL FACTORS

The environmental qualification of mechanical equipment is based on a determination of the equipment's capability to withstand the effects of environmental and process conditions to which it is exposed, if such exposure could cause the loss of its safety function. These factors include:

- Radiation
- Wear
- Chemicals
- Pressure and Relative Humidity
- Temperature
- Vibration

<u>Radiation</u>

Exposure of equipment to normal and accident radiation sources is a unique condition not found in non-nuclear applications. Section 3.0 notes that the integrated normal radiation exposure over the plant life can vary from 10^3 Rads to 10^7 Rads, depending on location within the plant. For replacement parts, the frequency of replacement will reduce normal radiation exposure accordingly. The integrated normal radiation exposure plus design basis accident radiation exposure can vary from 10^4 to $2x10^8$ Rads, again depending on location within the plant. Radiation then becomes a unique environmental condition that must be factored into the total environmental qualification process.

Table 4-1 is a chart showing the integrated doses required to cause radiation effects in materials (<u>17</u>). According to this information, metals are not affected by doses expected under the most severe radiation encountered.

TABLE 4-1

DOSE LEVELS FOR RADIATION EFFECTS
(Reference 17)

Rads	
10^{12}	Aluminum Alloys, Stainless Steel Ductility Reduced Stainless Steel Strength Triples
10^{11}	Carbon Steel Loses Ductility
10^{10}	All Lubricants Gel
10^{9}	Some Metals Increase Yield Strength All Lubricants Thicken
10^{8}	
10^{7}	Common Lubricants Usable
10^{6}	All Lubricants Evolve Gas
10^{5}	Glass Changes Color

Organic materials such as lubricants are less resistant to radiation. This is because organic materials are held together with relatively weak chemical bonds as compared to those of metals. As a result, the parts of mechanical equipment that may be radiation sensitive are non-metallics such as the organic materials used in seals, gaskets and lubricants. A review of non-metallic materials properties is given in Section 4.3. The conclusion of that section is that (a) the threshold of radiation effect on non-metallics is 10^5 Rads except that the threshold for oils and greases is 10^7 Rads and (b) asbestos and carbon based materials have thresholds of 10^9 Rads or greater.

Environmental conditions other than radiation can limit the useful life of a component and determine the replacement cycle of the component. These include:

- Normal wear of metallic and non-metallic parts
- Dryout of certain non-metallics such as packing
- Evaporation of certain constituents of oils and greases

In summary, radiation is a unique, but not necessarily controlling, factor in environmental qualification of mechanical equipment for nuclear safety-related service. Other factors, such as wear and dryout, can require more frequent inspections and parts replacements. Periodic replacement of parts can be a method of both controlling the aging of components and limiting radiation exposure.

Wear

Table 4-2 lists the types of parts that are subject to wear in mechanical equipment. The operational effect of wear, as noted in Table 4-2, can be addressed in a surveillance test program by monitoring such parameters as vibration, flow and leakage. Wear should accumulate slowly from equipment operation and can be accounted for in planned preventive maintenance. For example, 16,000 hours of operation is recommended for major overhaul, including some bearing and inspection and cylinder replacement, of diesel engines by one manufacturer (73). Since wear is strongly dependent on duration of operation, periodic surveillance testing does not contribute significantly to wear unless the test conditions are significantly different from the equipment design conditions. For example, the unbalanced rotor loads in a centrifugal pump at low flow during a test can be significant compared with those which exist at design flow. In general, the test conditions will be at least as severe for consideration of wear as those which are postulated for an accident or earthquake. The comparison of severity can be readily handled in the qualification process as discussed below.

TABLE 4-2

MECHANICAL EQUIPMENT WEAR FACTORS

EQUIPMENT	PART	OPERATIONAL EFFECT
Diesel Engine	Bearings	Vibration
	Piston Ring	Output, oil consumption
	Oil Seals	Oil consumption
Pumps	Bearings	Vibration
	Wear Rings	Head-flow
Valve	Packing wear	Stem leakage
	Packing swelling	Stem friction
Air or Hydraulic Valve Operators	Pistons & Rings	Force, leakage
	O-rings	Compression set,
	Diaphragms	Flexibility

Chemicals

Chemical solutions find limited use in nuclear power plants. These include:

- Sodium Pentaborate Reactivity Control Solution - BWR Standby Liquid Control System
- Boric Acid Reactivity Control Systems in PWRs
- Iodine Getters in Containment Spray Systems of PWRs. These are usually a caustic solution of sodium hydroxide or sodium thiosulfate in water.

The capacity to withstand the effects of these solutions is inherent in the materials of construction selected for the equipment. These solutions have been extensively evaluated in electrical equipment programs due to the potential for electrical faulting with chemical solutions. The problem is much simpler for mechanical equipment and involves the correct choice of materials to withstand corrosive attack. This requires the use of stainless steel for equipment processing boric salts and acids and avoiding materials such as aluminum that are vulnerable to attack by caustic solutions if such solutions can come in contact with the equipment. Both carbon steel and stainless steel and most non-metallic materials used in mechanical equipment perform well in caustic solutions.

Qualification of equipment for chemical solutions in nuclear power plants is best handled by design review of materials used and in contact with the solution. In addition, surveillance and maintenance are important to prevent a significant buildup of precipitated chemical crystals.

Pressure and Relative Humidity

These two environmental factors are listed for completeness. They usually have an insignificant effect on operability of mechanical equipment, whose operability is influenced by internal operating conditions, not external environments that cannot penetrate the passive boundary of the equipment or that is masked by the internal environment, e.g., internal wetted surfaces and process fluid pressures.

Temperature

The internal operating temperature of a component can have much more of an effect on the operability of a mechanical component than the external environment. For example, diesel engine temperatures are much higher than ambient temperatures. The fluid in a pump or valve may be higher or lower temperature than ambient, and, in either case, will control the basic temperature of the equipment.

Temperature as an environmental factor affecting operability may have to be considered from two standpoints.

1. The effect of a temperature increase on equipment clearances and dimensions affecting operability. In most cases operability can be demonstrated by the surveillance test that operates the equipment under anticipated safety-related conditions, e.g., cold-quick start, rapid heatup and operation at high internal temperatures. Examples are quick start and loading of diesel engines and mechanical drive turbines and testing of normally open primary system isolation valves. Some equipment cannot be surveillance tested under anticipated safety-related temperatures due to plant limitations. For example, containment isolation valves cannot be surveillance tested at post-LOCA containment temperature and ECCS

pumps may not be capable of being surveillance tested with process fluid temperature representative of those following LOCA. In these cases, an analysis of differential thermal expansion may be appropriate.

2. The effect of elevated temperature on degradation of non-metallics such as oil and elastomers. These materials require controlled operating temperatures in order to function properly. Cooling is often provided for these materials by safety-related component cooling water and component or area coolers. For example, the oil may be cooled by a heat exchanger or the oil in the sump may reject heat to the ambient. As a rule, non-metallics may be qualified by surveillance tests when the operating temperature of the test is approximately the temperature anticipated during the safety-related event. If high temperatures are anticipated to exist during design basis events, then a material evaluation should be performed.

Vibration

Vibration is an environmental factor which should be considered in qualification. It can manifest itself in two ways.

- Ambient vibration or vibration transmitted to the equipment through piping connection and mounting by sources external to the equipment. For standby equipment this is the main source of vibration.
- Vibration created by operation of the equipment. Examples are vibration caused by operation of a reciprocating engine, by flow, or by unbalance in a rotating element.

There are several aspects of vibration which need to be discussed within the perspective of equipment qualification:

- Excessive vibration is often observable by the plant operator and involves engineering judgement as to acceptability.
- Normal vibration does not usually cause failures in the major metallic structural and operational components of equipment because the stresses caused by vibration are below the endurance limit. Vibration amplitudes large enough to cause vibration stresses beyond the endurance limit in such components as equipment bodies, shafts or crankshafts will usually be manifested by early failure or by external evidence such as visual vibration or sound.
- Normal vibration can be amplified and lead to failure in the smaller, "tuned" components of equipment such as bracket mounted components, hoses and components secured by locking devices such as set screws.

Failure from these causes is prevented by periodic replacement, restraint and improving the locking device.

In summary, qualification of vibration is mainly a process of observation and inspection during pre-operational and surveillance testing so that early failure modes can be detected and corrected and vibration prone components can be properly secured against failure.

4.2 NON-METALLIC MATERIAL APPLICATIONS

The important non-metallic material applications for mechanical equipment are discussed below.

Bearings

There are several types of bearings used in safety-related mechanical equipment.

(A) Process water lubricated journal bearings. These are bearings that are submerged in and are lubricated by the water being pumped. The bearing material is usually metallic or graphite and is unaffected by coolant temperature or radiation. Most of the ECCS related pumps have bearings lubricated by the process water.

(B) Oil and grease lubricated journal bearings. Pumps that are oil lubricated are high head, multistage units used in such service as BWR RCIC and HPCI service and PWR auxiliary feedwater and chemical and volume control charging pump service. Some Residual Heat Removal and Containment Spray Pumps also use oil lubricated bearings. The bearings are metallic and are not subject to deterioration due to temperature or radiation.

Oil lubricated bearings are also used in diesel engines and mechanical drive turbines.

(C) Self-lubricated bearings. These are bearings which employ non-metallic materials with self-lubricating properties. Materials include teflon. These types of bearings, if used in nuclear plants, will be found in light load application such as fans.

(D) Rolling element bearings. This type falls into two categories, ball bearings and roller bearings. Both use metal materials for the rollers and the races and are grease or oil lubricated. Rolling element bearings find application in small, lightly loaded pumps and fans and in gear reducers and speed drives.

In summary, practically all heavy-duty bearings found in pumps, engines and mechanical drive turbines are metallic. Many depend on oil or grease lubrication. The application of non-metallic bearings is limited to lightly-loaded components. The non-metallic materials commonly used in the latter application are teflon and nylon. Graphite is not considered in the same category as a non-metallic and has radiation damage resistance comparable to metals.

<u>Lubricants</u>

Lubricants in the form of oil and grease, are used in mechanical equipment rotating shafts and bearings. Oil is used in rotating equipment such as pumps and turbines and in reciprocating equipment such as diesels. Greases are used to pack bearings and also for some valve shafts and valve operator shafts.

Although oils and greases are fairly resistant to radiation, they are not immune from other aging mechanisms including thickening and hardening caused by evaporation and chemical processes. These require replacement of oils and greases at intervals more frequent than those which may be required by radiation damage.

<u>Gaskets and Stem Packing</u>

Gaskets are used primarily in flanged joints as, for example, between a valve body and bonnet. Typically, these gaskets are of the asbestos type: controlled compression, spiral-wound asbestos with metal inserts. In addition, they are generally fully retained on all sides by metallic surfaces. This feature protects the gasket from continuous direct Beta radiation exposure either from the process fluid or from the external ambient environment.

For safety-related pumps, it should be noted that a gasket failure or a shaft seal failure does not necessarily result in the loss of active function. It results in leakage that may be a small fraction of the rated system capacity. Leakage may affect component operability if the leakage should impinge on a part required for operability or if the leakage should contaminate the lubricating oil. Common configurations for a pump often do not allow gasket leaks to mix with the lubricating oil. In the case of the horizontal pump a gasket leak may be in the radial direction, away from the axis of the shaft, so that the leak would not flow to an oil cooled bearing box. Leakage along the shaft from the mechanical seal may be deflected from the gear box by a leakage deflector installed on the shaft for that purpose.

O-Rings and Diaphragms

Other applications for sealing between mating metallic surfaces use elastomeric O-rings. The important parameter of elastomers used in O-ring seals is called compression set which is a loss in resiliency that can result in reduced sealing capacity, but not necessarily functionality. Elastomers are also used as diaphragms in air operators for valves. This application is discussed in more detail in Section 7.

TABLE 4-3

IMPORTANT CHARACTERISTICS OF MECHANICAL EQUIPMENT NON-METALLIC MATERIALS

COMMON NON-METALLIC APPLICATIONS	IMPORTANT PHYSICAL CHARACTERISTICS
Lubricants	Viscosity
Gaskets	Compression Set
Packing	Compression Set
Seals	Compression Set
O-Rings	Compression Set
Diaphragms	Elongation, Tensile Strength
Hoses	Elongation, Tensile Strength
Hydraulic Hoses	Viscosity

4.3 NON-METALLIC MATERIAL PROPERTIES

The characteristics of non-metallic materials which are important in ensuring the functional performance of the material as applied in mechanical equipment are shown in Table 4-3. The extent to which these characteristics can be degraded by environmental factors is the subject of environmental qualification. The most important of the environmental factors to consider for mechanical equipment are the effects of radiation and temperature since the system process conditions were addressed in the design specification.

Radiation and Temperature Effects

Tables 4-4 and 4-5 list information on allowable service temperatures and radiation exposure for several common non-metallic materials and lubricants used in mechanical components. These materials are listed by their popular names and their chemical designation. The material may exist under several trade names. The radiation doses which are labeled "serviceable" are based on evaluations of test data on elastomer-based electrical insulation-jacket combinations (16). The data were employed to define a maximum total integrated dose to which a given material such as wire insulation or a cable jacket might be exposed without a significant change occurring in each of the material properties studied (tensile strength, elongation, rate of oxidation, dielectric loss, electrical stability, dielectric strength). Since elastomers used in mechanical equipment applications are physically enclosed and retained, it is expected that they will be "still serviceable" at even higher radiation doses than those reported in Table 4-4. It should be noted that most of the data shown in Tables 4-4 and 4-5 are threshold values. As a result, they can be used as a screening process. It is not necessary to obtain data on higher doses and exact property changes if the environmental doses are below the threshold value.

TABLE 4-4

NON-METALLIC MATERIAL
ALLOWABLE TEMPERATURE AND RADIATION EXPOSURES

POPULAR NAME	CHEMICAL DESIGNATION	MAXIMUM CONTINUOUS TEMPERATURE °F (°C)	SERVICEABLE RADIATION LEVEL RADS	REFERENCE
Butyl GRI	Isobutylene-Isoprene	185 (85)	$5x10^6$	(16) pp. 7-44
Neoprene GRM	Chloroprene	200 (93)	$5x10^7$	(16) pp. 7-44
Buna	Butadiene	220 (104)	$5x10^6$	(68) p. 12
EPR	Ethylene Propylene	300 (149)	$1x10^8$	(16) pp. 7-44
Fluoro-elastomer (Viton)	Vinylidene fluoride + hexafluoro-propylene	400 (204)	$5x10^7$	(69)
Teflon (TFE)	Fluoroethylene	500 (260)	$1.7x10^4$	(16) pp. 7-42
Nylon	Polyamide	300 (149)	$4.7x10^6$	(16) pp. 7-42
Tefzel	Fluoropolymer	302 (150)	$5x10^7$	(70)

NOTE: Serviceable radiation level is intended to be an order of magnitude screening value to indicate the relative importance of radiation, as compared with other aging factors such as temperature and wear, in the overall evaluation of environmental qualification.

TABLE 4-5

NON-METALLIC MATERIAL
RADIATION DATA

	RADIATION	TRADE NAME
Oils	5×10^7	All Mobil Oils
	1×10^8	High Quality Conventional Mobil Oils
Grease	2×10^7	All Mobil Greases
	3×10^8	Mobilux 2 and 3 Mobilgrease Special Gargoyle 1200 and 1200W Mobilgrease 28 and 29 Mobiltemp 7S Mobilux EP2 Chevron Grease 159
Gaskets & Packing		
Asbestos & Steel	10^9	Flexitallic
Elastomer	See Table 4-4	

EPRI has evaluated radiation effects on oil (11). Several oil companies (12, 17) report data on performance of oils following irradiation. Much of the work was done in the 1950's under programs sponsored by AEC (20, 21) and involved radiation to levels well in excess of post-accident doses.

The following summarizes these findings on oils.

1. For 10^6 Rads or less, no unusual problem from irradiation noted (11).

2. For 10^7 Rads, no unacceptable degradation of lubricants containing the esters, ethers, silicones of polyalkenes and mineral oils, the most frequently seen oils in the marketplace (17).

3. The temperature threshold for accelerated oxidation when most oils are exposed to over 10^9 Rads is $285^{\circ}F$ ($140^{\circ}C$) (17).

4. Mobil Oil reports all of its oil can provide satisfactory service when irradiated to $5x10^7$ Rads (12) and suggests that high quality conventional lubricating oils are suitable for doses up to 10^8 Rads (23).

5. Table 4-6 summarizes the changes on a typical, highly refined mineral oil (22) when irradiated to dose well in excess of 10^7 Rads.

6. Reference (16) indicates that lubricants are satisfactory to 10^7 Rads and do not suffer catastrophic damage even at 10^8 Rads.

TABLE 4-6

RADIATION INDUCED CHANGES
TO A
TYPICAL, HIGHLY REFINED MINERAL OIL
(Reference 17)

Gamma Dose at 24°C, 10^8 Rads in Air	0	0.9	9.0
ASTM Color (Estimate 23)	0.5	1	2
Viscosity at 99°C, cSt	7.4	8.6	41
Viscosity at 38°C, cSt	71	88	1010
Viscosity Index	65	67	80
Pour Point, °C	-34	-34	-21
Flash Point, °C	224	204	202
Vapor Pressure, Pa at 204°C	278	1067	1333

It should be noted that almost all the safety-related mechanical equipment using oil lubricants such as pumps, diesel engines and mechanical drive turbines are all located outside of containment where integrated accident radiation doses are 10^7 R, or less.

With regard to greases, Mobil (12) indicates that greases can be selected which have good thermal stability in addition to high radiation resistance. Mobil conducted radiation tests that were carried out predominantly with gamma radiation to avoid contamination of the grease. Static and dynamic tests were usually performed using standard ASTM tests. It is significant that lubricating greases showed higher radiation resistance during dynamic irradiation tests than during static tests, in particular in the case of soap and complex-soap greases.

Two of the most striking effects occurring to an organic grease as a result of exposure to radiation are gas evolution and changes in viscosity. The viscosity usually increases upon continual exposure to radiation until the grease has polymerized into a solid form. Tests performed by Mobil (12) indicate sufficient resistance to radiation to give satisfactory service with normal and accident exposures:

2×10^7R	All Mobil Lubricating Greases
3×10^8R	Selected conventional lubricating greases of which the radiation resistance has been tested, e.g., Mobilux 2 and 3, Mobilgrease Special, Gargoyle Grease 1200 and 1200W, Mobilgrease 28, Mobiltemp 7S, Mobilgrease 29, Mobilux EP2

The latter greases have radiation resistance greater than required for the accident environment.

Chevron reports similar results for their Nuclear Radiation Resistance (NRR) Grease 159. This grease consists of a synthetic alkyl aromatic oil with a sodium terephthalamate gellant and a selenide oxidation inhibitor. NRR Grease 159 has a maximum operating temperature of 325°F (162°C). Designed for use in high radiation environments, NRR Grease 159 retains its lubricating ability up to 3×10^9R (62).

Table 4-5 lists information on radiation for several other non-metallics used in mechanical components.

__Radiation Dose Rate Effects__

Radiation effects data cited in Tables 4-4, 4-5 and 4-6 were obtained from tests performed employing high dose rates. Recent tests at Sandia National Laboratories indicate that, for some elastomers, a low dose rate causes more damage than an equal gamma radiation at a high dose rate (__71__ and __72__). The presence of oxygen is required to produce the increased detrimental effects at low dose rates, and high total radiation (over 10^7 Rads) appears necessary to cause significant effects.

The low dose rate irradiation effects investigated by Sandia Laboratories should have little, if any, effects on elastomeric materials used in active safety-related mechanical equipment. The gaskets, seals, and O-rings used in this equipment are normally totally enclosed, and have very restricted access to oxygen. In addition, the total low dose irradiation, to which this equipment is exposed, should be below the amounts which cause significant effects.

4.4 ENVIRONMENTAL QUALIFICATION TEST FACILITIES

Test facilities usually have the capability to achieve high temperatures and up to 100% relative humidity as an environmental load on the test specimen. Some facilities have the ability to control temperature and humidity with time, including the ability to achieve fast temperature and humidity ramps to simulate a LOCA. The facility may or may not include the capability to concurrently expose the specimen to X-ray or gamma radiation, as the radiation exposure is often applied to the specimen by an independent facility prior to the thermal aging process.

Typical ranges of test chamber capabilities in test facilities are:

Test chamber size (LxWxH): 1 ft. x 1 ft. x 1 ft. to 8 ft. x 8 ft. x 10 ft.

Maximum temperature and humidity (^{o}F/%RH): 300^{o}F/100% to 650^{o}F/100%

Maximum pressure (psig): 0 to 150 psig

Radiation: 0 to 1 MRad/hr.

4.5 ENVIRONMENTAL QUALIFICATION GUIDE

The materials listed in Tables 4-4 and 4-5 all have serviceable radiation levels of 5×10^6R or greater. This suggests that a specific review for non-metallic material qualification is appropriate only for mechanical components located in the high potential radiation zones. A threshold for all organic materials (except Teflon which is reported as 1.5×10^4R) has been reported as 10^5R (11), while the discussion under Section 4.3 shows that an oil and grease threshold should be 10^7R. These facts suggest the following guidelines.

1. Oils and greases: Conduct a design review only if the integrated post accident radiation dose exceeds 10^7R.

2. Organic materials: Conduct an analysis or test only if the radiation dose is $>10^4$ or if the equipment contains no teflon and radiation doses are greater than 10^5R.

3. Graphite and asbestos materials: Waive further design review if the only non-metallic materials used in the equipment are graphite or asbestos based.

Figure 4-1 is a flow diagram illustrating guidance for an environmental qualification methodology which could be used for active safety-related mechanical equipment in an operating nuclear power plant. The methodology emphasizes the role of preventive maintenance and surveillance and the evaluation of the capability of the mechanical equipment to withstand the effects of environmental conditions. Note that the methodology does not begin with a comparison of design specifications with operational and service conditions. The environmental, seismic and dynamic loads have already been identified in accordance with the guidance in Section 3. The use of design specifications could be helpful in identifying the active safety-related mechanical equipment while information from other documents such as drawings, bills of materials and instruction manuals is more useful in the qualification process itself.

The methodology illustrated consists of a series of basic questions whose answers may be used to determine the need and extent of further qualification. These are discussed below.

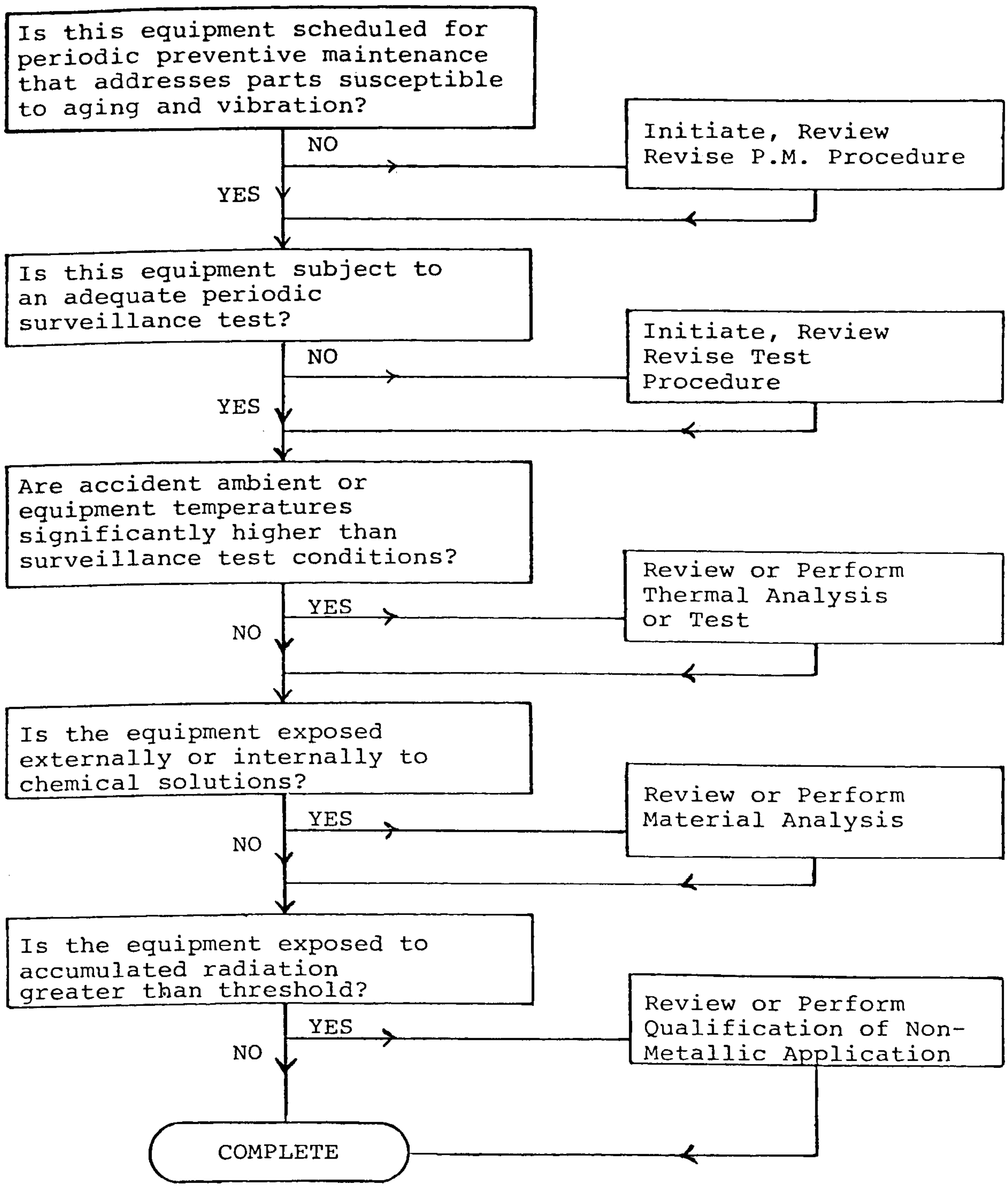


FIGURE 4-1: MECHANICAL EQUIPMENT ENVIRONMENTAL QUALIFICATION PROCESS

1. Is this equipment scheduled for a periodic preventive maintenance program that addresses parts that are susceptible to aging?

   Comment: The preventive maintenance program should address all parts subject to aging. This may range from an instruction to replace parts with equipment or better parts at prescribed intervals, or to replace or refurbish when a specified vibration or performance degradation is observed during surveillance testing. As a rule, the preventive maintenance program should specify the replacement or disposition of non-metallic materials at specified intervals to provide margin for the post accident environment.

   There are no specific rules governing replacement of ageable parts. One article (14), discussing selection of elastomeric seals, suggests a five-year replacement interval may be conservative. One plant performs maintenance on all motor operated valves inside of containment every refueling outage and every three years for valves outside of containment (60).

2. Is this equipment subject to an adequate periodic surveillance test?

   Comment: The surveillance test should duplicate to the maximum practical extent the performance conditions that will exist when required for the safety-related function. These conditions include start time, open or close time, head, flow and power output. The extent that safety-related conditions cannot be produced during surveillance testing will indicate the need for special analysis or testing. In many cases safety-related conditions can be produced in surveillance testing. For example, most pumps, drive turbines and diesel engines can, by virtue of the system design, be rapidly started, loaded and operated at design flow, head, pressure, speed and load during surveillance. Other equipment, such as isolation valves, may have to be surveillance tested under conditions not typical of the safety-related accident condition and may require special evaluation of the effects of the accident conditions. These conditions that cannot be produced in surveillance testing are usually infrequent accident related dynamic loads that are discussed in Chapter 5.

3. Are accident ambient or equipment temperatures significantly higher than surveillance test conditions?

   Comment: This question relates to the ability to simulate operating temperatures that can affect operation. The most important consideration is internal temperatures that can cause adverse thermal expansion, leading to rubbing and wear, or accelerated deterioration of non-metallic parts. As a rule of thumb, surveillance test conditions that are within $100^{o}F$ ($55^{o}C$) of safety-related operation should be considered adequate without need for special analysis or test. The answer to the above question is "no" for most safety-related pumps, diesel generators and drive turbines which are surveillance tested under load and temperatures comparable to accident conditions.

4. Is the equipment exposed externally or internally to chemical solutions?

   Comment: Internal exposure should have been considered in the basic equipment design, including the selection of materials appropriate to withstand chemical attack. External exposure during accidents should be considered if materials in the mechanical equipment could be damaged by chemical attack.

5. Is the equipment exposed to less than threshold doses?

   Comment: Nearly all materials have a threshold for material change that is greater than 10^5 Rads. The notable exception is Teflon which has a threshold of 10^4 Rads.

Figure 4-2 is a flow diagram illustrating guidance for qualification of non-metallic parts used in applications where the accumulated exposure exceeds 10^5 Rads. The elements of this process are a set of data, questions and tasks:

Radiation data: This is the data describing the radiation exposure on the equipment. Several categories are included to account for the fact that periodic replacement of non-metallic parts reduces radiation and that parts internal to some equipment are not exposed to Beta radiation because of shielding by metal parts.

Comments on Figure 4-2

1. Does this equipment use oil or grease that is exposed to greater than 10^7 Rads?

   Comment: The evaluation in Section 4.6 shows radiation damage to oil and greases due to 10^7 Rads or less to be minor. A design review test of the actual oil or grease used if radiation exposure is 10^7 Rads or less is not warranted.

2. Does this equipment use non-metallic bearings?

   Comment: Metallic or graphite bearings are most commonly used. These materials are not detrimentally affected by the most severe radiation encountered in nuclear safety related equipment.

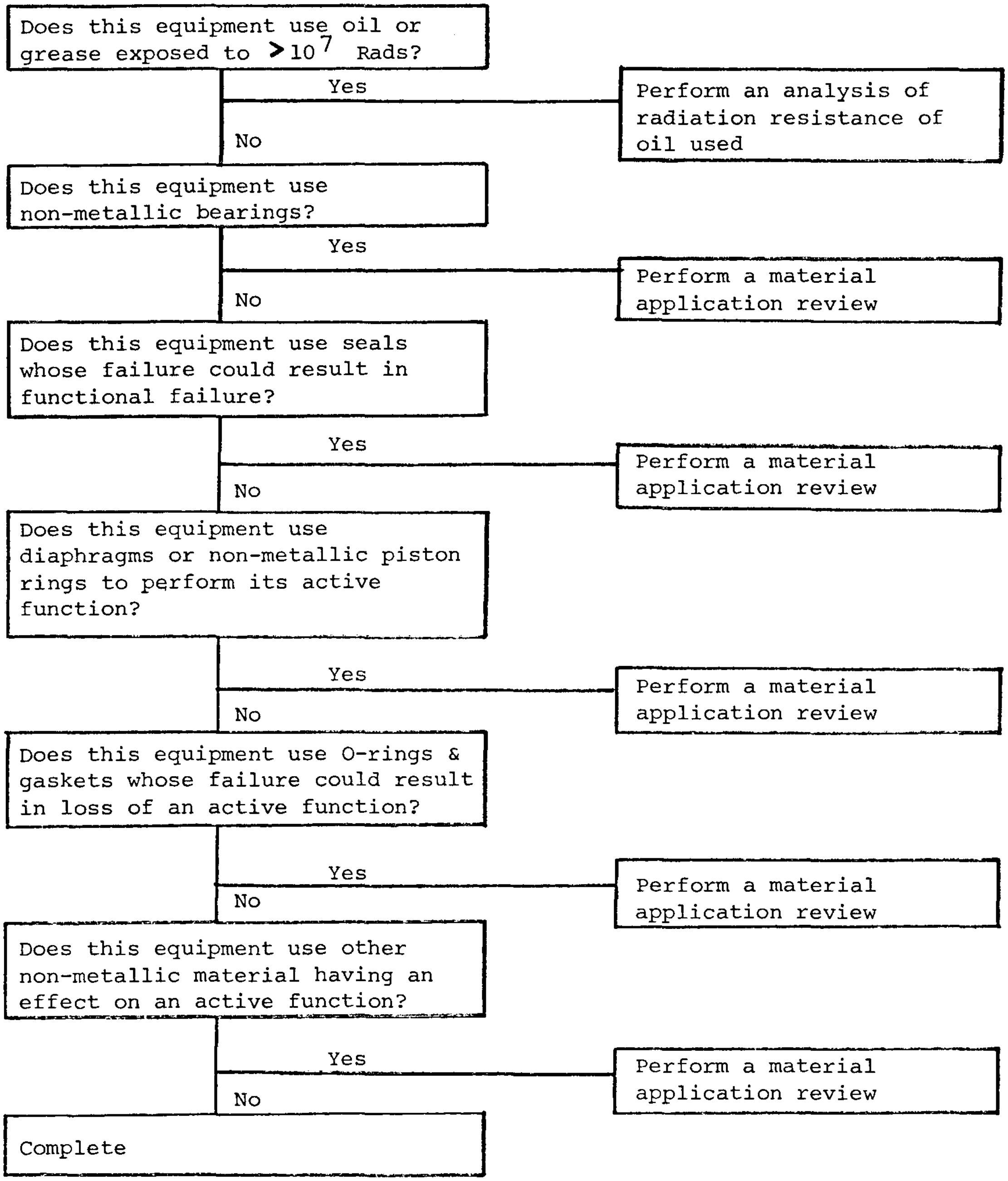


FIGURE 4-2: RADIATION QUALIFICATION PROCESS (FOR RADIATION $>10^5$ RADS)

3. Does this equipment use seals whose failure could result in functional failure?

<u>Comment</u>: The question is directed to radiation and thermal degradation of seal material. The answer is "yes" if

(1) Seal failure causes loss of the equipment lubrication, or

(2) The seal is the pump or valve shaft seal and seal leakage could mix with and deteriorate the lubrication or otherwise cause loss of function of the equipment or other safety-related equipment in the vicinity.

(3) The seal is a valve seat whose failure could defeat the primary purpose of providing flow shutoff after an accident.

4. Perform a material application review.

<u>Comment</u>: A material application review, wherein the capability of the material to maintain acceptable physical characteristics after irradiation is the primary objective of such a review. This review usually is based on the existence of acceptable characteristics prior to the accident as the result of a preventive maintenance program that provides for replacement and refurbishment of parts susceptible to aging from other environmental stresses.

Section 5
DYNAMIC LOADS QUALIFICATION

Dynamic loads that are considered in this report are transient loads on mechanical equipment that can occur during plant design basis events but that are not seismic related, are not due to normal operation and testing, nor are they capable of being duplicated in surveillance testing. They are categorized in this manner so that they can be distinguished from normal vibratory loads occurring during normal plant operation and testing and the vibratory loads from seismic events.

It is important to note that design basis events such as LOCA do not place unusual loads on most safety related active mechanical equipment which is located outside of primary containment. This equipment is separated from the LOCA dynamic loads inside containment by piping and valving and is usually located in a structure which is not subject to these loads. For example, an ECCS pump starting and operating in response to a LOCA will have the same operating parameters such as fast start, flow and pressure, as will exist during surveillance testing.

Identification of such loads requires answers to questions such as:

- Are there unusually high flows or flow transients through the equipment during design basis events that may affect the function of the equipment during or after the event?
- Are there unusually high vibratory loads transmitted to the equipment through building structures and connecting systems during the event?

Two types of significant dynamic loads have been identified during the course of this study: pipe rupture blowdown loads and BWR hydrodynamic loads.

Pipe Rupture Blowdown Loads

A primary coolant system pipe rupture will cause a high velocity, two-phase flow of primary coolant through the piping and associated components. This high flow condition will persist until either system depressurization occurs or an isolation valve in the line closes and shuts off the blowdown flow. The isolation valve is usually required to achieve shutoff in a short period of time before significant depressurization can occur. The isolation valve is therefore subjected to a transient, two-phase, high differential pressure load during the closing period. This is a load condition that cannot be duplicated in surveillance testing. This load is unique to isolation valves and is discussed further in Section 7.6, Main Steam Isolation Valves.

Hydrodynamic Loads

These are vibratory loads that can be generated in the suppression pool of a BWR containment by loss-of-coolant accidents (LOCA) and safety relief valve (SRV) discharge. These loads are transmitted to mechanical equipment that have a basemat or foundation that is common with the suppression pool. The hydrodynamic loads are characterized as having a relatively high energy at frequencies that are high compared to seismic loads. A typical response spectra for hydrodynamic loads is shown in Figure 5-1.

For equipment with deformation or strain energy dependent failure modes, there is evidence that the presence of higher frequency hydrodynamic loads does not affect the equipment operability where seismic qualification has been performed for the plant's seismic response spectrum (61). This is based on the principle that the total kinetic energy of a linear system subjected to a vibratory load can be determined by the superposition of its modal strain energies and that the work done by higher modes decays exponentially. The effect of higher modes of vibration on equipment response is insignificant on the basis that missing strain energy not accounted for by limiting dynamic analysis to the seismic frequency range of interest is insignificant. Other equipment using relays and latches may have acceleration dependent failure modes.

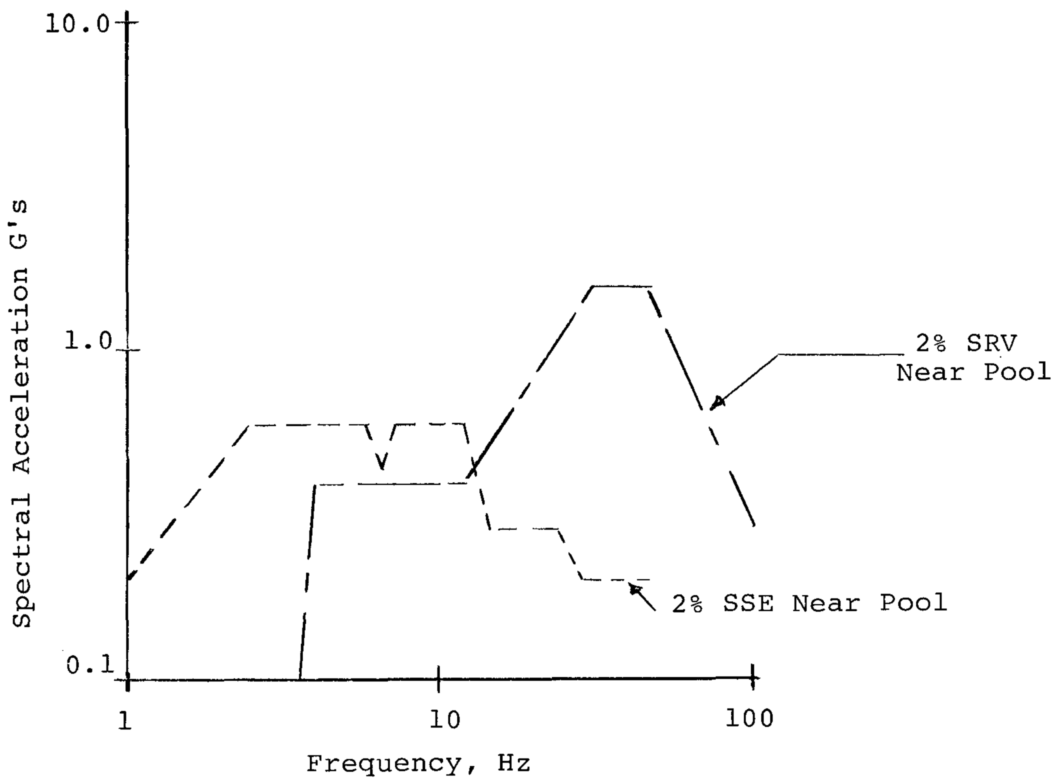


FIGURE 5-1: TYPICAL SSE AND SRV SPECTRA

One approach for addressing such modes is the utilization of military test data, as demonstrated in an EPRI research project (125). Military specifications require that the equipment be tested (or demonstrated by other procedures) to comply with shock and vibration environments. Because of the similarity between the load characteristics of the military hardening programs and the nuclear plant high frequency loads, this EPRI study adapted the data and procedure developed by the United States Army Engineers at the Huntsville Division (USAEHD). The study applied the adapted procedure and data to selected nuclear plant equipment to demonstrate both the applicability and the validity of the methodology. Summaries of the available (non-classified) military test data are presented in References 126 and 127.

The hydrodynamic loads are in the nature of high frequency vibratory loads with small displacements. Equipment components that are adversely susceptible to small displacement vibration should be reviewed for potential adverse effects of hydrodynamic loads. These components are also susceptible to ambient vibration and should be a part of the preventive maintenance program. They include:

1. Components such as adjustable linkages that are secured by set screws.

2. Small piping and hoses that are relatively unsupported and configured such that they fail by fatigue.

In summary, qualification for hydrodynamic loads should consist of:

1. Identifying active, safety-related components that are subjected to hydrodynamic loads. In BWRs, these are located either in the containment or out-of-containment where high frequency excitations from the suppression pool or SRVs are transmitted through supports and foundations to the mounted equipment.

2. Reviewing the equipment and components in the areas where the high-frequency loads are significant for susceptibility to these loads. This should include a review of equipment linkages, small piping, hoses and similar components susceptible to damage or loosening under vibration.

3. The adequacy of the suspect items can be demonstrated by one or more of the following approaches:

   (a) The functionability requirements of specific suspect items under high-frequency loads can be used to show that some items need not operate during or after events causing hydrodynamic loads.

   (b) Survey the industry and use experience data to show qualification. Some NTOLs have qualified the components to high-frequency loads.

   (c) Use the military data and procedures to screen out items with large hardness margins.

   (d) Perform a dynamic test (or combine with seismic test) to qualify items. The TRS of such a test should include the hydrodynamic excitation in the high-frequency range (>35 to 100Hz).

Section 6
SEISMIC QUALIFICATION OF MECHANICAL EQUIPMENT

6.1 OVERVIEW

The purpose of seismic qualification of active, safety-related mechanical equipment is to demonstrate its ability to perform its function as required during and/or after the time it is subjected to the forces resulting from an earthquake. For design purposes, two magnitudes of earthquakes are defined. These are an operating basis earthquake (OBE) and a safe shutdown earthquake (SSE). As the names imply, an OBE could reasonably be expected during the life of a plant. Continued plant operation during and following an OBE is anticipated. An SSE, on the other hand, is the maximum potential earthquake for a specific plant site. The maximum ground acceleration of the SSE is often taken as twice that of the OBE. Safety-related structures, systems and components must be capable of shutting down the reactor and maintaining it in safe shutdown condition following an SSE.

Qualification criteria have undergone considerable evolution during the last two decades. Plants designed in the 1960s did not have specific seismic qualification criteria. Initial criteria were published in the early 1970s and subsequently revised a few years later. In the intervening period a large amount of feedback was received and reviewed. The present criteria reflect technical refinement and recognition of testing and analysis limitations derived from these reviews.

Current general requirements for seismic qualification are contained in General Design Criteria 1, 2 and 4 of Appendix A to Code of Federal Regulations (CFR) Part 50 (2). Additional general requirements are contained in Sections III and XI and Appendix B to 10CFR Part 50 and Criterion VI of Appendix A to 10CFR Part 100 (1).

TABLE 6-1

NATIONAL STANDARDS AND NRC REGULATORY GUIDES RELATED TO MECHANICAL EQUIPMENT SEISMIC QUALIFICATION

NATIONAL STANDARDS

ANSI/ASME B16.41-1983	Functional Qualification Requirements for Power Operated Active Valve Assmeblies for Nuclear Power Plants
ANSI/ASME N278.1-1975	Self-Operated and Power Operated Safety-Related Valves Functional Specification Standard
ANSI/ASME OM-1-1981	Requirements for Inservice Performance Testing of Nuclear Power Plant Pressure Relief Devices
ANSI/ASME OM-2-1982	Requirements for Performance Testing of Nuclear Power Plant Closed Cooling Water Systems
ANSI/ASME QNP-1-Draft	Standard for Qualification of Pump Assemblies for Safety Systems Service - General Requirements
ANSI/ASME QNP-2-Draft	Standard for Qualification of Pumps in Pump Assemblies for Safety Systems Service
ANSI/ASME QNP-3-Draft	Standard for Qualification of Shaft Seal Assemblies in Pump Assemblies for Safety Systems Service
ANSI/ASME QNP-4-Draft	Standard for Qualification of Motor Drivers in Pump Assemblies for Safety Systems Service
ANSI/ASME QNP-5-Draft	Standard for Qualification of Turbine Drivers in Pump Assemblies for Safety Systems Service
IEEE Std. 627-1980	Standard for Design Qualification for Safety Systems Equipment Used in Nuclear Power Generating Stations

NRC REGULATORY GUIDES

RG 1.29	Seismic Design Classification
RG 1.48	Design Limits and Loading Combinations for Seismic Category I Fluid System Components
RG 1.60	Design Response Spectra for Seismic Design of Nuclear Power Plants, REV.1, Dec. 1973
RG 1.92	Combining Modal Responses and Spatial Components in Seismic Response Analysis, REV.1, February 1976
RG 1.148	Functional Specifications for Active Valve Assemblies in Systems Important to Safety in Nuclear Power Plants, REV. 0, March 1981

Currently there are few specific guidelines for seismic qualification of mechanical equipment. Much of the guidance is derived from industry standards developed for seismic qualification of electrical equipment. However, recent standards activity on the part of the American National Standards Institute (ANSI) and the American Society of Mechanical Engineers (ASME) has taken a significant step in recognizing the fundamental differences between mechanical and electrical equipment. This is important in determining the methods which are appropriate to the seismic qualification of mechanical equipment.

The basic methods of seismic qualification are:

1. Qualification by experience
2. Qualification by analysis
3. Qualification by test
4. Qualification by a combination of test, analysis and/or experience

Seismic qualification involves the selection of one or a combination of these methods.

Seismic experience can play a key qualification role for plants with a moderate intensity seismic criteria. In particular, recent California earthquakes have been studied under the sponsorship of the Seismic Qualification Utility Group (31). A range of valve and pump types and sizes were identified in this study as being qualified for a bounding set of ground response spectra developed from the study. It may be possible to extend application of seismic experience with additional data.

In general, qualification by analysis can be used if the failure modes can be expressed and evaluated in structural or quasi-structural terms such as a bearing load or shaft deflection. An example is a pump rotating element and its bearings. Testing may be required when potential failure modes are not known or an analysis may not readily predict a potential failure point. An example may be the effect of seismic accelerations on a fly-ball overspeed trip mechanism. Finally, qualification by experience may be possible when similar equipment has successfully survived in a related nuclear or non-nuclear application.

Qualification may employ a combination of testing, analysis and/or experience. For example, the fly-ball governor, previously mentioned, may be shake-table tested while being operated at the required speed, while the remainder of the equipment might be qualified by analysis. On the other hand, some simple analyses might be used in combination with experience to demonstrate that the equipments have similar dynamic characteristics. Also, in-situ testing, a special form of testing components in the factory or in the installed condition at the power plant, can be used to supplement qualification by analysis, laboratory testing and/or experience.

The input excitation used for qualification of equipment is a representation of the seismic loading which an active component must be designed to withstand without loss of operability. The input excitation is usually presented in one of the following forms:

- Peak acceleration
- Required response spectrum (RRS)
- Time history

The different forms of input excitation are shown in Figure 6-1.

The peak acceleration, shown in Figure 6-1a, is simply the peak acceleration to which the equipment can be subjected, considering amplification of the seismic ground motion through the building structure. It is most often taken as the peak of the time history input or the ZPA of the required response spectrum discussed below.

The acceleration imposed on the equipment by an earthquake is usually expressed in the form of a required response spectrum (RRS), which is the maximum response of a hypothetical series of single degree of freedom oscillators to the seismic excitation. The seismic RRS typically has the form shown in Figure 6-1b. At very low frequency the input response spectrum contains little energy; over a middle range of frequency the maximum amplification occurs, and above a certain frequency there is again little input energy. Equipment response at very high frequencies is called the zero period acceleration (ZPA) of the RRS. The response of the middle range of

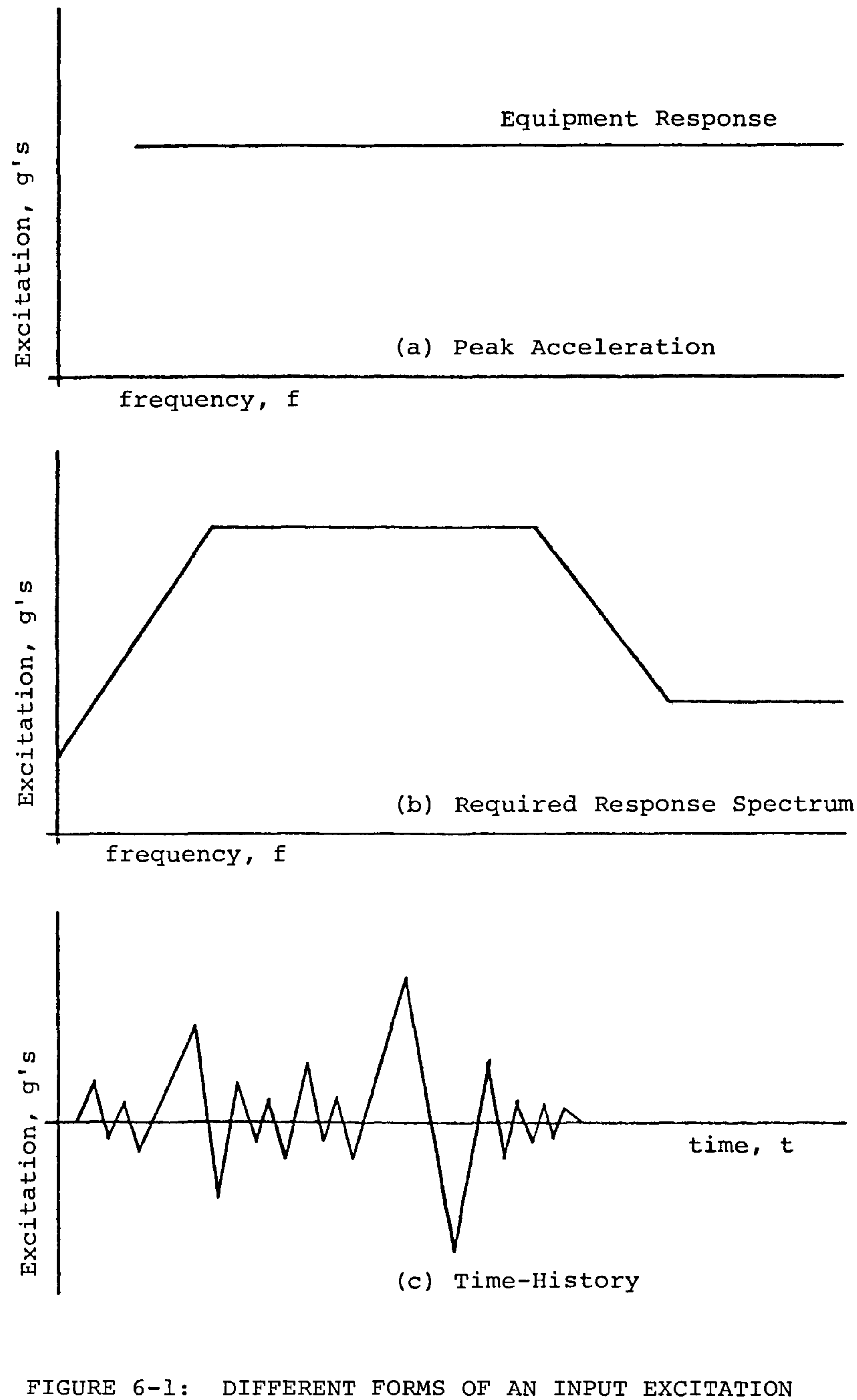


FIGURE 6-1: DIFFERENT FORMS OF AN INPUT EXCITATION USED FOR EQUIPMENT QUALIFICATION

frequency depends on the damping used to calculate the RRS. (It should be pointed out that the damping used to develop the RRS is the damping of the hypothetical series of single degree of freedom oscillators and is not necessarily the damping associated with the equipment being qualified.)

Time-history functions may be developed to characterize the complete time dependent dynamic input motion to a component resulting from siesmic ground motion. The time history function represents the full frequency content and phase relationships of the input excitation for the duration of the earthquake event. A time-history excitation is shown in Figure 6-1c. Like the RRS, the time-history may be in the form of acceleration, velocity or displacement. The most commonly used form is acceleration versus time. The excitation used to generate an equipment response may be either a time-history or a RRS appropriate for the equipment location. The approach usually used to generate a time-history or RRS for equipment is to use a dynamic model of the reactor building to calculate time history data for the various floors and elevations. If the equipment is not mounted on the floor but is, for example, pipe mounted, a dynamic model of the mounting structure must also be analyzed. The output from these models becomes the excitation to the single-degree-of-freedom oscillator used to develop the RRS at the equipment base.

In the case of pipe mounted equipment, the results of a piping system dynamic analysis are often not available. In these cases it may be necessary to do an analysis or to conservatively estimate the maximum acceleration the equipment is likely to see. Appendix C provides a method for estimating the maximum seismic loads for pipe mounted equipment.

The most important dynamic characteristic of a mechanical component is its set of natural frequencies or modal frequencies and mode shapes. Equipment dynamic characteristics determine if the component should be evaluated as a rigid or as a flexible structure. Mechanical components are considered to be rigid if their lowest resonance frequency is greater than the lowest excitation frequency for which there is significant dynamic energy. See Figure 6-2.

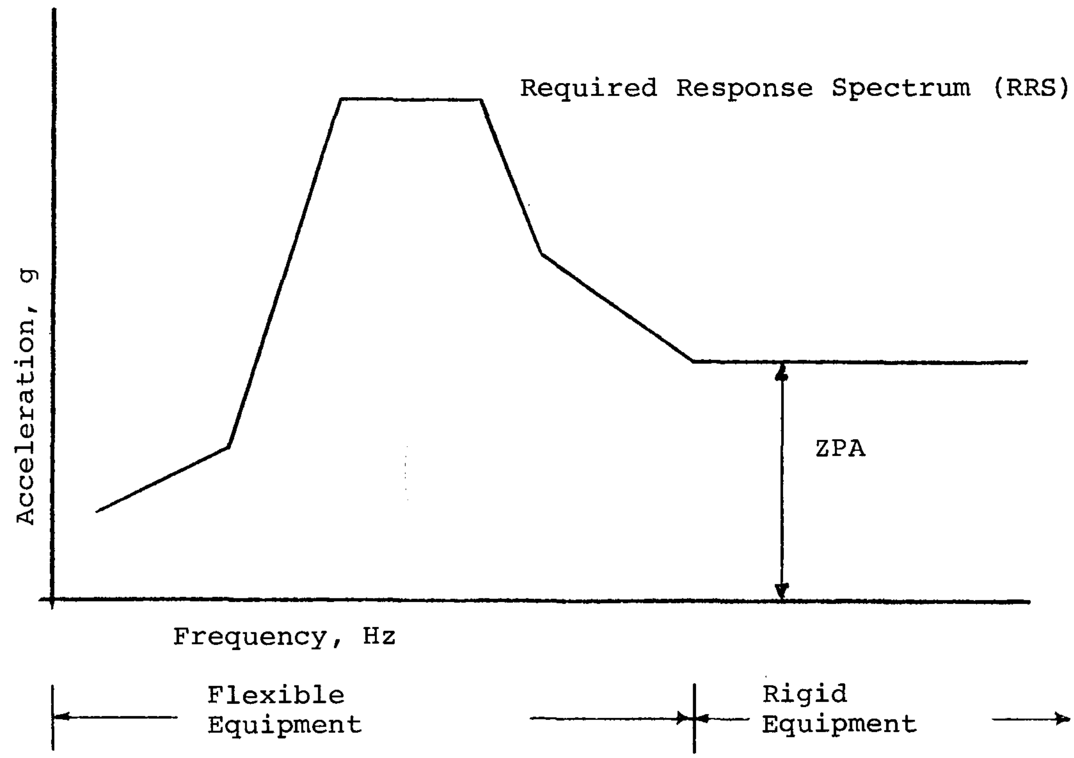


FIGURE 6-2: DEFINITION OF FLEXIBLE AND RIGID EQUIPMENT

Rigid equipment is usually qualified by static analysis methods. If the equipment is flexible, either static or dynamic analysis or testing is applicable. In general, static methods are the simplest and least expensive method but are the most conservative while testing is the most expensive but usually has the least conservatism.

6.2 QUALIFICATION BY EXPERIENCE

One method of qualifying active mechanical equipment is to review and evaluate performance of equipment that has been subjected to vibratory motions of an earthquake. A major study and evaluation of conventional power plant equipment has recently been undertaken by the Seismic Qualification Utility Group (31, 118-123), a group of U.S. utilities formed to develop an alternative approach to seismic equipment qualification based on seismic experience. Among the classes of mechanical equipment reviewed by SQUG were the following:

- Motor-operated valves
- Air-operated valves
- Horizontal pumps
- Vertical pumps

In addition, a Senior Seismic Review and Advisory Panel (SSRAP) was formed to review and comment on the SQUG study (124). After a review of the full range of the available experience data base compiled by SQUG, the SSRAP conclusions for these classes are:

1. Equipment installed in nuclear power plants is generally similar to and at least as rugged as that installed in conventional power plants.

2. This equipment, when properly anchored, and with some reservations as discussed in Reference (124), has inherent seismic ruggedness and a demonstrated capability to withstand significant seismic motion without structural damage.

3. For this equipment, functionality after the strong shaking has ended has also been demonstrated.

Typical SSRAP reservations and limitations on the equipment covered by the SQUG data base were:

1. The motor-operated valve housing and yoke construction is not of cast iron.

2. Limitations on distance from pipe centerline to the top of valve operators.

3. Horizontal pumps and their driver are rigidly connected through their base to prevent damaging relative motion.

Mechanical equipment subject to shake table testing is also a source of experience data, but is largely uncompiled at this time. A project currently sponsored by EPRI is directed towards compiling this data. Appendix B contains a brief survey of other major earthquakes that could be a source of qualification data.

6.3 QUALIFICATION BY ANALYSIS

Many of the potential modes of failure for active mechanical equipment involve overstress, loss of alignment or unacceptable distortion in moving parts. These strain dependent failure modes can often be evaluated by analysis in which the calculated stresses, loads and deformations are compared with explicitly stated limits or acceptance criteria. For example, allowable stress may be limited to less than yield stress to prevent permanent deformation, load on a bearing may be limited to its momentary load rating and impeller/shaft deflections may be limited to the clearance provided with allowance for momentary rub of the wearing rings.

Qualification by analysis involves the choice of static or dynamic analysis. A static analysis may employ either the static method or static equivalent method. The static method is the simplest and applies the rigid equipment. Static analysis is based on ZPA loads being applied to the equipment centers of mass. The static equivalent method is based on the equipment dynamic response being unknown and the equipment being subject to peak acceleration loads multiplied by a coefficient of 1.5 to account for unknown closely coupled dynamic modes. Dynamic analysis involves modelling the equipment by masses and structural elements and determining the response of the model to a spectrum of dynamic seismic loads. Dynamic analysis almost always involves the use of computer programs designed to solve the time dependent equations of motion. These methods are described in more detail in the following paragraphs.

Static Method

In many cases, a simple static analysis can be used to demonstrate that the equipment will perform its safety related function(s) during and following an SSE. The cases when this is acceptable are where the equipment can be shown to be rigid (i.e., it has no resonant frequencies below 33 Hz). In these cases, the force on each component of the equipment is found by multiplying its weight by the ZPA of the applicable response spectrum. Depending on the simplicity/complexity of the model, the ZPA can be multiplied by the entire mass of the component concentrated at its center of gravity or by each mass of a distributed mass system which represents the various portions of the equipment. The applied force is generally multiplied by $\sqrt{2}$ to account for seismic accelerations in two orthogonal directions.

Often all that is needed to demonstrate that equipment has no resonant frequencies below 33 Hz is a simple one or two degree of freedom model of the equipment. Such models provide an assessment of the first one or two modes of vibration and if these are above 33 Hz, higher modes will have even higher resonant frequencies. In-situ test results are also very helpful in determining the lowest resonant frequency of equipment and for classifying equipment as rigid or flexible.

Static Coefficient Method

The static coefficient method, or the equivalent static method requires no determination of the natural frequencies of the equipment. The static coefficient method is an acceptable but conservative method of qualifying equipment for seismic loads. With this method the response of the equipment is assumed to be at the peak of the required response spectrum at a conservative and justifiable value of damping. This peak response is then multiplied by the static coefficient of 1.5 to account for the effects of both multifrequency excitation and multimode characteristics of equipment. The inertia forces on the equipment are obtained by multiplying weights by the value of 1.5 times the peak of the required response spectrum. Depending on the complexity of the equipment the entire mass of the component can be concentrated at its center of gravity, or a distributed mass system with adequate level of mass concentration may be used. Alternately, for equipment whose lowest resonant frequency is known and is higher than the amplified region of the RRS but less than the frequency at which the ZPA begins, the spectral acceleration value can be read from the RRS at the appropriate frequency. The factor of 1.5 must still be applied. As with static methods the applied force is generally multiplied by $\sqrt{2}$ to account for loads in two directions.

The use of the static coefficient method with a multiplier of 1.5 is allowed by IEEE Std. 344-1975 (30) and is endorsed by NRC (74) with the provision that the use of 1.5 as the static coefficient (multiplier) is acceptable for verifying structural integrity of frame-type structure that can be represented by a simple model. For equipment having more complicated configurations justification should be provided for use of a static coefficient analysis.

The static coefficient method is often used for seismic qualification of pumps and valves with extended operators if the built-in conservatism of the method does not penalize the design.

Dynamic Analysis Methods

Dynamic analysis methods are employed where static analysis methods cannot be justified or if the results of the static analysis calculations exceed the acceptance criteria because of overconservative analysis assumptions. Dynamic analysis methods usually employ finite element techniques to model the physical characteristics of the equipment in sufficient detail to adequately predict the component dynamic response. The primary physical characteristics used are the distributed weights and stiffness of the equipment members. These properties of the equipment are obtained by dividing the equipment into a series of lumped masses and connecting weightless beams (stick model) the latter representing the bending and torsional stiffness between lumped masses. Computer programs, such as ANSYS, EASE, NASTRAN, SAP, STARDYN and STRUDL, are available in the public domain for use in dynamic analysis. The input excitation in the form of an RRS or time history is applied to the model. (The response spectrum method of analysis is the most commonly used.) The dynamic analysis results in the form of stresses and deflections are compared with the acceptance criteria. Finite element analysis modeling involves many detailed techniques beyond the subject matter of this report. The reader is directed to References (55-57) for additional information.

As an example, the mathematical model for a pump and motor assembly may contain relevant portions of the motor, motor bearings, rotor shaft, pump and motor mountings, pump bearings, impellers, impeller housing, pump casing and exterior shell casing. The effect of the pumped fluid mass may also be included in the model. The actual mass of the fluid contained in the pump is used to determine the effect on the shell and foundation. However, for the internal components, such as the rotor, the virtual mass of the fluid must be determined and included in the analysis. This virtual mass normally does not have any effect on the pump thrust, which consists of the unbalanced axial thrust and for a vertical pump, the weight of the rotor. However, for the rotor elements the damping effect of the pumped fluid needs to be included in the mathematical model. Reference (27) contains a more complete treatment of this subject.

Model complexity and verification are important issues related to equipment qualification by analysis. A complex structure is often synthesized by an assemblage of different forms of finite elements (e.g., beams, plates). The degree of confidence on the predicted response of such a model will depend on several parameters, among which the uncertainty associated with mass and stiffness magnitude and its distribution, and possible errors due to selection of inappropriate finite elements, damping valves, discretization and geometric nonlinearity can be mentioned. In-situ testing offers one means of demonstrating the adequacy of models.

Response Spectrum Analysis Method

Response spectrum analysis method is commonly used for qualification to obtain the maximum response of the equipment. Response spectrum analysis method is a very cost effective tool since it only provides the maximum value of a response quantity within the frequency range of interest (often the frequency range of input excitation).

To perform a response spectrum analysis, first the physical system is mathematically idealized in terms of sets of modes as an assemblage of different elements. Beam elements with six degrees of static freedom and the degrees of dynamic freedom are the most popular elements used for the majority of equipment. Next, the undamped natural frequencies and mode shapes are determined. By performing a suitable transformation, using the undamped modal properties obtained above, a decoupled system of equations are obtained which describe the systems maximum response in the generalized coordinate system. Each of these equations, representing one degree of freedom of the system, provides the maximum response of that mode. The individual model responses are then summed to obtain the total response of the system. Often an SRSS method and sometimes an absolute sum method is used to calculate the total response. Appendix E provides a more detailed treatment of this widely used method for equipment qualification.

Time History Analysis Method

The time history analysis method is also used for qualification of equipment. This method is not as cost effective as the response spectrum method of analysis because of additional computer capability and running time required. However, it does provide a complete response of the equipment in the time domain, while the response spectrum method of analysis only provides the maximum value of the response.

The time history analysis method also requires that the equipment be mathematically idealized in terms of sets of nodes at which masses are concentrated. These nodes are connected by flexible elements allowing all six degrees of freedom to be present. The equation of motion is developed for all degrees of freedom. These equations include inertia, damping and stiffness terms. The input excitation is also defined in the time domain. This time history of excitation could be an artificial time history synthesized from an RRS or directly obtained from a time history analysis of the supporting structure.

Once the equations of motion are obtained and the input excitation is specified, the response of the equipment can be obtained using one of the two commonly used procedures. One is the modal superposition method and the other is the direct integration method.

Modal superposition: This method requires the use of normal coordinates to transform the coupled set of equations of motion to an uncoupled set. The original set of equations of motion are coupled by the off-diagonal terms in the mass and stiffness matrices. The uncoupling of these equations produces a series of independent, single degree of freedom equations corresponding to each mode of the equipment. The modal response is then obtained by solving separately for each significant mode at each time step of input excitation. These modal responses are then transformed back to the physical coordinate system of the equipment to produce the response time history corresponding to each degree of freedom. The equipment response is then obtained as an algebraic summation of the contributions of the response of each mode at each time step of input excitation.

Direct integration: In the direct integration method, the coupled equations of motion are solved simultaneously for each time step of the input excitation by using numerical techniques. This generally involves integrating the coupled equations of motion using a numerical step by step procedure to obtain the time history of equipment response directly. Several numerical techniques are available to produce unconditionally stable solutions. See Reference (28) for a more detailed treatment of this subject.

6.4 QUALIFICATION BY TESTING

Testing involves subjecting the equipment to vibratory motion or loading which conservatively simulates excitation during an SSE. Testing may be for the purpose of demonstrating the equipment adequacy for a specific application, a more generic range of applications (both of these are called proof tests) or determining the maximum equipment capabilities (fragility tests). In general, qualification testing of mechanical equipment is only undertaken if it is concluded that the potential failure modes may not be related to structural response, or if the equipment cannot be realistically or economically modeled. Even in this case a combination of testing and analysis may be practical. Guidelines for testing methods are provided in IEEE-344 (30). Although this standard is specifically written for Class IE electrical equipment, portions of it can often be used to guide testing of mechanical equipment.

The categories of testing that may be applied to mechanical equipment are:

- Static testing: In this type of test, a static load equal to or greater than that derived from the peak acceleration estimated to apply to the equipment is applied to the center of gravity of the equipment while the equipment is operated in its safety related mode.
- Dynamic testing: In this type of testing the equipment is vibrated on a shake table while it is operated or parameters essential to operation are observed. One advantage of this type of testing is

that the equipment is vibrated at energy levels and frequencies comparable to its expected seismic excitation.

Static Testing

Static testing may be used to qualify equipment where

(1) The maximum force on, or deflection of a component of, the equipment can be determined from the required seismic excitation such as the RRS,

(2) the force or deflection is the key parameter affecting operability, and

(3) the operability of the equipment can be demonstrated while the force or deflection is applied.

The most common application of static testing is where the force or deflection is applied to an extended mass and the physical effect is distortion of a component affecting operability. An example of this is to apply a horizontal load to a valve operator to determine the capability of the valve to open or close while the operator extension, valve yoke and valve stem are distorted by the load on the operator. The load on the operator would be the acceleration times the operator mass where the acceleration is determined by

(a) The RRS ZPA acceleration if the assembly is rigid, or

(b) The peak acceleration of the RRS times 1.5 if the assembly is flexible or its natural frequency is unknown.

As is the case for static analysis, the applied force is generally multiplied by $\sqrt{2}$ to account for seismic accelerations in two orthogonal directions.

Dynamic (Shake Table) Testing

Shake table testing is conducted by mounting the equipment to be tested (specimen) on a shake table that can vibrate in one, two or three orthogonal directions. The mounting of the equipment is in a manner that simulates the intended service mounting to the extent practical. Services such as power and fluid flow must be provided if such services are essential to demonstrate operability. For example, it is necessary to provide a driver and closed loop for recirculation of pumped fluid to adequately shake table test a pump.

The types of motion used to simulate the seismic environment fall into two broad categories, single frequency and multiple frequency. Multifrequency is preferred because seismic motions generally include more than one frequency and in that sense, multifrequency testing is believed to be more realistic. On the other hand, when the seismic ground motion has been filtered due to one predominant structural mode, it may be adequate to qualify equipment with a single frequency test. For example, the excitation of a pipe mounted valve is often characterized by one predominate frequency, that of the piping system influenced by the large, concentrated mass of the valve. Also, single frequency testing may be justified if equipment has no significant resonances, has only one resonance, or has widely spread resonances that do not interact to produce a likely failure mode.

In using either multiple or single frequency vibration motions, the approach used in qualification by testing is to define a required response spectrum (RRS) and produce a test response spectrum (TRS) which envelopes it. In addition, the peak table input acceleration should equal or exceed the ZPA of the RRS. Finally, to properly account for vibration buildup and for low-cycle fatigue, there are some requirements for the duration and number of cycles of the input waveform. (Duration requirements is one area being considered for inclusion in a revised IEEE-344 Standard.)

There are many methods in use for generation of test input motions. Single frequency tests include continuous sine wave, sine beat, decaying sine, and sine sweep. Multi-frequency tests include time history, random motion and combination of random motion and sine wave based single frequency tests.

Wave form and signal duration used for equipment qualification should possess certain duration, amplitude and frequency content. The basic requirement is that the excitation time history should conservatively simulate the strong motion portion of a postulated earthquake event. Unlike the time history method, the response spectrum method (the most widely used technique for equipment qualification) does not define the exact wave form or duration of the excitation. It also does not define the number of cycles of each frequency which is present in the signal. Since sufficient number of strong motion response cycles and adequate signal duration is required for the equipment peak response to build up, the use of response spectrum method for qualification (especially in testing) and enveloping the required response spectrum may not be sufficient. The distribution of frequency in the wave form and the stationarity of this distribution is another important parameter in the generation of an adequate wave form.

Along with the different types of motions that might be used to seismically qualify equipment, both single and multiaxis tests have been used. While seismic ground motion occurs simultaneously in three directions, in the past and in certain cases now, single axis excitation may be used. Single axis testing may be appropriate if the equipment being tested can be shown to respond independently in each of three orthogonal directions or if the mounting is such that the equipment effectively sees a single axis motion. When this is not the case, biaxial testing is used.

The performance characteristics of a seismic test facility determine its usefulness to a qualification program. These characteristics include:

- Load capacity: This is the test specimen weight that can be accommodated by a facility.
- Frequency range
- Table dimensions
- Maximum table acceleration
- Maximum table displacement
- Maximum table velocity

The last two limit maximum acceleration at the lower frequencies.

Other characteristics need to be considered in the ability of the seismic test facility to provide services such as electrical power and compressed air to the table-mounted test specimen. For pumps, the capability to provide on-table motion power and to accommodate a piping loop to recirculate pumped fluid must be considered. This latter consideration places a severe limitation on the ability to test pumps. Only one such test has been identified during the investigations conducted for this report. This was a seismic test of a relatively small component cooling water pump by Union Pump Company.

Table 6-2 summarizes the range of seismic test capabilties of facilities in the United States and Japan of interest to testing of mechanical equipment. By far, the largest table capacity exists at the Tadotsu facility in Japan. This facility has been designed to shake very large components including pumps in a piping configuration.

Shake table testing of large mechanical equipment is usually not feasible and qualification by analysis alone may not be sufficient as discussed previously. In some cases it may be possible to shake table test components of the equipment and use the testing results in combination with analysis and/or scale model testing to demonstrate that the full scale equipment is qualified. Another approach to combined methods is to use "in-situ" testing described below to supplement analysis.

<u>In-situ Testing</u>

In-situ testing is a form of testing that has general application to mechanical equipment and is used, in conjunction with analysis, to aid in qualifying a broad spectrum of mechanical equipment. The results of in-situ testing are usually in a form that can be used to:

(1) Determine if the equipment is flexible or rigid and, if applicable, the equipment's resonant frequencies.

(2) Determine the amount of damping and establish if damping in excess of that normally allowed can be used in the analytical qualification process.

(3) Verify an analytical model by comparing the measured resonant frequencies and calculated frequencies.

In-situ testing is usually performed on equipment in their as-installed condition at a plant site. In other cases, the testing may be done at other locations, for example at the factory, in order to obtain data during the design or application stage or because the location is more convenient. In such cases, the test mounting and

TABLE 6-2

SEISMIC TEST FACILITY CAPABILITY

FACILITY SIZE	SPECIMEN LIMIT		TABLE MAXIMUM INPUT				MODE OF OPERATION
	SIZE LxWxH - ft.	WEIGHT lbs. (G limit)	MAXIMUM FORCE	ACCELERATION g	VELOCITY in/sec	DISPLACEMENT inch	
Small (U.S.A.)	6.7x4x8	3,000 lbs.	7500	3.2@1500 lbs		6	Single Axis H or V
Medium (U.S.A.)	20x20x20	20,000 lbs.		1.5g H 1.0g V	25 H 15 V	10 H 4 V	Simul-taneous Biaxial
Large (Japan)	39x39x -	1,000,000 lbs. H 440,000 lbs. V	790,000	0.55g H 1.0g V		2.5	Single Axis H or V

anchorage details should be similar to the anticipated installed mounting and anchorage details. A strong argument in favor of in-situ testing at the plant site is that the test mounting and anchorage details are identical to the as-installed details. If the tested anchorage is substantially different from the installed conditions, it will be necessary to demonstrate that significant response modes are not introduced by the actual mounting details.

In-situ is a term encompassing several techniques having application to mechanical equipment. These include simple static tests in addition to more complicated tests to determine the response of the equipment to static or dynamic forces.

Static In-Situ Tests. By measuring the deflections caused by known applied static loads, the stiffness of various complex mounting or anchorage details may be assessed. If the applied loads are of sufficient magnitude, the anchorage or mounting detail may be actually qualified by the in-situ tests.

Transient In-Situ Testing. This type of testing measures the transient response of the equipment to an applied forcing function or initial condition. The measurement of resonant frequencies, mode shapes and damping ratios is the goal of this type of testing. The methods of obtaining a transient response include:

> Ambient excitation: This method employs the background vibration at the equipment's mounting to excite its resonant frequencies. This method is not effective for damping measurements if the background vibration is low, however, the major system frequencies can usually be identified. This type of testing is almost always conducted as a preliminary investigation prior to conducting other in situ testing.
>
> Snapback testing: In snapback testing, a static force is applied to the structure, causing an initial displacement. The force is suddenly released, allowing the structure to undergo free vibration with the initial displacement as the initial condition. This method has been used successfully to test large exhaust stacks, heavy equipment including steam generators, piping systems and other equipment that is flexible with intermediate range resonances.
>
> A method of applying the force and a method of quick release are required. Winches, cables, cranes, or hydraulic rams can be used to apply the force; high-speed

hydraulic valves, unlatching mechanisms, or frangible links (which fail at a known force) can be used to release the force.

The level of response in snapback testing is dependent on the initial displacement. For individual component testing, the level of response attainable is limited only by available force application. Therefore, the snapback method is particularly useful for large component tests.

In practice, it is sometimes difficult to isolate vibration modes of interest because the method tends to excite more than one mode at a time. Pre-analysis and experience as well as repeated tests with force application at different locations may be necessary for effective isolation of individual modes.

Impulse tests: Impulse loadings with rubber mallets or hammers are often sufficient to excite the fundamental vibration modes of large mechanical equipment. In recent years, the development of impulse hammers and supporting instrumentation has made this technique the dominant method in use today.

High-level impulse loading methods include the use of mechanical pulse generators and chemical rockets. Mechanical pulse generators produce force by drawing metal bars with variable cross sections through a cutting tool. High-level impulse loading methods have found application to military hardware qualification but application to qualification of nuclear components is minimal.

Random-transient testing: This method uses small electromagnetic or hydraulic shakers to provide a short burst of random (white noise) excitation to the equipment followed by shut-off of the shaker. The free-vibration response is then measured.

Steady State or Random In-Situ Testing. These include:

Sinusoidal testing: This method of testing employs an eccentric mass, hydraulic or electro-mechanical vibrator to excite a structure or component with a sinusoidal forcing function. The use of variable frequency vibrators allows the resonances of the component to be determined by observing the component's response (spikes) to varying input frequency. Sufficient exciting force is often obtained by portable, strap-on vibrators.

Random testing: This method of testing employs an electromechanical or hydraulic vibrator to excite the structure or component with a random (white noise) excitation. The transfer functions observed between input force and output response at various points provide the necessary data for frequency, mode and damping determination.

Identification of Modal Parameters. The process of extracting the modal characteristics of a structure from experimental data is called modal testing. The impact (hammer) testing technique is probably the most popular excitation technique used in modal testing work today. The primary reason for its popularity is that the test set-up is typically fast and easy to do as compared to a shaker test. It should be noted, however, that the impulse technique can be very poor for testing a structure with moderate nonlinearities (e.g., gaps).

To provide the reader with an understanding of the power of the modal testing technique, the case study shown in Figure 6-3 has been prepared. A component of a larger, rigid mechanical equipment assembly is considered. As indicated in Figure 6-3, this component is supported by a cantilever with varying rigidity and base mounting plate. A calibrated impulse hammer and a calibrated (piezoelectric type) accelerometer are used along with a dual-channel spectrum analyzer as test equipment. The testing effort can be conducted by a single individual without assistance.

The component mass and support can be generally idealized as a single-degree-of-freedom system. An in-situ impulse test can provide an estimate of the system frequency and damping and even provide the effective mass and stiffness of the submount component as it is mounted on the mechanical equipment.

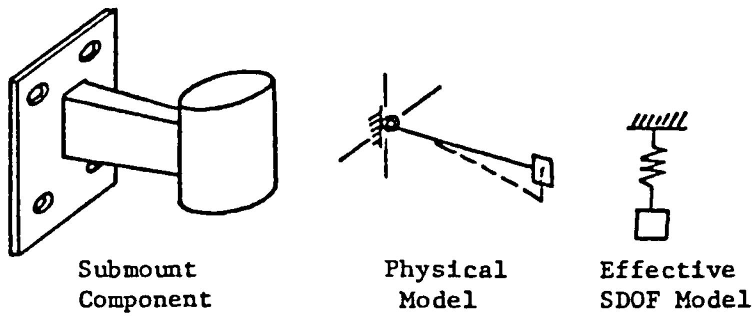
The maximum of the transfer function denotes the system frequency (f_n=25 Hz). An estimate of the system damping uses the formula

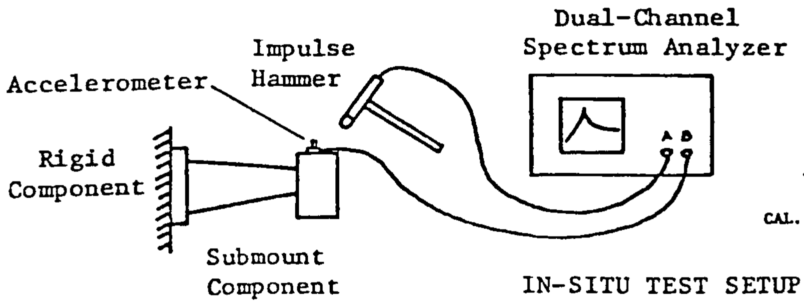
$$\beta = \frac{\Delta f}{2f_n}$$

Where ΔF is measured at the half-power points (peak amplitude less 3dB) See Figure 6-3.

In carrying out a modal test an accelerometer is mounted on the component and the component struck with the instrumented hammer (a very light blow is best). The response signal is digitized and immediately processed by the spectrum analyzer to provide the transfer function (both amplitude and phase) by comparing the applied

CASE STUDY: IN-SITU TEST OF SUBMOUNT COMPONENT





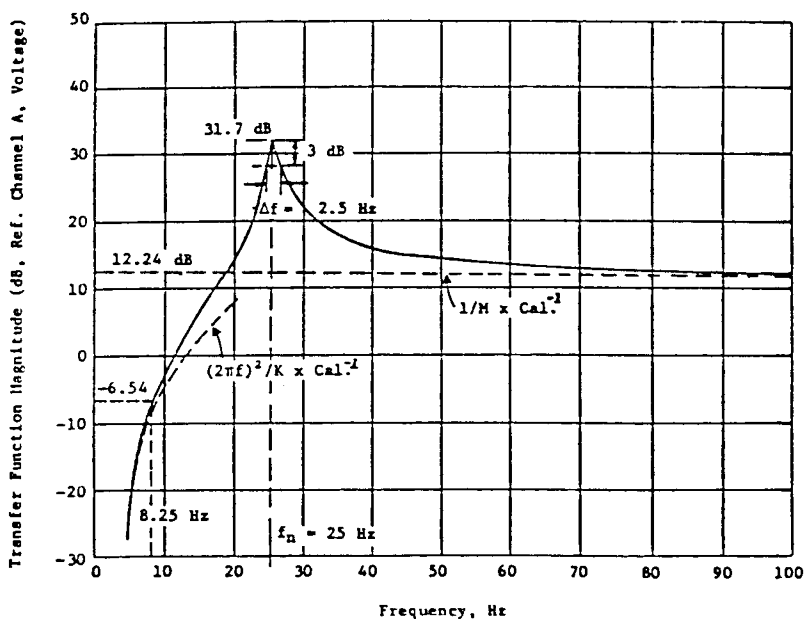
IN-SITU TEST SETUP

CALIBRATION DATA

HAMMER: CAL. HAM = 300 lb/V

ACCELEROMETER: CAL. ACC = 1.25 g/V

$$\text{CAL.} = \frac{\text{CAL. ACC}}{\text{CAL. HAM}} = \frac{1.25 \times 386.4}{300} = 1.61 \frac{\text{in.}}{\text{s}^2\text{-lb}}$$



INERTANCE TRANSFER FUNCTION

EFFECTIVE MASS

$$r = f/f_n = 100/25 = 4$$

$$\left(\frac{1}{1 - 1/r^2}\right)\left(\frac{1}{M^*}\right) = 10^{12.24/20}(\text{CAL.})$$

$$M^* \cong 0.161 \text{ lb-s}^2/\text{in.}$$

EFFECTIVE STIFFNESS

$$r = f/f_n = 8.25/25 = 0.33$$

$$(1 + r^2)\left[\frac{(2\pi f)^2}{K^*}\right] = 10^{-6.54/20}(\text{CAL.})$$

$$K^* \cong 3929 \text{ lb/in.}$$

EFFECTIVE DAMPING

$$\beta = \frac{\Delta f}{2f_n} = \frac{2.5}{2(25)} \cong 0.05$$

FIGURE 6-3: IN-SITU TESTING CASE STUDY

force and the acceleration response of the submount component. The resulting transfer function is shown in Figure 6-3. It should be noted that the units of the ordinate are presented in dB units (20 log [A/F] where A is the amplified response and F is the input forcing function). The high frequency asymptote provides an estimate of the system mass while the low frequency asymptote is proportional to the inverse stiffness times frequency squared. Thus, this processed data display provides all the information necessary to characterize the effective single-degree-of-freedom system for the component. In practice, this measurement of transfer function would be repeated several times (i.e., several hammer blows) and averaged to obtain a valid estimate.

For more complex sub-components or a large component, the same basic procedure would be utilized except the hammer blow would be given to several locations on the equipment and the resulting transfer functions (the transducer would stay in one reference location) would be recorded.

A curve fitting process would then be conducted to provide estimates of the system frequencies, mode shapes, and modal damping factors. Specialized microprocessor based equipment is available (with software) for this purpose.

6.5 SEISMIC QUALIFICATION METHODOLOGY GUIDE

As has been pointed out, seismic qualification of mechanical equipment may involve a number of different methods and combinations of methods. These range from a very simple static analysis which is justified if the equipment is rigid or can be modeled as a simple structure for the mode of failure of concern to a complete functional dynamic test which may be necessary if the equipment can not be adequately modeled, the failure mode is not easily related to the structural response or the mode of failure is unknown. For a given piece of equipment, there is not one correct method or combination of methods for qualification. In fact, a number of different approaches may be adequate.

What is done in qualifying mechanical equipment is to choose a qualification approach which is sufficient to demonstrate the equipment's ability to perform its intended function during and after an earthquake. Like the design process itself, seismic qualification of equipment is a process requiring considerable engineering judgement.

Figure 6-4 is a flow diagram illustrating a process of establishing the method of seismic qualification to be applied to a particular mechanical component. The process integrates the types of qualification methods and some of the issues discussed above. The methodology consists of a series of basic questions whose answers can determine the methods most appropriate for the particular equipment being evaluated. These are discussed below.

1. Is the equipment similar in design and application to equipment already qualified.

   Comment: It is likely that much mechanical equipment can be qualified by experience. This experience may include previous qualification of similar equipment, application in a facility (nuclear or non-nuclear) which has experienced an earthquake and/or operational surveillance experience.

2. Is the equipment characterized by active, functional failure modes that are structural or quasi-structural?

   Comment: Most mechanical equipment is subject to seismic failure modes that are structural or quasi-structural. Examples are shaft deflection, bearing load and wear ring interference. In the event that the equipment can be mathematically modelled and its failure modes can be described as structural or quasi-structural, then qualification by analysis is an applicable methodology. In the event that there are active, functional failure modes that are not structural or quasi-structural, then some testing may be necessary.

3. Is there a shake table of sufficient capacity?

   Comment: The capability to test the equipment must be evaluated in light of the capacity of test facilities. This capability must include the ability to provide supporting services to operate the equipment while it is mounted on the shake table. The capability of testing facilities is discussed in Section 6.7.

   In the event that shake table capacity is not sufficient, then it may be necessary to do scale model testing or to use analysis as the best method of qualification.

4. Can equipment be qualified by static analysis?

   Comment: In order to be capable of qualification static analysis, the equipment must be of such a configuration that seismic forces and accelerations can be rather simply related to a structural or quasi-structural response.

5. Is equipment rigid?

   Comment: This can be determined by analysis or by in-situ testing. The equipment is considered rigid if the resonant frequency is 33 Hz or greater.

6. Are acceptance criteria exceeded?

   Comment: In the event that the acceptance criteria are exceeded, then the evaluator has several choices:

   - Perform a dynamic analysis to remove some of the conservatism inherent in the static analysis and static equivalent methods. (This is often sufficient.)

   - Change the configuration of the equipment. For example, add a support, stiffen the pump motor stand.

   - Qualify by testing.

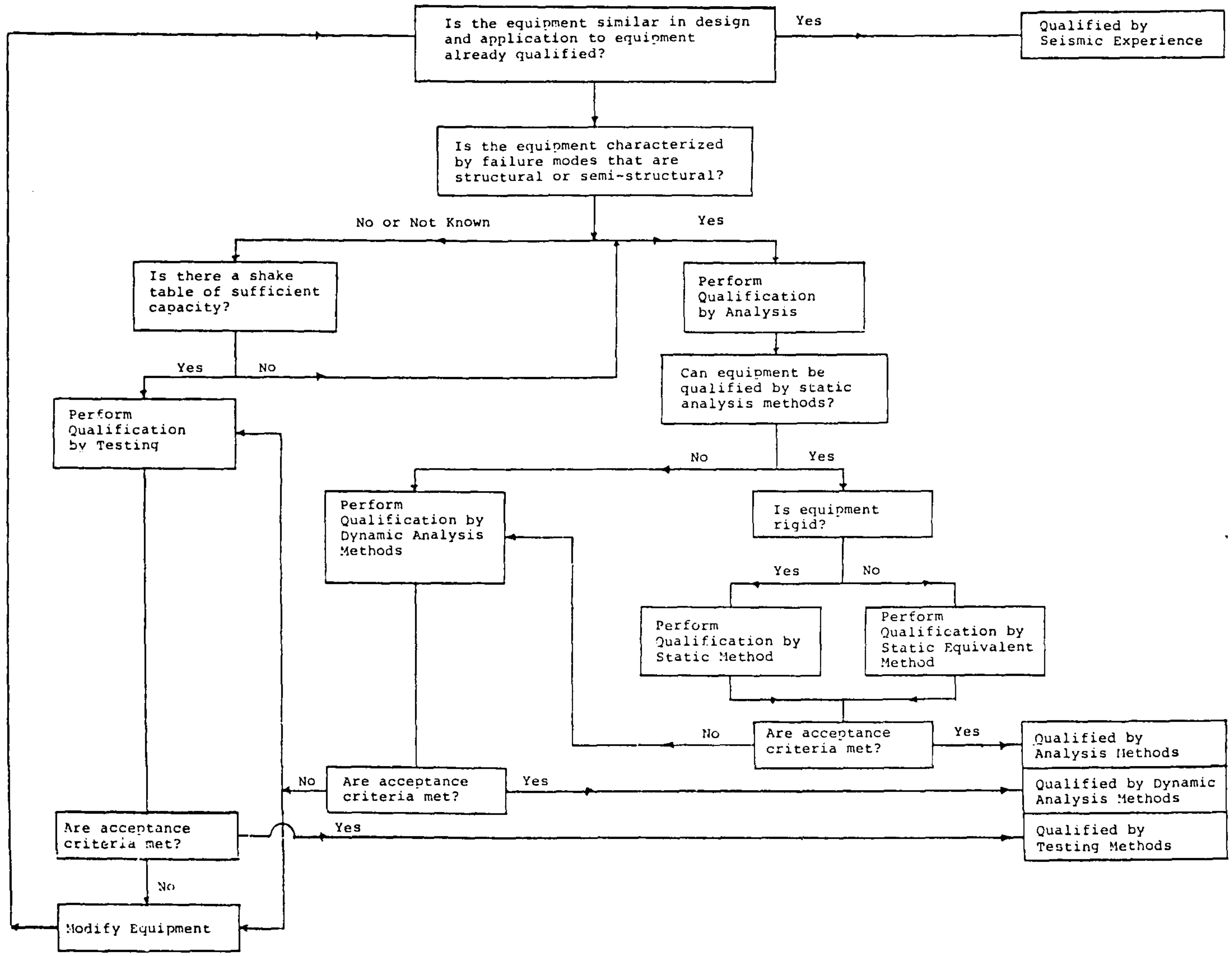


FIGURE 6-4: SEISMIC QUALIFICATION GENERAL PROCESS

Section 7
EVALUATION OF SPECIFIC COMPONENTS

INTRODUCTION AND SUMMARY

The mechanical equipment which is discussed in this section has been selected as representative of the equipment types which are used in nuclear safety related service. The list of active, safety-related mechanical equipment will vary from plant to plant, depending on a number of factors which also distinguish one plant from another. Despite some differences in details, however, the selection is considered appropriate as a guide for reviewing qualification status and developing appropriate qualification methodologies for mechanical equipment.

The types of mechanical equipment are described in sufficient detail to provide a basis for visualizing potential failure modes which can be addressed by proper consideration of environmental, seismic and dynamic loading conditions.

The elements of an adequate qualification program, other than design/operating experience, include:

- Surveillance
- Maintenance
- Materials Review
- Analysis
- Testing

How each of these elements should be combined to comprise an appropriate qualification method is discussed for several types of mechanical equipment.

Visits to manufacturers of mechanical equipment and discussions with equipment users provided useful insights to ensure that the appropriate qualification methods were both thorough and practical.

Included in this section is a discussion of the design, application, and qualification of several types of valves used in nuclear service. The types considered herein are:

- Gate Valves
- Butterfly Valves
- Check Valves - Vacuum Breakers
- Globe Valves
- Safety/Relief Valves
- Main Steam Isolation Valves

These valves vary considerably in design and service conditions. By discussing the design, application, and failure modes of this broad spectrum of valve types the common and special concerns of valve operability will become apparent. Based on this perspective, the section includes a suggested general process for the operability qualification of valves.

7.1 GATE VALVES

Description

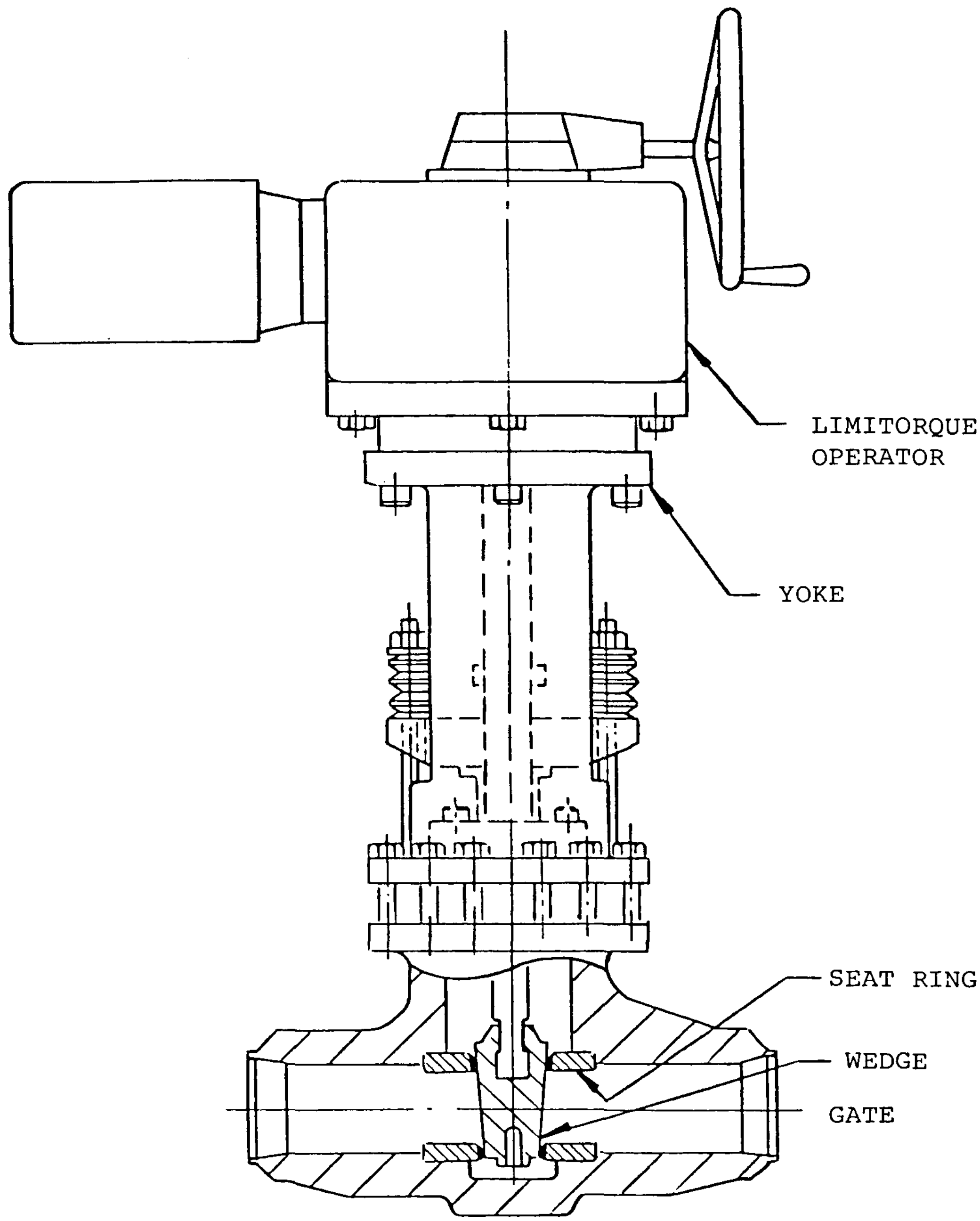
Figure 7-1 shows a cutaway of a typical motor operated gate valve. The principles of operation are simple. In the closed position, as illustrated, a gate or wedge intercepts the flow path preventing flow by pressing against stationary seat rings on either side of the gate. In the open position, the gate is withdrawn completely from the flow path permitting unimpeded flow.

The gate is connected to a stem which permits it to be pulled up, pushed down or held in position. The upper section of the stem is a worm gear which can be driven by an electric motor through a suitable gear train or alternatively by a manual hand wheel.

There are a number of detail design variations employed. Significant among these is the flexibility designed into the gate. Occasionally, the gate is made solid between the two sealing faces. More often material is removed at the periphery of the gate between the two sealing faces. This relief permits a degree of flexibility of the gate that helps prevent the gate from jamming in the seat rings. In some cases the gate is made in two pieces which are joined at a hinge at the stem connection. This also provides flexibility. In almost all cases the valve gate and seats are all metallic.

The main structure and pressure containing envelope are provided by the body which is usually cast alloy steel but may be of different materials or formed by other methods such as forging or machining. Weld ends, as shown, are often used in power plants, although flanged ends are also used. The body provides a space or bonnet above the line of flow into which the gate is drawn to open the valve.

A bonnet cover closes this extension of the body and carries the stem guide and seal components. A bolted bonnet closure is shown. Clamped bonnets are also commonly used. The stem seal shown is a typical double packing arrangement. The packing is usually a set of flexible graphite impregnated asbestos rings which is pressed into the space between the bonnet and stem by the packing compression bolts. When sufficiently compressed, the packing presses inward onto



(CLOSED POSITION)

FIGURE 7-1: TYPICAL MOTOR OPERATED GATE VALVE the stem providing a seal. The second or upper packing of a double packing design causes any leakage from the lower packing to be forced out the seal leakoff connection.

Gaskets, if used, seal the bonnet cover and flanged ends. They are normally of the metal and asbestos composition types.

Electric motor operators are frequently used to open and close valves. These operators surmount the valve bonnet on a yoke and engage and drive the valve stem worm gear with a high slip A.C. induction motor or a D.C. torque motor. Auxiliary electric devices such as limit switches and torque switches are provided on the valve operator to prevent overload. The gate, stem, guides and operator are integrated to meet specific operating conditions. One of the more important of these is the maximum pressure differential across the gate when the gate is to be opened or closed. With proper design, operation with very large pressure differentials can be achieved.

The qualification of the electric motors and their associated electrical devices are not considered in this report as they are covered in electrical equipment qualification programs. However, the motor operator does produce a large mass on the extension of the valve stem which must be considered in the seismic qualification of the valve. Furthermore, the torque capability of the operator sets the capability to open and close the valve against operating and design pressure differentials. This aspect of operability is discussed further later in this section.

Operating Characteristics

The flow losses through an open gate valve are very low. When the gate is fully withdrawn, the flow path resembles a straight segment of pipe with only minor impediments to flow. The flow characteristic is very nonlinear with gate position. Little flow resistance occurs from full open to 2/3 closed. From approximately 2/3 closed to closed, resistance increases at an ever increasing rate with closure.

Application

Gate valves are used widely in nuclear plants. They are used where low flow losses are important and/or very tight shutoff is needed. Use as control valves is generally avoided because of the highly nonlinear flow characteristics and relatively slow speed. Sizes range from 3/4" to 28" and larger. Nuclear applications include isolation valves, shutoff valves, injection valves, and maintenance valves.

Failure Modes

The failure modes to which a gate valve is potentially susceptible are: failure to close, failure to open, excess seat leakage, and excess stem leakage.

Failure to Close. Failure to close can be defined as the inability of the valve operator to move the gate from the full open position to the full closed position in a specified time duration. The dominant causes of failure include:

1. Large flow induced forces
2. Excess stem seal load
3. Low torque switch settings on motor operators

Large flow induced forces can occur in applications where the valve must close to shut off flow from a broken downstream pipe. Under these assumed conditions the flow through the valve can reach critical velocity as the valve approaches the closed position. The result can be a large pressure drop across the gate forcing the gate against its guides and increasing friction. A few tests have been conducted to reproduce these conditions. Reference (24) describes and reports the results of seven such tests.

Because tests are very expensive, time consuming and require large flow facilities, a substitute test has been devised and used in many instances. In this test the valve is closed and full pressure is applied across the gate. The valve is then commanded to open. If it is successful in opening, the inference is made that it would also close with full pressure across the gate. The supporting argument is then made that the pressure drop across the gate while closing off flow to a broken pipe cannot exceed the full pressure and, therefore, capability to close is demonstrated. This method has a reasonable technical rationale and is attractive from a cost and time standpoint. Recent full flow shutoff tests have also been conducted on gate valves. On one occasion during the testing reported in Reference (24) a gate valve opened with full pressure but did not fully close against full flow. This failure to close appears to be random as all other valves closed when tested. This same test series (24) also showed the need to establish the required torque relationship between the valve and the motor operator to provide for closure against full flow.

Excess stem seal load can result from pressing the stem packing too tightly against the stem by over tightening the packing compression bolts. The end result is that the packing to stem friction combination with other stem loads such as flow and pressure induced loads can be significantly higher than the operator can overcome. This condition may develop during packing maintenance either inadvertently or in an attempt to overcome the effects of incorrect packing and/or stem scoring. Functional testing after maintenance may be used to guard against over tightening of packing and other related anomalies. For example, at least one plant measures and evaluates valve motor current during functional tests following packing maintenance (60).

Low torque switch settings can lead to failure to close by turning off power to the motor operator prior to the valve reaching full closure. Incorrect field setting of the torque switch can occur during maintenance. Functional testing is an appropriate check test following maintenance.

Failure to Open. Failure to open a gate valve has safety significance only if failure to open results in the failure to deliver flow necessary to fulfill a safety function. Typical safety related applications are primary coolant safety injection valves, and essential cooling water supply valves.

Failure to open can be defined as the inability of the valve operator to move the gate from the full closed position to the full open position in a specified time duration. The dominant causes of failure are:

1. Hydraulic lock-up
2. Excess stem seal load
3. Incorrect torque switch settings
4. Thermal distortion

Hydraulic lockup can occur when an incompressible fluid such as water in the bonnet above a closed gate is at a higher pressure than the line pressure. Under such a condition there is a net hydraulic force tending to close the valve. To occur, both gate seal faces and the stem seal must seal tight to maintain the bonnet pressure. Given these conditions, it is possible that the motor operator will be unable to start the opening of the valve.

The possibility of having a gate valve closed at high pressure and then having the line pressure lowering or the bonnet heating makes hydraulic lock up a potential cause of failure to open. Hydraulic lockup is not necessarily a qualification issue. Strictly speaking, it is a system design problem where it exists. Nevertheless, its identification and resolution falls easily within the scope of a qualification review. A small hole through one of the gate faces will prevent lockup by relieving the bonnet to the line on one side of gate. This is not necessarily an attractive solution because it defeats one of the two gate seals. Providing means to vent the bonnet is another solution. This would require active components (valves) and a drain.

The first step to avoid lockup is to provide a review that identifies if the valve can be in a state of hydraulic lockup when the valve must perform its safety-related function. This involves the following evaluations.

1. Can the valve bonnet be pressurized when the valve is required to perform a safety-related opening function? If yes, proceed to answer the following.

2. Is the bonnet fully pressurized or are conditions leading to the possibility of bonnet pressurization in existence at the time the valve opening surveillance test is conducted? If yes and the surveillance tests are successful, no further action is required. If surveillance tests are not entirely successful then mechanical changes, as noted above, should be

considered.

3. If pressurized bonnet conditions can occur when the safety-related opening function is required but the conditions are not fully in existence during the surveillance test, then changes to the surveillance test, special tests or mechanical changes to the valve should be considered.

Excess stem seal load causes, effects, and solutions are the same in failures to open as in failures to close. See the discussion under Failure to Close.

Low torque switch settings also have the same causes, effects, and solutions in failures to open as in failures to close. See the discussion under failure to close. Also, if the torque switch permits excessive closing force, the gate may be effectively jammed between the seat rings making opening in combination with other design basis loads doubtful. Setting for excess closing force can be motivated by an attempt to achieve a tight seat seal despite scored or worn seat faces. Obviously, design basis torque switch settings should be established and maintained.

Thermal distortion may cause the gate to be effectively clamped by the seat rings so that the operator cannot open the valve. Large temperature gradients within the valve are required to generate significant distortion. Rapidly changing process temperature or less likely rapidly changing environmental temperature may develop the necessary temperature gradients. An evaluation similar to that for bonnet pressurization may be required.

Excessive seat leakage. Excessive seat leakage is most likely caused by scoring and wear of the gate and seal ring mating faces. This is based on the observation that most gate valves are subjected to a leak rate test by the manufacturer and leak checked during plant startup and periodically thereafter. Thus, their intrinsic ability to adequately seal is demonstrated repeatedly. Therefore, good maintenance is the key to avoiding this kind of failure.

Excessive stem seal leakage. This failure is similar to that of excessive seat leakage in that the key to preventing it is good maintenance. Acceptable seal leakage is usually demonstrated in the factory and periodically during service life. The packings for active, safety-related valves are almost always asbestos impregnated with graphite or solid graphite neither of which are significantly affected by environmental conditions such as temperature, humidity or radiation.

It should be noted that when a gate valve is closed, the gate isolates the stem seal from both upstream and downstream line pressure. In isolation valve applications this observation can be very useful in allaying concerns for stem seal leaks.

7.2 BUTTERFLY VALVES

Description

Figure 7-2 shows a cutaway of a typical motor operated butterfly valve. The basic elements of the valve are the disc, body shaft and seat. The circular disc is rotated by the shaft through 90° from full closed to full open. In the full closed position the disc fills the bore of the valve and presses against the seat. In the full open position the edge of the disc is presented to the flow providing minimum flow resistance. Because of the rotary motion of the disc the seat must be flexible if tight shutoff is to be achieved. The seat material is usually made of a compliant elastomeric material. When tight shutoff is not required metal seats can be used.

The shaft is supported by bearings above and below the disc and sealed by a compressive packing. The packing is compressed by the packing follower until the packing is pressed inward onto the shaft effecting a seal. The shaft and, hence, disc are positioned either by a power actuator surmounting a yoke or by manually moving a hand wheel or a lever. Electric motor, pneumatic and hydraulic operators are used as power actuators.

The body provides the necessary structure, pressure retaining envelope and line mounting capability. The body is mounted in the pipe line either by bolting it to two flanges or it is bolted between flanges sealed by gaskets.

Ref. No.	Part Name
1	Body
2	Packing Flange Stud
3	Packing Flange Stud Nuts
4	Bearing
5	Spacer Tube
6	Packing Ring
7	Packing (V-Ring)
8	Packing Follower
9	Packing Flange
10	Belleville Washer
11	Retaining Ring Set Screw
12	Taper Pin
13	Shaft
14	Thrust Washer (Outer)
15	Thrust Washer (Inner)
16	Vane
17	Seal Ring
18	Seal Ring Energizer
19	Retaining Ring

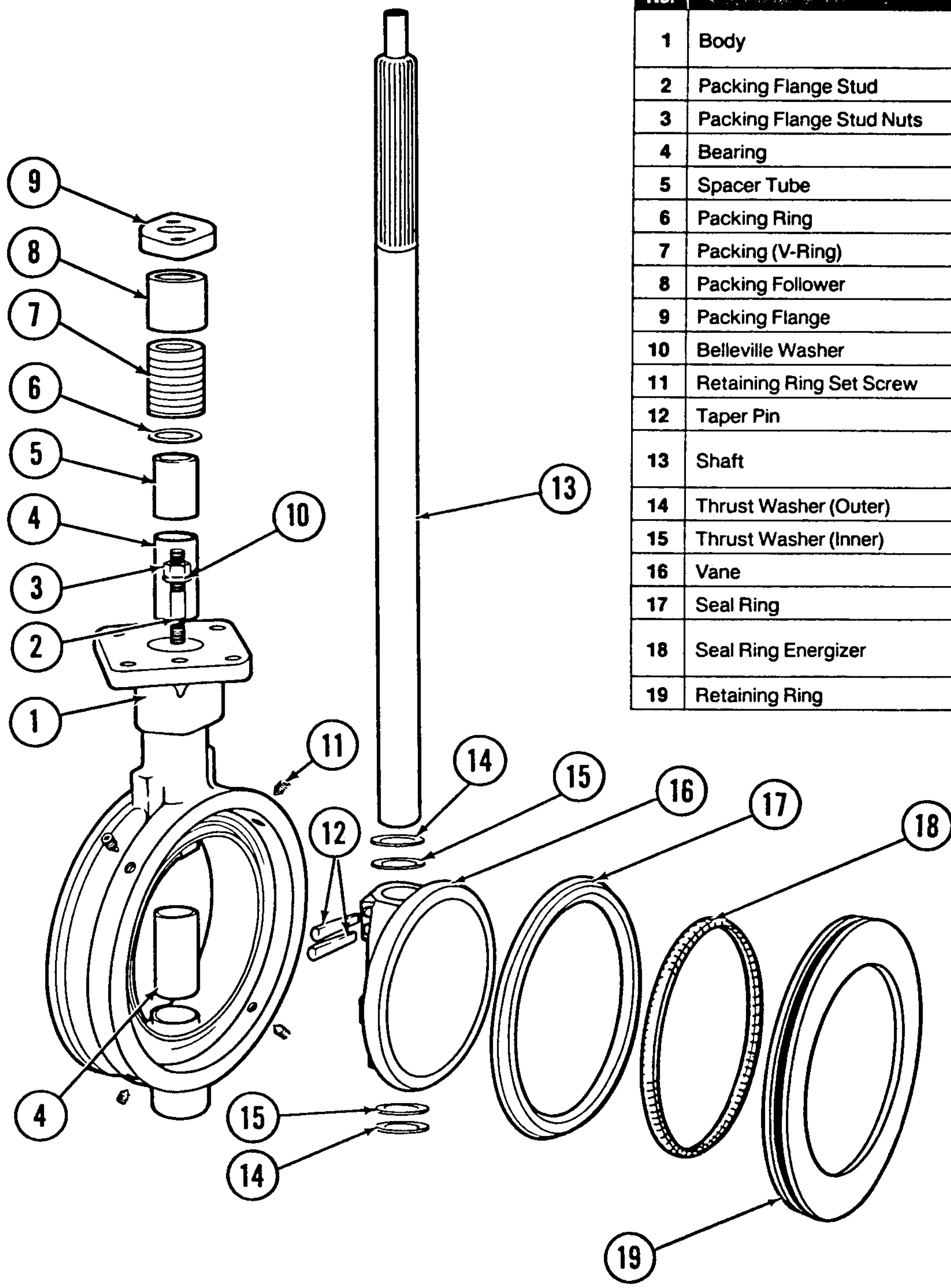


FIGURE 7-2: WAFER TYPE BUTTERFLY VALVE

Except for seat and shaft seals, gaskets and portions of the operator, metal construction is used. The shaft seals are normally graphite impregnated asbestos or wholly graphite and thus not susceptible to environmental aging. Similarly, gaskets are normally of asbestos with metal reinforcing. In tight shutoff application, the seat, as mentioned above, must be flexible and usually is made of elastomeric material. Because these materials are susceptible to aging and wear, the valve construction permits their replacement.

Operating Characteristics

The flow losses through a fully open butterfly valve are fairly low because the edge profile of the disc presented to the flow is a relatively small fraction of the flow area hence the fluid velocity need not increase greatly while passing the disc. Also, only minor changes in flow direction are needed in passing through the valve.

In the fully closed position, the flexible seat can develop a tight seal.

At intermediate positions flow losses increase at an increased rate as the closed position is approached. This means the valve has a non-linear flow characteristic which must be considered in flow control or pressure control applications.

Torque requirements to position the valve are modest. The flow or pressure head generating torque around the shaft tends to be balanced. The degree to which the center line of the shaft is offset from the disc tends to unbalance the torque. Further, the center of force on the disc moves as the disc position is changed causing the degree of unbalance to change. Depending on the particular geometry and kinematics of the design, the torque on the shaft may reverse as the shaft is rotated.

Application

Butterfly valves are commonly used in power plants including nuclear plants as shutoff valves, isolation valves and maintenance valves. Use as control valves is occasionally found but is usually avoided because of inherent nonlinear characteristics. Heavy throttling conditions are particularly avoided because of wear consideration.

Elastomeric seat materials restrict applications to less than 750 psig at 150°F and less than 100 psig at 950°F or lower depending on the particular material used. These restrictions essentially eliminate use in the primary system of light water reactors.

The relatively small size of the valve makes it attractive where space and/or weight considerations are important. Also the low operating torque requirements make the operator and its power requirements relatively small, yielding minimal space weight and power consumption.

Failure Modes

The failure modes of a butterfly valve are the same as those of a gate valve: failure to close, failure to open, excess seat leakage and excess stem leakage.

Failure to close. Failure to close can be defined as the inability of the valve operator to move the disc from the full open position to the full closed position in a specified time duration. The dominant causes of failure are:

1. Large flow induced forces
2. Excess stem seal load
3. Low operator overload protection settings
4. Operator failure

The first three causes are analogous to the causes of gate valve failure to close. While the flow forces on a butterfly valve tend to be balanced so that large torques are not required of the operator, operators may have to be sized to be able to close the disc and cut off large flows such as those due to a broken downstream line.

Excess stem seal load applied in an attempt to achieve a seal can contribute to overloading of the operator by causing high friction between the shaft and packing. The analogous cause of failure was discussed in the gate valve section and reappears in other valves where an operating shaft passes through a seal.

If operator overload protection devices are used, as in electric motor operator applications, the potential for setting these devices too low is present. Again this potential is similar to that in the gate valve.

Failure to open. Failure to open can be defined as the inability of the valve operator to move the disc from the full closed position to the full open position in a specified time duration. The dominant causes of failure are:

1. Excess stem seal load
2. Low operator overload protection settings
3. Operator failure

See the discussion above (Failure to Close) as these are similar contributors to Failure to Open.

Failure to open because of high pressure differential across the disc is not considered likely since the forces are well defined and so can be accommodated in the design. Further, factory confirmation tests are quite practical.

Excess seat leakage. Butterfly valves are one of the few process valves which typically use an elastomeric seat. They are, therefore, susceptible to environmental degradation leading to excess leakage.

Excess stem seal leakage. This failure potential is similar to gate valves.

7.3 CHECK VALVES

Description

Check valves permit flow in one direction only. Although they are made in a number of different configurations, the principles of operation are similar. Pressure in one direction opens the valve permitting free flow while flow and pressure in the opposite direction closes the valve and prevents flow.

A typical check valve is illustrated in Figure 7-3. Pressure in the indicated flow direction causes the gate to swing up into the bonnet opening the valve and permitting flow. If flow attempts to reverse, the gate swings down and closes against the stationary seat.

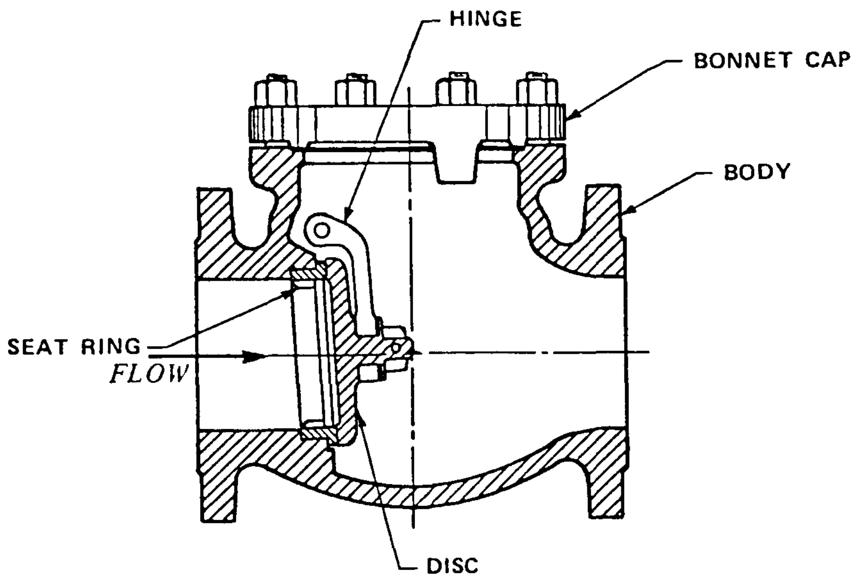


FIGURE 7-3: TYPICAL SWING CHECK VALVE

The bonnet is closed by a bolted cover similar to a blind flange which permits access to the gate for inspection and maintenance. This cover is sealed by a gasket usually of asbestos and steel construction. Similarly, small bolted closures are typically provided at opposite ends of the shaft around which the gate rotates permitting disassembly. The gasket seals are similar in construction to that of the bonnet cover.

Aside from the gaskets, the materials of construction are almost uniformly metal with cast or forged alloy steels or stainless steels being typical. The valve illustrated has weld ends for permanent welding into the piping system. This is a typical installation technique although bolted flange ends are occasionally used.

Position indicating elements are sometimes added to the shaft to detect rotation of the shaft. In some special applications power devices are added to permit the gate to be rotated if there is no tendency for flow. Such devices permit inservice testing to confirm that the gate is free and functional. The design is such that the testing device cannot interfere with the normal function of the check valve.

Operating Characteristics

Check valves are on/off devices permitting flow in one direction only. Opening and closing action is direct, positive and rapid being directly controlled by the flow and pressure direction. Pressure drop in the flow direction is fairly small. Position of the valve cannot be controlled externally without the addition of special features.

Application

Check valves are in very general use in nuclear plants. Typical uses are as isolation valves on lines flowing into the primary system or containment. Also, they are used to prevent negative pressure in tanks and pipe (i.e., as vacuum breakers).

Failure Modes

The failure modes to which a check valve is potentially susceptible are failure to open or close, excess seat leakage and excess shaft seal leakage.

Failure to open or close. Intrinsically, check valve operation is simple and direct as described above. The most likely cause of failure is wearout caused by oscillation of the gate during normal operation. At very low flow conditions it is possible that flow forces are not great enough to hold the gate firmly open and instead the gate swings slightly in the flow stream. If protracted, it is possible that the shaft and guides will wear out and the valve will be inoperable. Further, the failure or incipient failure may not be apparent. Numerous failures of check valves have occurred in a variety of applications (84, 85, 86, 88, 89, 90, 91, 92, 93). Application review, surveillance testing and periodic inspection are necessary to preclude this type of failure.

Excess seat leakage. Like gate valves and all metallic seated valves, the key to avoiding excess seat leakage is surveillance testing and good maintenance.

Excess shaft leakage. In many applications, the shaft is sealed behind a static cover thus precluding this type of failure. However, in some applications, access to the shaft is provided for position indication and/or testability. Here a shaft seal is required and may leak. Surveillance and maintenance of these seals is required to assure leak tightness.

7.4 GLOBE VALVES

Description

This description of a globe valve is of a general nature because there are a great many styles and variations of valves which are termed globe valves. A diaphragm air operator will also be described as they are commonly used in conjunction with globe valves.

Figure 7-4 shows a cutaway section of a typical globe valve. The essential elements of the valve are the seat and plug. By positioning the plug with respect to the seat, the restriction to flow through the valve can be controlled from no flow area to full open. The shape, contour or "trim" of the plug is carefully selected to give the desired relationship between plug position and flow at fixed pressure drop or pressure drop at a fixed flow.

The main structural and pressure containing element is the body. It is typically cast alloy steel, however, other materials and forming methods such as forging or machining are employed. The valve shown in Figure 7-4 is to be welded into a pipe line although flanged ends are also used.

Bolted to the body are the cover and bonnet. The cover provides access to the plug's lower guide while the bonnet permits insertion of the plug into body and supports the upper plug guide and stem seal. The stem is a rod attached to the plug by which the plug may

TYPICAL NON-METALLIC MATERIALS		
NO.	NAME	MATERIAL
1	Diaphragm	Neoprene & Rayon Fabric
2	Yoke Gasket	Asbestos Fiber
3	String Packing	Asbestos & Graphite
4	Packing	Asbestos & Graphite
5	Body Gasket	304 S.S. w/ Asbestos
6	Seat Ring Gasket	304 S.S. w/ Asbestos

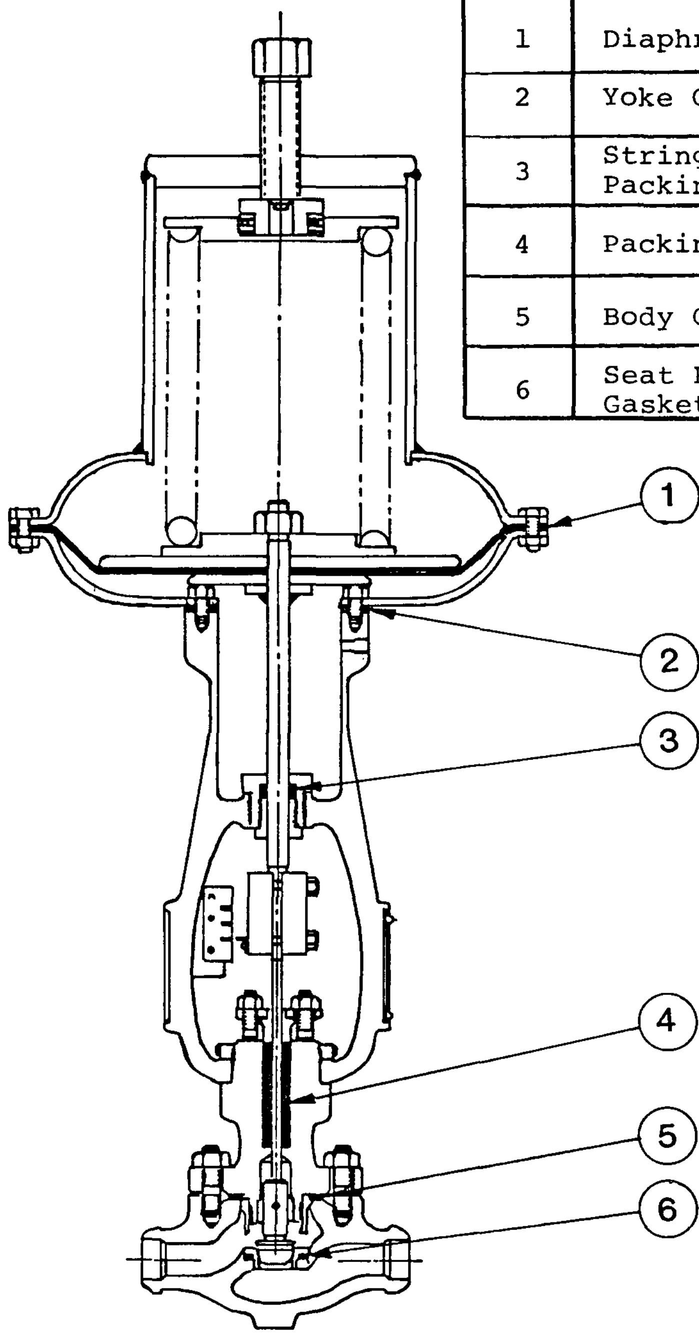


FIGURE 7-4: TYPICAL GLOBE VALVE

be moved up and down and held in position. The stem seal shown is a typical double packing arrangement with a leak off tap between packing sets. Each packing set is usually made up of rings of flexible impregnated asbestos which is pressed into the space between the bonnet and stem by the packing compression bolts. When sufficient compression on the packing is reached, the packing presses inward onto the stem providing a seal but not preventing the stem from being moved. Leakage which may occur can be drained away via the leak off tap to a convenient sump.

Attached to this globe valve is a diaphragm type air actuator. In the particular actuator shown, the spring holds the stem up and hence the valve open. When air is applied to the diaphragm its pressure forces the stem down in the close direction. This is commonly termed "air to close." By controlling the pressure on the diaphragm the plug position is controlled.

The spring and air pressure relationship can be reversed in what is known as the "air to open" configuration.

Gaskets are used on the cover and bonnet to seal. These gaskets are usually of the metal and asbestos combination type particularly at high pressure and temperatures.

Operating Characteristics

Globe valves are characterized by a controllable pressure drop versus stem position. They have a relatively short stroke as compared with a gate valve. They have a relatively high pressure drop when full open and, generally, are not as effective in achieving tight shutoff as with a gate valve. This is not to say that tight shutoff cannot be achieved.

Speed of operation is limited primarily by the operator because the stroke is usually short. Stroke times of a fraction of a second are achieved with special operator designs.

Application

Globe valves are used where flow or pressure drop control is needed. Additionally, they are used where the low pressure drop of a gate valve is not required. In these applications, the globe valve is usually preferred because it is smaller and has a shorter stroke and hence needs a smaller operator. Thus, even when flow control or pressure control are not needed they may be the preferred on/off valve. Consequently, the globe valve in its many forms is found throughout power plants.

Failure Modes

The failure modes to which a globe valve is potentially susceptible are failure to reach a demanded position (i.e., closed, open or intermediate), excess seat leakage and excess stem leakage.

The discussion of the causes is abbreviated because of their similarity to the discussion of gate valves and butterfly valves.

Failure to reach a demanded position. This failure can be caused by:

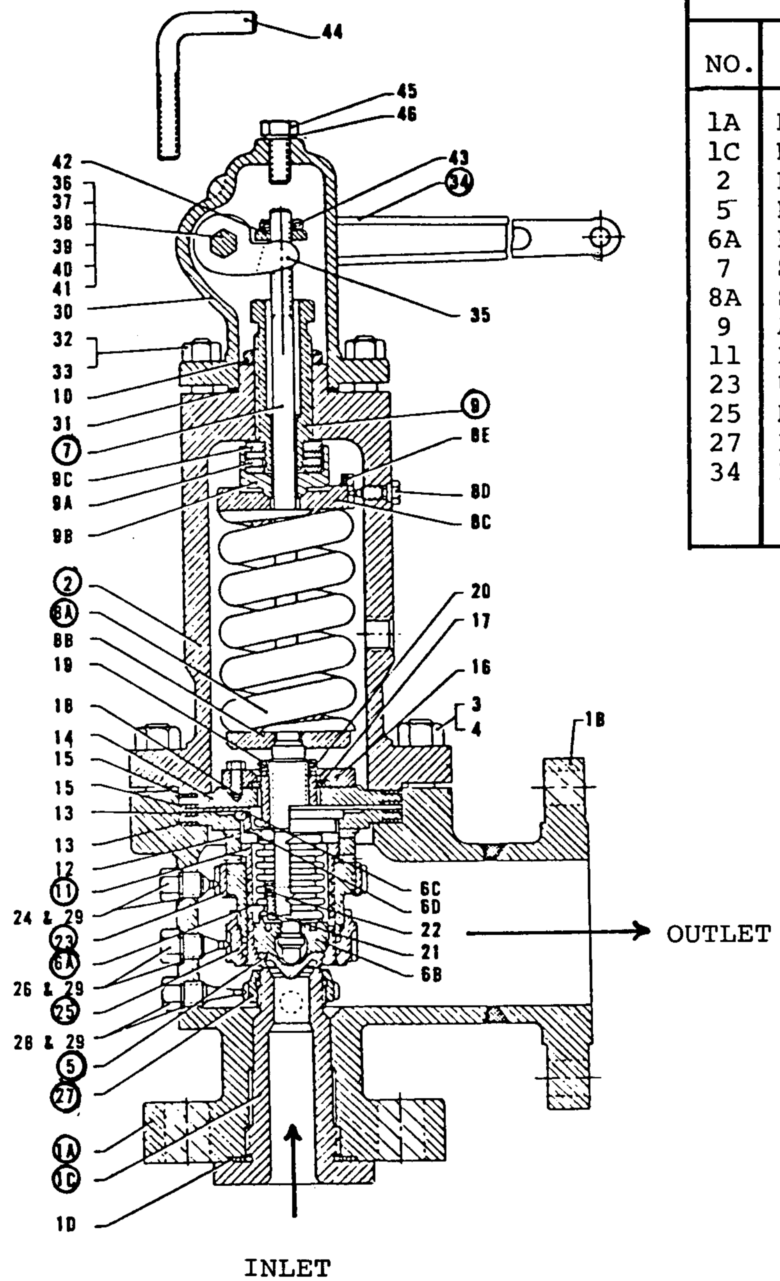
1. Large flow induced forces
2. Excess stem seal load
3. Low operator overload protection settings
4. Operator failure

If large flow induced forces are present during performance of the throttle valve's safety function then there may be a need to qualify the valve by test. If the flow induced forces are present during the valve's period surveillance test, then the surveillance test should suffice as the qualification test. Unless the operator has been previously qualified, its elastomeric components may be susceptible to environmental degradation and should be reviewed.

7.5 SAFETY AND SAFETY/RELIEF VALVES

Description

This description will begin with a direct acting safety valve. As the name implies safety valves are used to dependably prevent overpressure. The relief function will be explained later. This can be done since the safety function is independent of the relief function.



KEY TO SELECTED PARTS	
NO.	PART NAME
1A	Body
1C	Nozzle
2	Bonnet
5	Disc
6A	Bellows
7	Spindle (or Stem)
8A	Spring
9	Adjusting Bolt
11	Disc Carrier
23	Upper Adjusting Ring
25	Middle Adjusting Ring
27	Lower Adjusting Ring
34	Lifting Lever

FIGURE 7-5: SAFETY VALVE

Figure 7-5 shows a cutaway section of a typical safety valve. The essential elements of the valve are illustrated in Figure 7-6. The spring holds the disc down against the nozzle preventing the escape of fluid from the inlet. Pressure at the inlet applies a force upward on the disc. When pressure is high enough its upward force causes the spring force to be overcome and the disc moves upward. This permits fluid to escape through the nozzle and thus limit system pressure. The inlet pressure at which the disc moves or opens is the "set pressure." When pressure goes below set pressure the spring force again forces the disc against the nozzle closing the valve.

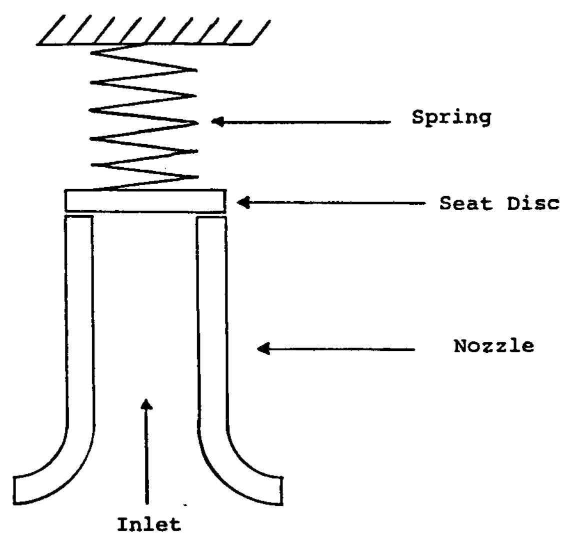


FIGURE 7-6: ESSENTIAL ELEMENTS OF A SAFETY VALVE

This simplified explanation of operation belies specific characteristics of these simple operations which are a practical necessity. Chief among these are the characteristics that when the valve opens at set pressure it does so with a rapid movement to the full open position. The valve is said to "pop." Pop action is required to assure prompt pressure relief when set point is reached and is achieved by the size and shape of the disc, its carrier and the lower adjusting ring. See Figure 7-5. As the name implies, the adjusting ring permits adjustment of the valve to achieve the pop action. Pop action on the order of 0.05 seconds is typical.

A second characteristic, "blowdown," is controlled by flow passages and adjustable blowdown rings. Blowdown is defined as the difference between the pressure at which the valve opens (set pressure) and the pressure at which it closes and is usually expressed as a percentage of set pressure. Five percent is a typical blowdown value. The blowdown characteristic assures that valve does not "chatter," that is, attempt to open and close at the same pressure.

The important flow capacity characteristic is controlled by the minimum flow area in the nozzle. Capacity is usually dictated by critical flow at this area.

In summary, the principles of operation of a safety valve are simple while the important refinements of its characteristics such as pop action and blowdown are achieved by subtle fluid controls.

The main structural element of the valve is the body to which are attached the nozzle and bonnet. Typically, the body is alloy steel cast or forged. It is almost always flanged to permit removal for maintenance and testing. The bonnet or a similar structure supports and restrains the spring and its associated stem, guides and adjusting bolt which permits precise setting of the spring load to control set pressure.

The inlet, outlet and bonnet closures are sealed by gaskets usually of a metal and asbestos composite type. The stem penetration into the cavity of the body is sealed by a bellows as shown in Figure 7-5.

A lifting lever as shown in Figure 7-5 may be provided. This lever and its associated linkage permits the valve to be manually opened. Note that this device and the automatic opening of the valve at set point are mutually independent in that either can open the valve and neither can prevent opening. This manual mode of opening can be replaced by an air operator to produce a safety/relief valve as shown in Figure 7-7.

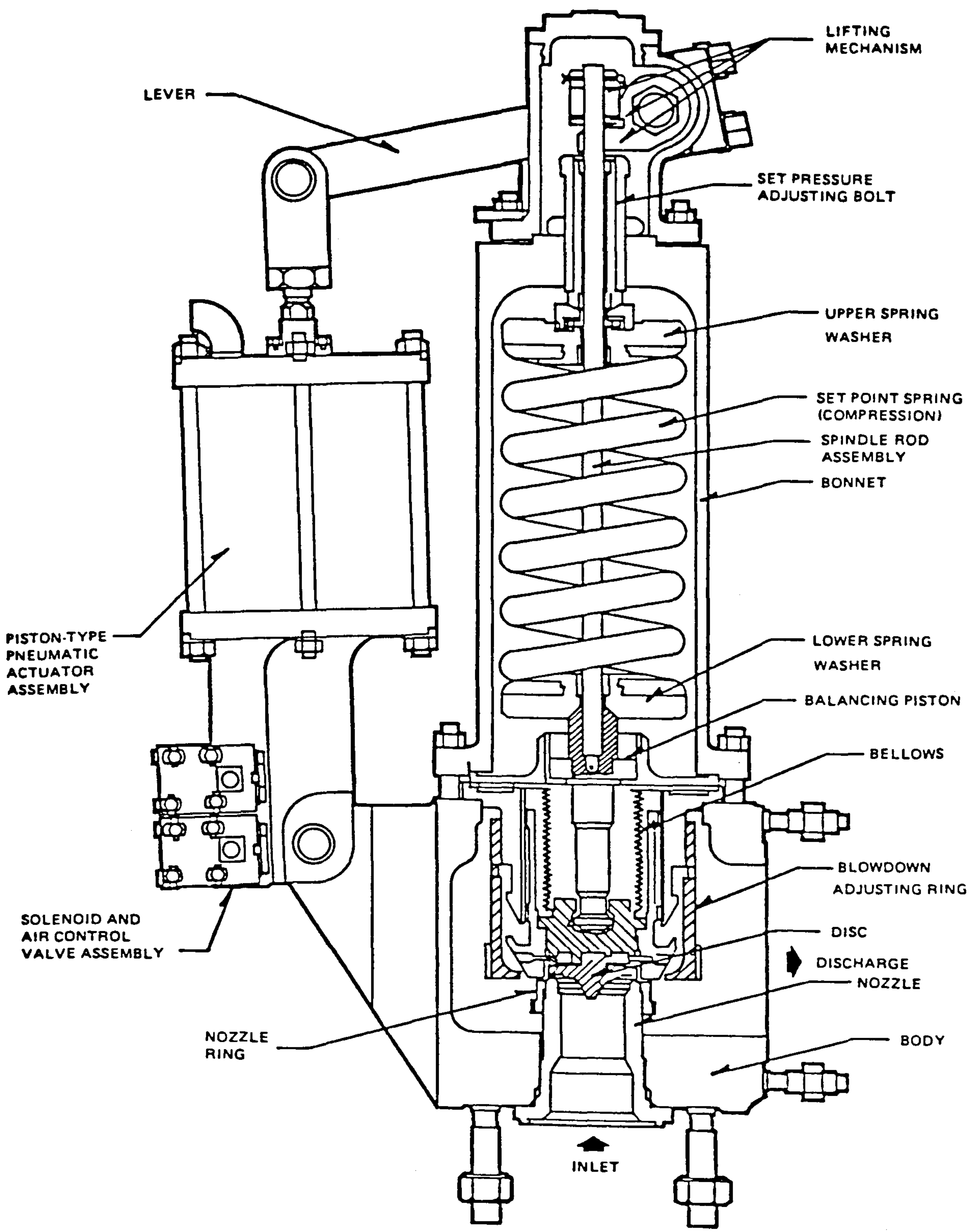


FIGURE 7-7: SAFETY/RELIEF VALVE

The air operator is a simple piston in cylinder arrangement with suitable pivots. When air is supplied below the piston it moves upward forcing the stem and hence the disc up, opening the valve. Exhausting the air permits the valve to be closed by the main spring. Control of the supply air is accomplished by solenoid operated valve and downstream air valves in the air control subassembly. Thus an electrical signal can cause the valve to open on command providing the relief mode of operation. A typical air control subassembly is shown schematically in Figure 7-8. It is often provided with redundant solenoid valves and air valves to assure opening on demand. When the solenoid coil is de-energized the solenoid closure spring holds the solenoid plunger assembly against the main seat preventing supply air from reaching the air valve. When electrically energized, the plunger assembly is magnetically drawn upward opening the main seat and closing the back seat, thus admitting air to the air valve. When de-enegized the plunger returns to close the main seat and thus exhausting the air to the air valve via the back seat.

Figure 7-9 is a sketch of a downstream air valve. The air valve acts as an amplifier of the solenoid valve. When air is not supplied by the solenoid valve, the air valve spool is held in the position shown by the spring with the air operator ported to exhaust and the supply air port blocked by the central land on the spool. When air is supplied by the solenoid valve, pressure on the end of the spool forces the spool to compress the spring porting the air supply to the air operator and blocking the exhaust port with the central land. Thus a small flow of air from the solenoid valve results in a large flow of air to the air operator opening the safety valve in the relief mode.

When de-energized the solenoid valve as described before exhausts air on the end of the spool permitting return to its normal position. This exhausts air from the air operator and closes the safety valve.

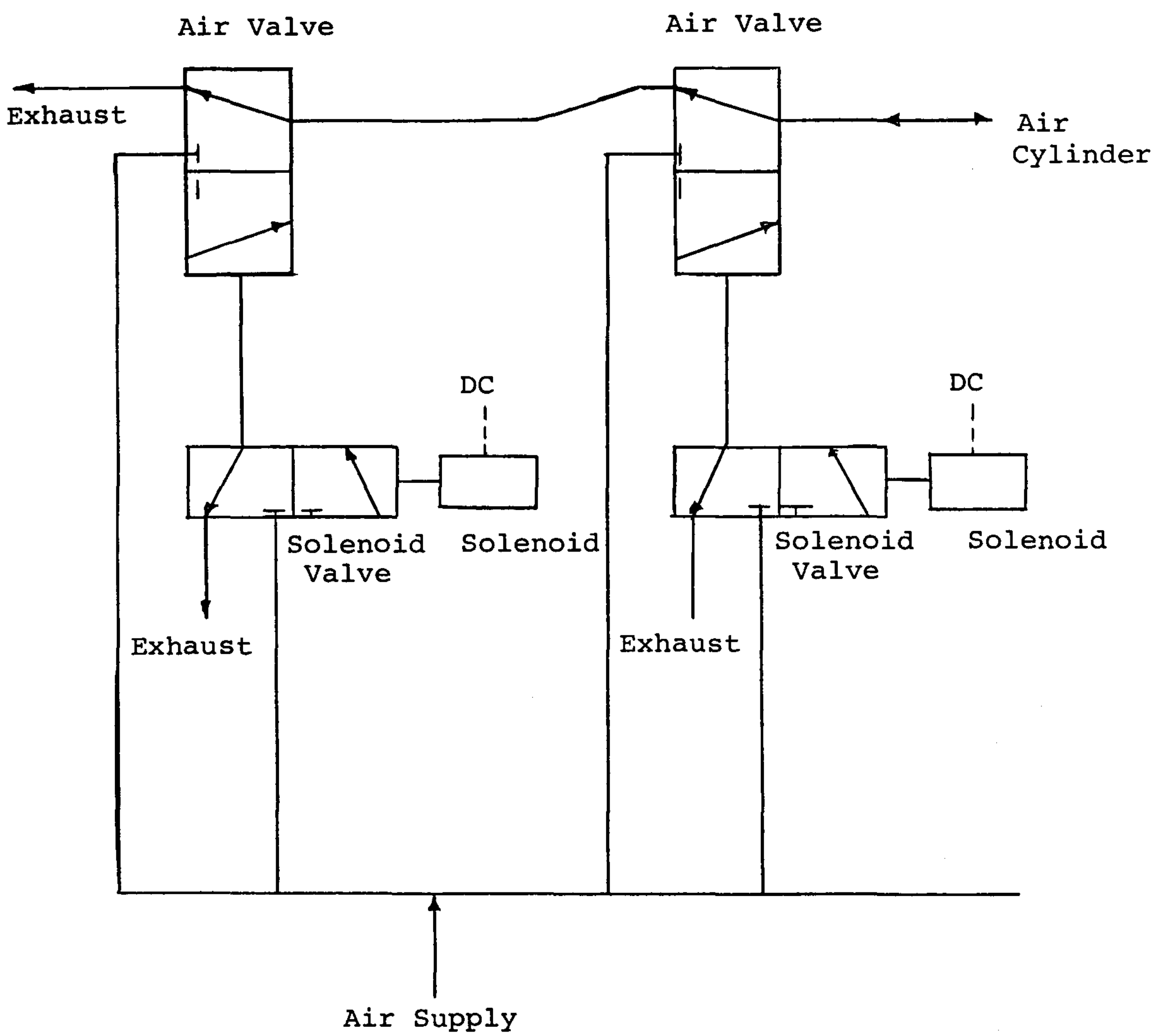


FIGURE 7-8: AIR CONTROL SUBASSEMBLY

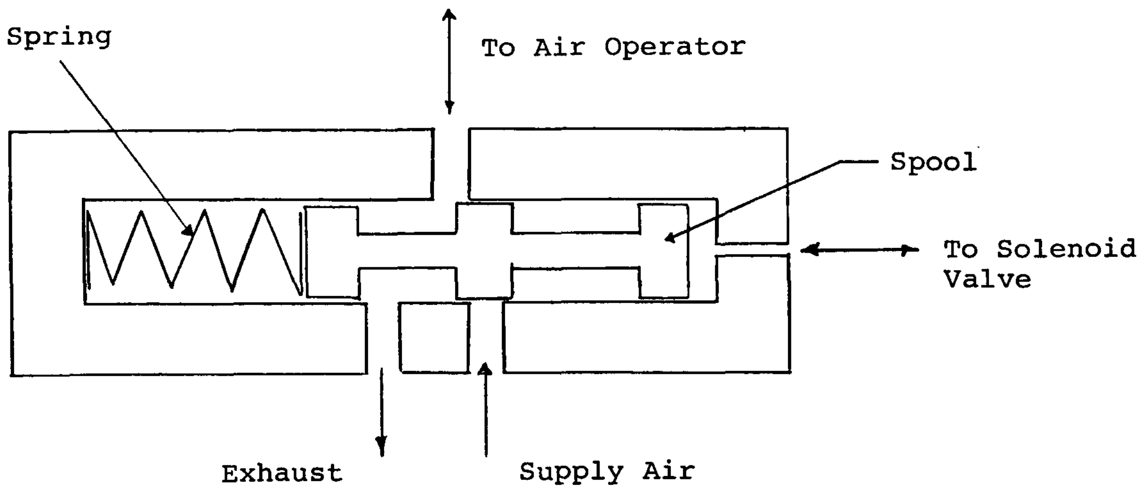


FIGURE 7-9: AIR VALVE

Operating Characteristics

Some of the operating characteristics of a safety-relief valve are apparent from the foregoing description. The valve will open automatically and abruptly in about 0.05 seconds, when inlet pressure reaches prescribed pressure. This is called the safety mode. It will reclose when pressure has been reduced to a fraction of opening pressure, nominally 95%. Closure is abrupt but not as fast as opening. Choked flow in the nozzle controls flow rate.

Opening in the relief mode is initiated by an electrical signal that results in the valve being opened by an air operator regardless of inlet pressure. Closure requires removal of the electrical signal. The speed of operation is rapid being on the order of 0.5 seconds. It can open in either safety or relief mode and neither mode can inhibit the other mode of opening.

Leak tightness of the disc to nozzle depends on proper design and maintenance of the mating parts and difference between operating pressure and set pressure. A difference of 10% and preferably 15% of set pressure is needed to make a tight seal possible.

Applications

Safety or Safety-Relief valves are used in pressurized systems to give extremely high assurance that system overpressurization cannot occur. Applied to boilers, they back up normal pressure controls and thus rarely are used. The relief mode is used as an alternative or backup means to control pressure in the boiler such as depressurizing an isolated boiler.

Failure Modes

Failure modes are here divided between the two operational modes of the valve: safety mode and relief mode.

Safety Mode Failures

Failure to open at set point: Drift of the opening pressure may occur between prescribed functional tests. This drift is known to be small, +-3% of set point, with +-5% being an extreme value. Causes of drift are subtle and not well understood but are probably related to aging of the spring, friction in guides and minor adhesion at the disc to nozzle seal interface. Operating experience and plant analyses have shown that opening pressure drift is tolerable and therefore does not constitute a functional failure. Extreme upward drift of opening pressure, while conceivable, does not occur as a practical matter.

Failure to close: In the safety mode of operation, reduction in inlet steam pressure causes the force balance on the disc to change such that the valve closes automatically, thus failure to close is extremely unlikely.

Failure to remain open: This type of failure can be characterized by the valve closing prematurely or not remaining fully open, i.e., when the inlet pressure is high. It can result in rapid opening and closing usually called "chatter." Chatter may occur when the fluid passing through the valve is other than steam. For instance, if the valve is designed and adjusted for steam service and then operated with water, chatter may result. Special testing will usually be required to demonstrate immunity to this type of failure. Chatter is not necessarily a failure to open and relieve pressure.

Failure to seal: Failure to seal may occur in either the safety mode or relief mode and is usually caused by flow-induced damage to the seat surfaces. Leakage is also strongly influenced by the difference between set pressure and operating pressure. In other words, the closer it is operated with respect to set pressure, the more likely it is to leak.

Relief Mode Failures

Failure to open: Assuming that pressurized air is supplied and an electrical signal is present, failure to open would likely be caused by failure of the solenoid valve or the air valve. Since these devices use elastomeric seals and insulating materials, which are susceptible to environmental aging, there is a potential for aging to cause them to stick or otherwise fail to move. Normally, the solenoid valve is qualified as a Class IE electrical component.

Failure to close: Like failure to open, failure to close is most likely to be caused by degradation of the environmentally susceptible elastomers in the solenoid or air valve.

Failure to conserve supply air: Excess supply air may be consumed because of leaks in the controls or air operator. This can lead to a failure to open in the relief mode if the air supply is fixed, stored in an accumulator and is exhausted. The dominant cause of excess air leakage is failure of elastomeric gaskets and seals in the solenoid valves, air valves, and air operator. Since they are environmentally aging susceptible, preventive maintenance, material application review and testing is required.

7.6 MAIN STEAM LINE ISOLATION VALVES

Description

Figure 7-10 shows a cut away view of one type of main steam line isolation valve (MSIV). In more general terms it is a "Y" pattern globe valve with a double acting air operator and spring assisted closure. The main elements of the valve proper are the body, poppet, pilot, stem and bonnet cover.

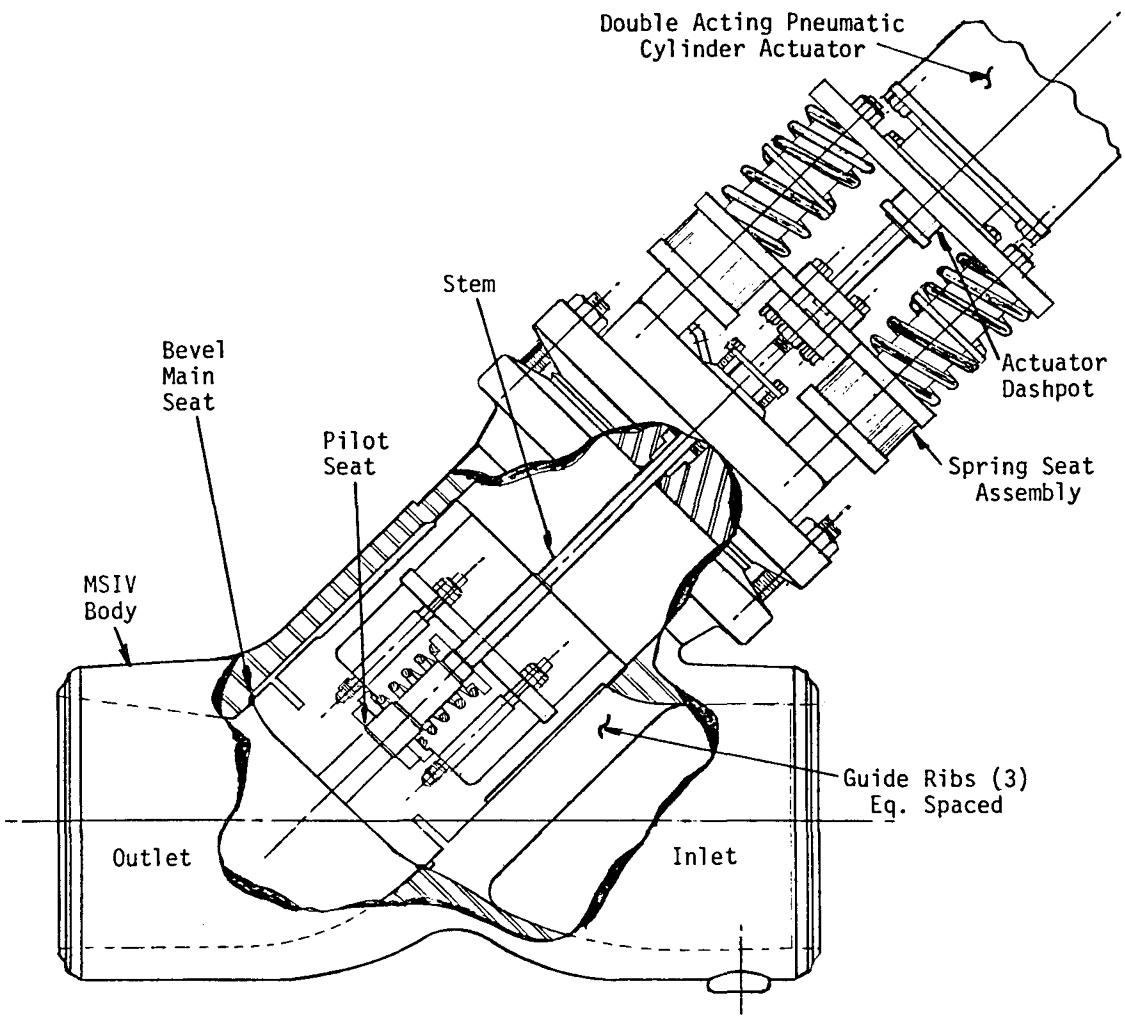


FIGURE 7-10: MSIV CROSS-SECTIONAL VIEW
(Reference 116)

The poppet, shown in the closed position, rests on the stationary seat in the body providing a seal and shutting off flow in the indicated direction. In the open position, the poppet is withdrawn by the stem into the extension of the body called the bonnet. With the poppet in this position, the valve offers little resistance to flow. When the valve is being closed, the flow tends to assist closure. When it is being opened, up stream pressure tends to hold it closed. In the particular valve shown, the initial motion of the stem first opens the pilot. Flow through the pilot raises downstream pressure and thus reduces the operator force required to withdraw the poppet. As the stem continues to move it engages and withdraws the poppet.

A bonnet cover is bolted to the extension of the body and sealed by a gasket usually of the composition asbestos and steel type. The cover provides a guide for the stem and stuffing box for the stem seals. Graphite impregnated asbestos or solid graphite fills the stuffing box and is compressed to provide a seal with the stem.

The operator of this particular valve is highly specialized to give maximum assurance that the valve will close on demand or in the event of control air or power failures. Compression springs on four yoke rods apply a force on the stem tending to close the poppet against the seat. To open the valve pressurized air is applied beneath the piston of the air cylinder opening the valve against the springs. To close the valve air is exhausted from beneath the piston and the springs close the valve. Closure is also assisted by flow through the valve. Air is also applied to the top of the piston to force the poppet against the seat to provide a better seal.

The speed at which the valve can move is limited by a hydraulic dash pot. This device provides a force proportional to the velocity of the stem and in the opposite direction to the motion. It can be adjusted to yield the desired speed limit.

Application of air to the air cylinder is controlled by a system of solenoid valves and air operated three and four way air valves. Loss of electric power to the solenoid valves will cause the poppet to close by exhausting air from below the air cylinder piston. This also causes air to be applied to the top of the piston to hold the valve closed.

Operating Characteristics

MSIVs are open or closed devices and are thus not suitable for controlling flow ratio or pressure. In the full open position resistance to flow is small because of the large flow area and minimal disturbance to the direction of flow. Closing speeds are rapid (3 to 10 sec). Tight shutoff is assisted by compression springs and line pressure, both of which force the poppet against the seat.

Applications

The particular valve described is used as an isolation valve on Boiling Water Reactor main steam lines. In this application they are applied in pairs (for redundancy) to each steam line, one inside the containment and one outside. In the event of a break in the steam line outside containment they would close and isolate the reactor from the leak.

Some PWRs also use MSIVs. In this case the closure speed is usually less and gate valves are used.

Failure Modes

The failure modes to which a MSIV are potentially susceptible are failure to close, excess seat leakage and excess stem leakage. Note that failure to open is not a safety-related failure mode because there is no safety related requirement that MSIVs be able to open.

Failure to close. The potential causes of failure to close are:

1. Large flow induced forces
2. Operator failures

MSIVs are required to be capable of interrupting flow to a broken downstream pipe.

Operator failures are not likely because of the "fail safe" design. While portions of the operator and its controls contain electrical and pneumatic components which are environmentally sensitive, most failures such as electrical shorts and seal leaks result in closure of the valve. Sticking in position of solenoid or air valves are conceivable failures which would result in failure to close. Sticking might be caused by environmental degradation of elastomeric components. The solenoids controlling air-to-open supply the valve operator are qualified as electrical equipment.

Excess seat leakage. As with other metallic seated valves, seat leakage is controlled by maintenance practice.

Excess stem leakage. MSIV stem packings are similar in design to those of gate and globe valves and are, therefore, amenable to the same maintenance practices to avoid this failure.

7.7 GENERAL VALVE QUALIFICATION PROCESS

Introduction

It is apparent from the sampling of valves described and their potential operability concerns that many of the concerns are common to many valves. Also, it must be noted that there are many possible variations in valve designs other than the typical designs considered.

At the present time there is considerable activity in the industry to prepare standards for the qualification of valves. Specifically, ANSI Standard B16.41-1983, Functional Qualification Requirements for Power Operated Active Valve Assemblies for Nuclear Power Plants (78), has been issued. Standards are in the preparation stage for self-operated valves, check valves and power-operated relief valves.

The main emphasis of ANSI Standard B16.41-1983 is to qualify valve assemblies using the concept of a parent valve and candidate valves. The standard defines a qualified parent valve assembly as one which has been qualified by all appropriate testing required by the Standard and a qualified candidate valve assembly as one qualified by analyses that demonstrate design similarity with the parent valve assembly with additional testing as appropriate. The ANSI Standard provides requirements on applicable testing and guidance on qualification of Candidate Valve Assemblies.

In summary, industry standards being prepared appear to rely mainly on testing to qualify prototype or parent designs with analyses limited to qualifying other sizes and design variations. The process discussed below recognizes that less rigorous approaches may be applicable to equipment in service and that other aspects of qualification, namely in service maintenance and surveillance are also a vital part of the overall qualification process.

The process suggested progresses in two major steps. First, identification of the specific operability requirement and second, appropriate qualification methods to provide assurance of meeting the requirement.

Identification of Operability Requirements

It is essential that the specific safety related operability requirements be identified. In many applications, if this is carefully done qualification is greatly simplified. It is suggested that an operability requirements sheet be prepared for each valve being considered since the requirements differ considerably for different valves. A sample sheet is shown on Table 7-1. Having identified the general requirements, the qualification process can proceed.

Qualification Process

Before undertaking an independent assessment of operability, it is recommended that the supplier of the valve be requested to supply any applicable analysis and test reports for the particular valve or similar valves. Many analyses and tests have been performed in recent years which are not in the public domain but may be available to a valve owner. These reports may be directly applicable or may require some additional evaluation to justify applicability. In any case, discussions with the valve manufacturer will determine specific open items, if any, which should be addressed in a qualification program.

Depending on the valve and the general requirements delineated for the valve, a portion of the following evaluations will apply to the qualification process.

TABLE 7-1

VALVE OPERABILITY REQUIREMENTS SHEET

VALVE TYPE:________________________ PLANT IDENTIFICATION NO._______

OPERATOR TYPE:_____________________

1. Operation (eg., requirement to close, open, cycle, move to demanded position).

2. When Operation Is Required (eg., immediately after accident, during accident, after accident, during long term recovery from accident).

3. Process Condition During Operation (eg., No flow, normal flow, flow to broken pipe, normal ΔP, high ΔP)

4. Environmental Conditions (eg., In containment during accident, in containment post accident, outside containment)

5. Seat Leakage (eg., No Requirement, Requirement)

6. Stem Leakage (eg., No Requirement, Requirement)

7. Seismic Conditions

8. Special Condition or Requirement (eg., Required to remain open 2 hours)

1. Environmental Load Evaluation
2. Seismic and Other Piping Applied Loads Evaluation
3. Dynamic Loads Evaluation

These are discussed below.

Environmental Evaluation. The portions of the valve which must function, particularly non-metallics, should be identified. This will include springs, sliding parts and such items as gaskets, seals, packings, diaphragms and electrical components. Eliminate those items, such as solenoid valves, previously qualified during the Class IE operability assurance evaluation.

The non-metallic material should be evaluated by comparing the environmental conditions with the materials susceptibility to aging in the form of change in important physical parameters. In general, asbestos gaskets and packings and graphite packings can be shown to be not significantly affected (See Section 4). Environmental qualification of some metallic parts such as springs that affect SRV setpoints and blowdown should also be included.

Environmental evaluation may include:

1. Assurance that preventive maintenance procedures adequately address periodic replacement of lubricating greases and periodic replacement or inspection of continuously compressed or extended springs, which affect operability and setpoint.
2. Surveillance or special tests at safety-related temperature to assure there is no excess setpoint drift.

Seismic and Other Piping Applied Loads. Unlike many other plant components, valves are part of the piping system, that is, they are a special segment of the pipe and supported almost entirely by the piping. As a segment of the piping system, they are subject to the axial, torsional, shear, and moment loads in the piping system. These loads may be static, caused by dead weight and piping thermal expansion, and dynamic, caused by flow, operational transients, and seismic. There is an important difference between how the static and dynamic piping loads occur on the valve. Static loads enter the valve at the flanged or weld valve body/piping interface as forces and moments. These forces and moments act directly on the valve body, but not on the operator. On the other hand, dynamic loads also include piping accelerations at the valve body/piping interface which also results in acceleration in other valve parts and the valve operator.

The codes and practices used in the design of the pressure containing portion (body and cover) of the valve usually give high assurance that piping loads will not distort the valve sufficiently to affect operability. The normal practice is to specify the pressure-temperature rating of the valve. In many cases this practice results in body strength greater than the strength of the adjoining pipe, providing inherent protection against valve damage (94). Simply put, this practice results in the valve body being stronger than the attaching pipe. In this case, stresses in the valve body will remain below yield, assuring no permanent valve body distortion, since the practice in piping design and analysis is to remain below yield stress in the piping. References (95) through (113) provide industry pressure-temperature ratings for valves.

In some cases, the valve application may have the potential for high valve body stresses and distortion under high applied piping forces and moments. The following are examples of possible problems:

- "Venturi" type valves, made by providing enlarged ends on smaller basic valves.
- Cast iron valves installed in steel piping.
- Any "standard" valves installed in "overweight" piping. With the extra wall thickness, the pipe may be stiffer and stronger than the valves.

The nonpressure retaining portion of the valve, namely, the yoke structure and operator are not governed by codes and practices used for the pressure retaining portions. The yoke, operator and other valve extensions are not subject to internal working fluid pressure or forces and moments from the piping. They are, however, subject to seismic accelerations transmitted to the valve body by the piping. The qualification process is one of seismic qualification as discussed in Section 6.

With the loads defined, the evaluation proceeds along several alternate courses: qualification by experience, analysis, or test. Seismic experience can be the basis for qualifying many valves. See Section 6 and Appendix B. In the event that the valve being evaluated is not within the experience data base or limits of application, then analysis or test may be the appropriate method of qualification.

Analysis is performed by first demonstrating that lowest natural frequency of the extended structure is greater than 33 Hz. This is done to assure that the structure will not resonate and amplify the dynamic input. This recognizes that seismic spectra do not have significant energy content above 30 Hz. The equations needed to calculate the lowest natural frequency are simple but do require specific data describing the valve yoke and operator. Alternately, the frequency could be determined by in-situ test which is attractive because it is relatively simple, quick and definitive. These tests amount to a resonant search of the in place valve.

Once it is assured that resonance will not occur, the critical stresses are calculated. This is done by reducing the seismic and other dynamic input to a static "g" value. In many cases, the input will be specified in static g's. However, if it is in the form of a spectrum the ZPA value may be used or if it is a time history, a spectrum and hence a ZPA may be derived. Stresses are to be less than yield and are typically calculated for the yoke, operator attachment and bonnet closure. Bonnet closure stress calculations typically follow applicable ASME valve code procedures with the exception that recommended gasket stresses are not required to be maintained as long as yield in the other members is not exceeded.

Staying below yield assures that permanent deformation does not take place and that deflections are small. Static pull tests can be used to show that these small deflections do not affect operability. These static pull tests often consist of subjecting the valve ends to static loads which simulate the most severe combination of seismic design and piping loads while simultaneously applying static overturning load to the valve topworks which is equal to the seismic acceleration times the mass of the extended valve operator. Under these loading conditions, the valve is cycled open and closed to demonstrate operability.

The refined analysis or test alternatives would normally be used only if the analysis could not demonstrate operability. The simplest test is a static pull test in which a force equal to the static "g" times the mass is applied and the valve operated.

Dynamic Loads Evaluation. Section 5 of this report defines dynamic loads to be transient loads on mechanical equipment that can occur during plant design basis events but are not seismic related, are not due to normal operation and testing, and are not capable of being duplicated in surveillance testing. Section 5 gives guidance for identifying such loads.

Section 5 identified that, for isolation valves, pipe rupture blowdown loads can be a significant form of dynamic load. The transient involves the break, up to and including the double-ended break, of a high energy line downstream of the isolation valve. In many applications, instrumentation is used to detect the break and signal to the valve, through its operator, to close, shutoff the blowdown flow and isolate the leakage path. Closure involves large dynamic forces on the valve gate or disc, as it closes, that must be accommodated by the valve operator and stem. This dynamic load cannot be duplicated in surveillance testing.

The qualification of an isolation valve for pipe rupture blowdown or other flow interruption capability requires a combination of test and analysis. ANSI Standard B16.41-1983 (78) provides guidance on this aspect of qualification.

7.8 CENTRIFUGAL PUMPS

Centrifugal pumps are the most commonly used type of pump at nuclear power plants in safety-related service. Typical applications are shown in Tables 7-2 and 7-3.

Centrifugal pumps similar to those used in nuclear power plants have been used for many years in other industrial applications, including conventional power plants, petrochemical plants, marine vessels and water supply service. The differences between nuclear and industrial applications are:

- The potential for exposure to radiation environments
- Nuclear safety-related pumps are normally in standby mode

Since these centrifugal pumps are important to the safe operation of the nuclear plant, extensive surveillance testing and preventive maintenance programs are performed. As a minimum, surveillance tests are performed every three months to assure that the pump is capable of delivering adequate flow and discharge head. Preventive maintenance programs are based on experience from both industrial and nuclear applications.

Typical areas covered by surveillance and maintenance programs include:

- Pump/driver performance
- Loss of suction pressure
- Excessive vibration or noise
- Excessive leakage
- Lubrication
- Seal cooling

In general, these programs do not provide a rigid time table for dismantling the pump, replacing wearing rings, renewing gaskets/seals or examining internal surfaces for evidence of wear since so much depends on the actual operating conditions. When surveillance testing shows the pump to be operating satisfactorily, it should not be disassembled. Potentially serious or troublesome problems can usually be avoided by systematic maintenance checks.

TABLE 7-2

TYPICAL CENTRIFUGAL PUMP APPLICATIONS (BWR)

SERVICE	TYPE	FLOW RANGE gpm	HEAD, ft	BHP	ESTIMATED MAXIMUM SPECIFIC SPEED
High Pressure Coolant Injection	Horizontal split-case	3000-5000	2800-3000	2500-5000	1000
Reactor Core Isolation Cooling	Horizontal multi-stage	400-800	2800-3000	350-800	500
Low Pressure Coolant Injection	Vertical Single-stage In-line	4800-10000	356-600	700-1800	2500
Residual Heat Removal	1)Vertical in-line single stage 2)Vertical, multi-stage	4800-10000	350-600	700-1800	2500
High Pressure Core Spray*	Vertical multi-stage	3600-7000	2800-3000	3000-7000	1000
Low Pressure Core Spray	1)Vertical in-line single stage 2)vertical multi-stage	3000-6200	650-750	650-1300	2000
Service Water*	1) Single stage turbine type 2) Two-stage turbine type	6000-9000	150-200	400-600	4000

*These pumps are typically oil lubricated: HPCI and RCIC use turbine-driven system: HPCS uses auxiliary lube oil pumps, SW uses splash lubrication.

TABLE 7-3

TYPICAL CENTRIFUGAL PUMP APPLICATIONS (PWR)

SERVICE	TYPE	FLOW RANGE gpm	HEAD, ft	BHP	ESTIMATED MAXIMUM SPECIFIC SPEED
Containment Spray	1) Vertical single-stage diffuser type 2) Horizontal	1200-4000	300-600	120-500	3000
Low Head Safety Injection	1) Vertical Single-stage 2) Vertical Multi-stage	2000-4500	200-600	400-500	4000
High Head Safety Injection (Charging Pump)*	1) Multi-stage 2) Horizontal Multi-stage	300-800	1900-3000	400-1000	500
Auxiliary Feedwater*	1) Horizontal Multi-stage 2) Horizontally-split single suction	300-900	2500-3500	350-800	500
Service Water*	1) Horizontally-split double suction 2) Vertical multi-stage	5000-22000	150-200	350-800	5000
Residual Heat Removal	1)Vertical in-line single-stage 2)Horizontal single-stage end suction overhung-impeller with vertically split case	2000-3000	280 375	200-300	3000

* These pumps are typically oil lubricated: HHSI and AFW use auxiliary lube oil pumps; SW uses splash lubrication.

General Description

Centrifugal pumps have only one moving part, the rotor element. They have large clearances, broad tolerances and withstand high temperature, erosion and corrosion relatively well.

The rotor element consists of the pump shaft and impellers. Impellers may be closed, open or semiclosed (partial-shroud) types. Closed impellers are most commonly used in nuclear safety related applications. They have the advantage of highest efficiency and lowest net positive suction head (NPSH) requirement. Open impellers are commonly chosen when scaling or depositing is prevalent since they are relatively self-cleaning. Semi-closed impellers are most often used for suspended solids, which can be broken up by the entrance vanes.

Seals may be fixed (packing) or rotating (mechanical). Both generally require lubrication and sometimes cooling. Fixed seals are lower in initial cost and are often selected for services where leakage is tolerable, for gritty services, start-up service and intermittent pumping. However, mechanical seals are now usually specified whenever possible for continuous duty, clean liquid service because of lower maintenance and lower leakage. Stuffing boxes are normally designed to carry either fixed or rotating seals. A typical mechanical seal for low pressure service is shown in Figure 7-11. For high pressure service, a bellows-type seal is usually not used. Single inside mechanical seals are normally standard for clean liquids. A single, outside, externally flushed mechanical seal is used for high temperatures. A compatible flush fluid is necessary. Double mechanical seals are used for critical, toxic or nonlubricating liquid services. Sometimes a pumping ring on the mechanical seal is provided to circulate the seal fluid through an external heat exchanger and back into the seal cavity.

Wearing rings are provided in pumps at the impeller inlet and at roughly the same diameter on the backface of the impeller hub. They minimize leakage from the high pressure side of the impeller back to the inlet and thereby improve overall pump efficiency. Their diametral position also affects axial thrust loads on the shaft. Design details of wearing rings vary with manufacturer and type of pump.

However, all are constructed with appropriate materials to minimize galling in the event of a rub and to minimize material loss due to erosion and corrosion. Clearances are chosen to minimize leakage, while at the same time they must be large enough to accommodate bearing clearances, shaft deflection, and misalignment due to assembly tolerances and casing distortion. For single stage pumps, typical diametral clearances for wearing rings are shown in Table 7-4.

TABLE 7-4

SINGLE-STAGE PUMP WEARING RING CLEARANCES
(Reference 25)

LEAKAGE JOINT DIAMETER (in.)	TOLERANCE (in.)	ACTUAL DIAMETRAL CLEARANCE (in.)
4	0.002	0.012 - 0.016
6	0.002	0.014 - 0.018
8	0.002	0.016 - 0.020
12	0.003	0.018 - 0.024
20	0.003	0.020 - 0.026
30	0.003	0.022 - 0.028
48	0.004	0.026 - 0.034
80	0.005	0.030 - 0.040

The actual diametral clearance between the impeller and casing, as shown above, is determined on the basis of typical clearances and tolerances on the casing ring inner diameter and impeller outer diameter. For other leakage joint diameters, the actual diametral clearance should be taken as the same as for the next higher joint diameter.

The lower value of the actual diametral clearance shown in Table 7-4 is the basic diameter clearance for single stage pumps. For multistage pumps, the basic diameter clearance should be increased by 0.003" for large rings (25). These wearing ring clearances are intended to serve as a guide for seismic and dynamic qualification unless specific information is available from the pump manufacturer.

Bearings commonly used in pumps include sleeve-type bearings, ball bearings, roller bearings and Kingsbury-type thrust bearings. Bearing manufacturers standards include information on the life expectancy, load limits, design and bearing materials selection. Based on this research, quite accurate methods are available for selecting the right bearing for any application. For nuclear safety related pump service, seismic loads must be included with the other loads that are used in selecting adequate bearings. Sleeve-type or journal bearings are lubricated by the process fluid. Other types are either grease- or oil-lubricated. Oil-lubricated bearings often are water-cooled to maintain acceptable lubrication temperatures.

Types

Most nuclear plant services are satisfactorily handled by end suction horizontal and vertical in-line pumps. However, for large capacity applications such as some Residual Heat Removal Systems, the vertical multi-stage pump is often used. For high pressure service, the horizontal multi-stage pump is often used. Tables 7-2 and 7-3 present a summary of the types of centrifugal pumps which are used in safety related applications in nuclear power plants.

A typical vertical in-line pump is shown in Figure 7-12. Their basic design has been used in years of commercial service. Basically, the vertical in-line pump is a close-coupled pump, designed to satisfy continuous heavy duty service criteria. The vertical in-line pump is installed in a piping system much like a valve, except that it may have a support placed under the casing to carry the dead weight of the unit. The pump shown in Figure 7-12 has bearings only in the motor. The impeller is mounted directly on an extended motor shaft. The principal advantage of this arrangement is positive alignment of pump and driver, which is accomplished with precision machined rabbet fits on the motor support. This isolates the running clearances between the rotor and stator elements from the influence of external piping loads and transient temperature variations in the pumped fluid.

Some vertical in-line pumps have bearings in the pump frame. These bearings reduce the requirements for precision machining discussed above and compensate for minor misalignments.

A typical vertical deep draft pump is shown in Figure 7-13. As designed for use in nuclear power plant service, these pumps are a modification of pumps conventionally used to lift water in well service. A horizontally mounted suction nozzle is provided to allow water to enter the suction barrel before it is pumped upward through a series of impellers. The suction barrel assures that adequate net positive suction head (NPSH) is available at the first stage inlet when low available NPSH conditions exist.

A typical horizontal end-suction pump is shown in Figure 7-14. This pump has an overhung impeller with oil lubricated bearings in the pump frame. This pump has a shrouded (closed) impeller with wearing rings at the impeller inlet and outlet. Typical impeller discharge diameter is in the range of 12" to 20".

A typical horizontal multi-stage pump is shown in Figure 7-15. Pumps of this type have been used for many years in main feedwater pumping applications. In nuclear safety related service these pumps are used in such high pressure applications as HPCI, RCIC, Auxiliary Feedwater and Safety Injection. The balancing of axial hydraulic thrust is one of the key considerations in the design of these pumps. Because some slight unbalance may exist, even at rated head-flow conditions, these pumps are always provided with thrust bearings.

Typical Service Conditions

Active safety-related pumps are located outside containment, usually in the Reactor Building (BWR) or Auxiliary Building (PWR). These pumps provide reactor coolant recirculation following a postulated loss of coolant accident (LOCA).

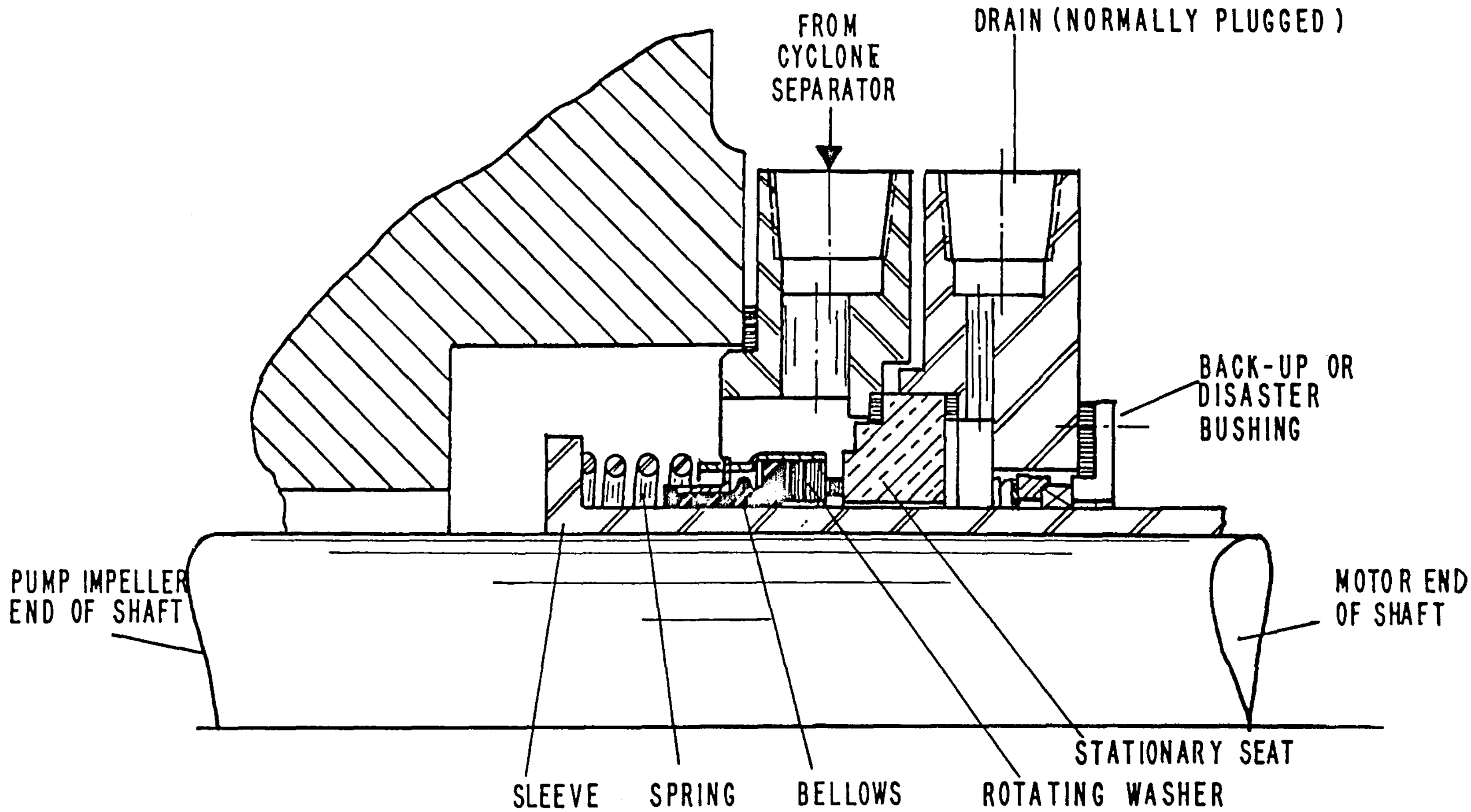


FIGURE 7-11: CROSS SECTION OF MECHANICAL SHAFT SEAL
USING COOLANT FROM PUMP DISCHARGE

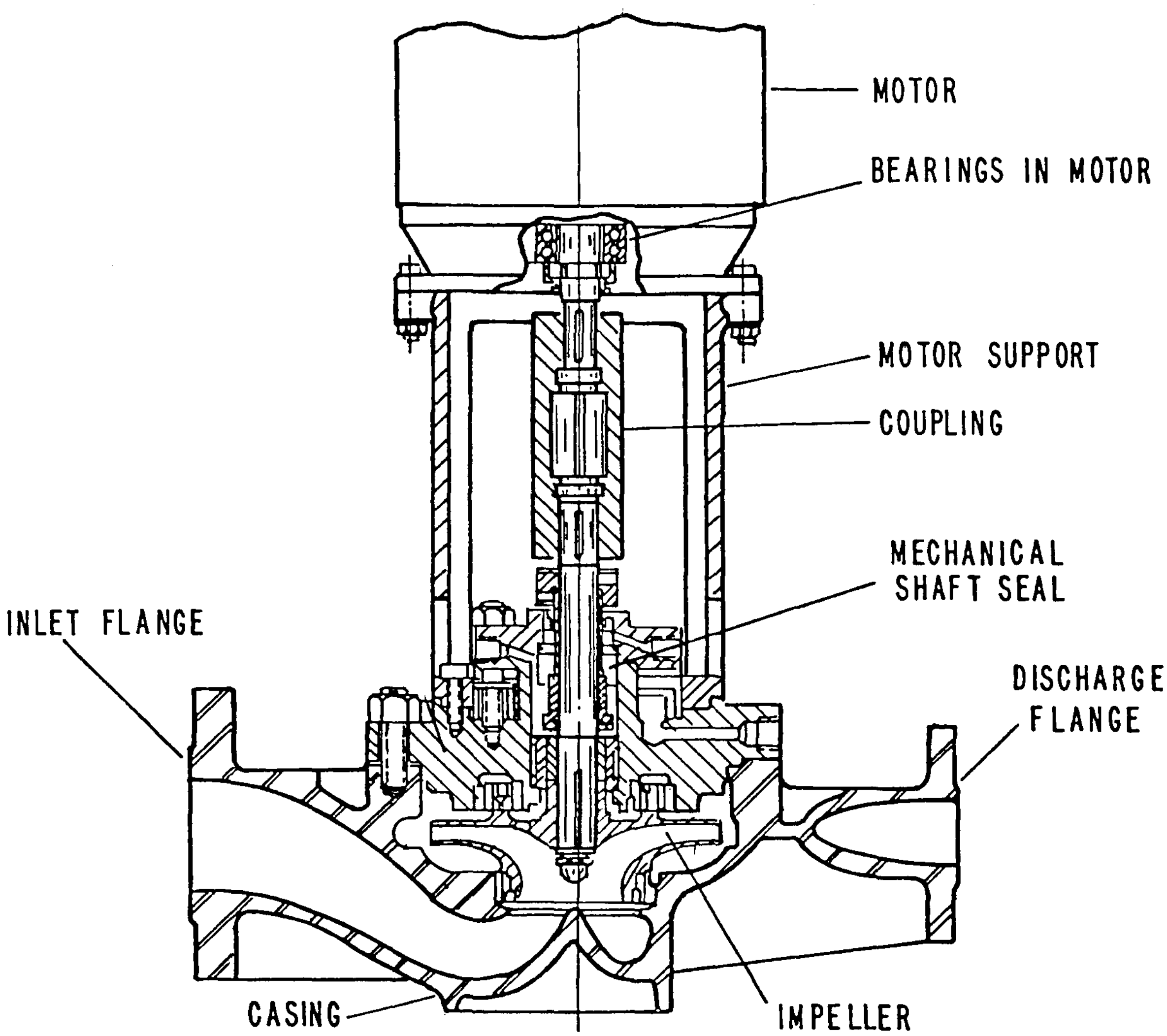


FIGURE 7-12: TYPICAL VERTICAL IN-LINE PUMP

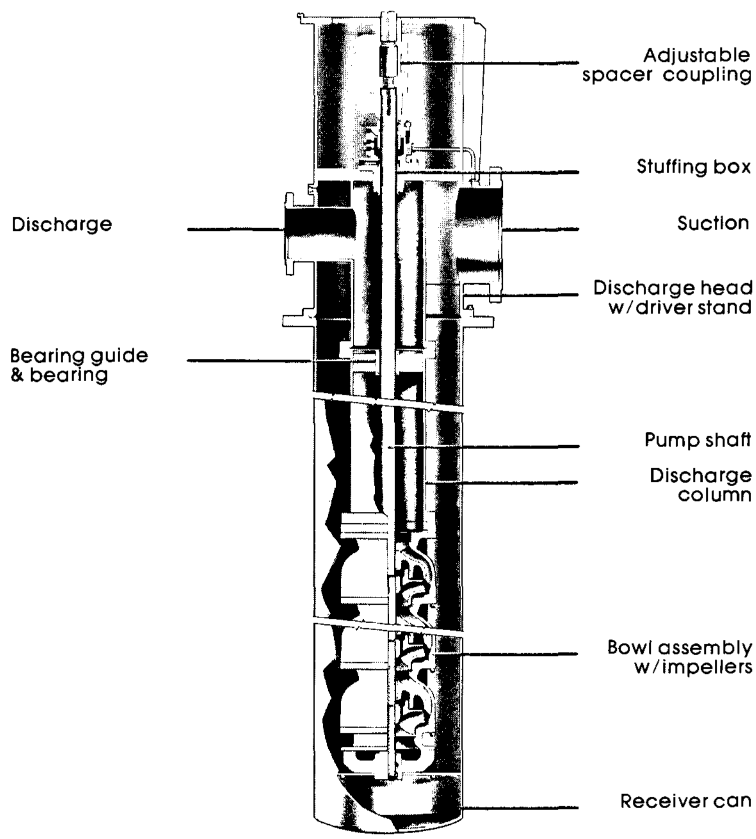


FIGURE 7-13: TYPICAL VERTICAL DEEP DRAFT PUMP

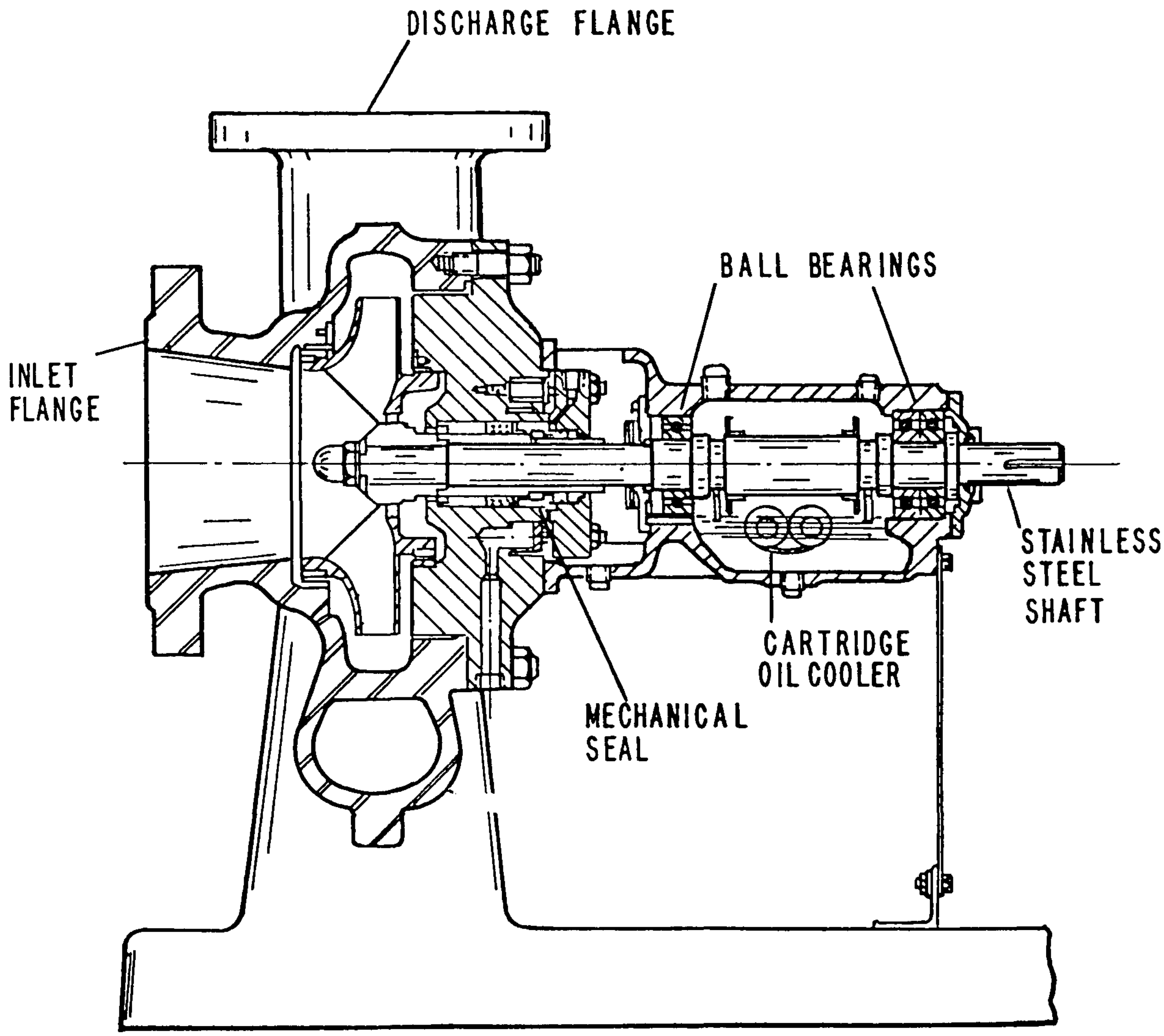
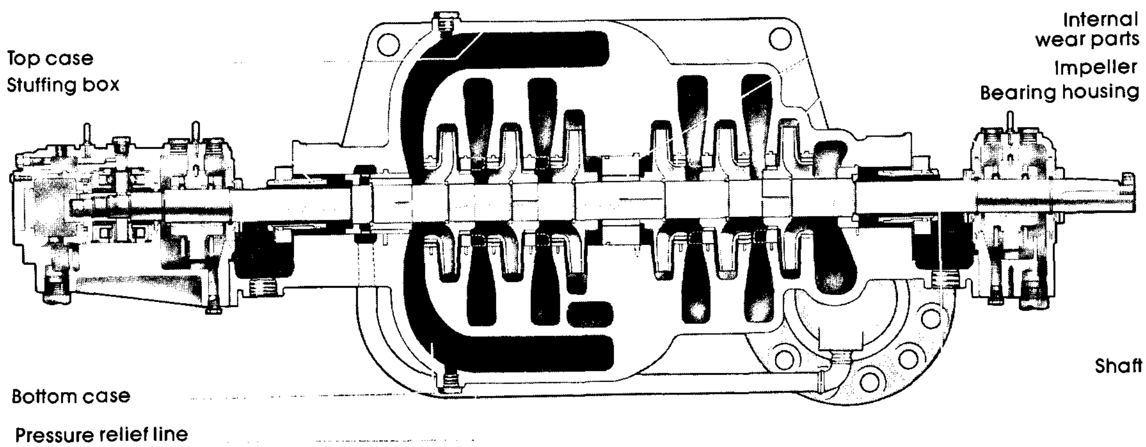


FIGURE 7-14: TYPICAL HORIZONTAL END-SUCTION PUMP



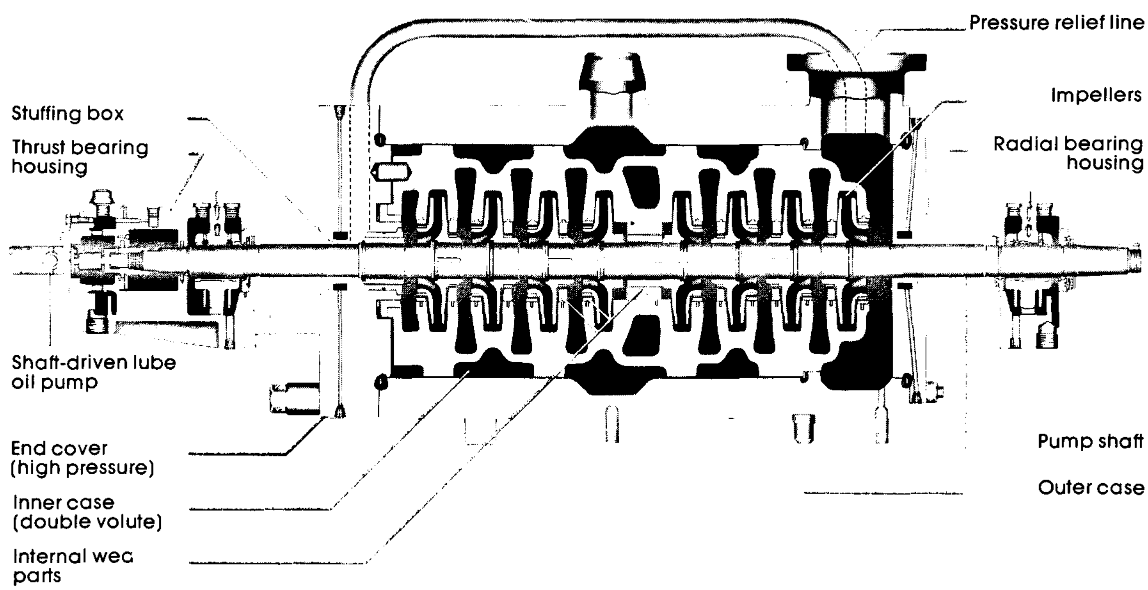


FIGURE 7-15: TYPICAL HORIZONTAL MULTISTAGE PUMP

Typical environmental conditions are listed below:

	BWR	PWR
Temperature		
Normal	85-105°F (30-40°C)	65-130°F
Accident	148°F (65°C)	132°F
Pressure		
Normal	-0.4 in W.G.	Atmospheric
Accident	Atmospheric	Atmospheric
Relative Humidity		
Normal	20-90%	20-80%
Accident	90%	80%
Radiation		
Normal	$3x10^5$R	$1x10^6$R
Accident	$1x10^7$R	$1x10^7$R

Most of the larger active safety-related pumps are located at or below grade elevation. Therefore, the seismic excitation at these pumps is the same as the site ground acceleration spectra. At other locations appropriate building floor response spectra should be used. The seismic information presented in Table 7-5 is derived from plants with an SSE horizontal ground motion of 0.2 g.

TABLE 7-5

SEISMIC ACCELERATIONS FOR ACTIVE SAFETY-RELATED PUMPS

LOCATION	DIRECTION	ZPA(f>=33Hz)	PEAK(β =2%)
Basemat	Horizontal	0.30 g	1.2 g
	Vertical	.24 g	0.8 g
Other Locations	Horizontal	0.50 g	1.8 g
	Vertical	.30 g	1.2 g

Failure Modes

The failure modes that can lead directly to failure of a pump to perform its active safety related function are summarized in Table 7-6. Also included are the most probable causes of failure, loading conditions, and the appropriate qualification methods to eliminate systematic deficiencies in design or operation.

TABLE 7-6

APPROPRIATE QUALIFICATION METHODS FOR PUMPS

FAILURE MODE/ PROBABLE CAUSE	LOADING CONDITION	APPROPRIATE QUALIFICATION METHOD
<u>Rotating Element Seizure/Drag</u>		
Bearing Overload	Seismic	Seismic Experience Analysis
Lubrication	Normal operational environment Radiation $>10^7$R	Surveillance Maintenance Materials Review
Wear Rings	Seismic	Seismic Experience Analysis Materials Review
Shaft Deflection	Dynamic Seismic	Seismic Experience Analysis Surveillance Materials Review
Differential Expansion	Accident induced Thermal transient	Analysis Test
<u>Loss of Rated Flow/Head</u>		
Wear Ring Wear	Normal operational environment	Surveillance
Impeller Wear	Normal operational environment	Surveillance
Gas Entrainment	Improper system venting	Surveillance
Seal/Gasket Leakage (excessive)	Normal operational environment Radiation $>10^5$R	Surveillance Maintenance Materials Review

The failure modes are grouped under two main headings:

Rotor Element Seizure/Drag
Loss of Rated Flow/Head

Rotor element drag can become severe enough to result in excessive slowdown, stall or trip of its prime mover. Loss of rated flow/head is the result of a combination of excessive internal bypass and external leakage or may result from excess unvented gas in the pump casing.

The primary contributor to rotor element seizure is the misalignment of rotating parts. Normal wear of rotating and stationary parts is the primary contributor to loss of rated flow/head. Other probable causes of failure modes are shown in Table 7-6.

Pump seal degradation does not necessarily lead to an active functional failure of the pump. Seal leakage is significant for pump qualification only if:

- Pump performance becomes inadequate
- Water leakage affects
  - -- pump lubrication (water in oil)
  - -- other pump internals (impingement)
  - -- adjacent safety-related equipment

Typical configurations, as shown in Figures 7-12 through 7-15, can be used to prevent process water leaks from mixing with the lubricating oil. In vertical pumps lubricated bearings are located above the process flow path (sometimes in the motor) where they may be protected from leakage of process water. In the case of a typical horizontal end-suction pump, a gasket leak would be in the radial direction away from the axis of the pump shaft, so that the leak would not flow to the oil lubricated bearings. Leakage along the shaft through the mechanical seal is deflected by the bearing splash protector. For multistage horizontal pumps the bearings are installed on the shaft horizontally outboard of the shaft seals.

Other potential sources of water leakage into pump lubrication systems include:

- Steam/water mixtures from turbine drivers
- Service water used for oil cooling
- Condensation of atmospheric moisture

If moisture from these sources could adversely affect pump performance, its presence can best be confirmed during routine surveillance and maintenance. If water in lubricating oil is known to be a recurring factor in plant Licensee Event Reports (LER), a thorough investigation should be undertaken to ascertain the root causes and determine corrective actions.

Appropriate Qualification Methods

Appropriate qualification methods for pumps, as summarized in Table 7-6, include:

- Seismic Experience
- Analysis
- Surveillance
- Maintenance
- Materials Review
- Test

Guidance in the use of these methods, as detailed below, is also the subject of a number of parallel industry efforts, most notably:

- On-going nuclear industry standards activities conducted under the auspices of ANSI and ASME
- Seismic experience reviews conducted by SQUG and confirmed by SSRAP

Most of the ASME standards are in the proposal stage with a few available in draft form, such as:

QNP-1	Standard for Qualification of Pump Assemblies for Safety Systems Service - General Requirements
QNP-2	Standard for Qualification of Pumps in Pump Assemblies for Safety Systems Service
QNP-3	Standard for Qualification of Shaft Seal Assemblies in Pump Assemblies for Safety Systems Service

Two features of QNP draft standard approach are of particular importance to the qualification of pumps in operating nuclear power plants:

1. Qualification by similarity is allowed
2. Maintenance and surveillance are recognized as necessary for on-going assurance of qualification.

The similarity of a "candidate" pump assembly to a "parent" pump assembly - to use the terminology employed in valve qualification - requires the following elements:

- "Parent" qualified to QNPE-1
- Justification of similarity using

  -- Analysis
  -- Supplementary Tests, or
  -- Evidence of operating experience

The Senior Seismic Review and Advisory Panel (SSRAP) was formed in 1983 to review and comment on the study performed on behalf of the Seismic Qualification Utility Group (SQUG), which uses past earthquake experience data of non-nuclear facilities to demonstrate seismic ruggedness. The SSRAP findings for centrifugal pumps are that horizontal pumps in their entirety and vertical pumps above their flange are relatively stiff and very rugged devices due to their inherent design and operating requirements. This conclusion emphasizes the justification of similarity using evidence of operating experience of equipment subjected to earthquakes which is allowed by QNP standards.

The use of other forms of operating experience with pumps to justify similarity should consider the significant role that the concept of specific speed has played in the utilization of test and design data on existing pumps to develop new designs of dimensionally similar pumps but of larger or smaller size because the specific speed remains independent of size. Tables 7-2 and 7-3 list the estimated maximum specific speed for typical pumps used in safety-related service in nuclear power plants. Specific speed is an index number which can be used to classify the general outline of impeller profiles, the typical performance characteristics of centrifugal pumps and the range of efficiencies attainable with each type of centrifugal pump. Further discussion and details can be found in Reference (25).

Routine surveillance and maintenance (S&M) is the appropriate qualification method to address many of the potential failure modes for pumps. Surveillance testing confirms the head/flow characteristics (acceptable internal leakage) and proper alignment/clearances for rotating parts. Excessive noise or vibration during surveillance testing may require disassembly to determine corrective measures. It is noted that the unbalanced rotor loads in a centrifugal pump which could occur during a low flow surveillance test can be significant compared with the unbalanced rotor loads which are minimized by design for the design flow conditions. The surveillance test conditions will, therefore, be at least as severe for consideration of wear as those which are postulated for an accident or earthquake. Correct alignment is properly addressed in maintenance activities during pump reassembly.

The metallic and non-metallic components of nuclear safety related pumps are subjected to temperature, pressure and humidity conditions which are common to many industrial applications. The temperature of the fluids being pumped in safety-related pumps is usually 375^{O}F or less, a mild condition compared to other applications in fossil power, petrochemical and the process industries.

The most common fluid pumped in nuclear safety related service is non-radioactive water. Hence, the pumps are often not exposed to high radiation doses from the process fluids. The notable exceptions are the emergency core cooling system (ECCS) pumps which recirculate radioactive water in a containment recirculation mode following a loss of coolant accident (LOCA). The typical integrated dose for these ECCS pumps is on the order of 10^7R. Environmental qualification for these pumps, in addition to normal surveillance and maintenance, is a design review of non-metallic materials to resist exposure to the calculated total integrated radiation dose. The non-metallic materials are contained in bearings, gaskets and seals. Details of the environmental qualification process are given in Section 4.

Metallic parts which are critical to pump operability include the rotor element (shaft and impellers), mechanical shaft seals, shaft supporting bearings and wear rings. The primary objective of operability evaluations is to verify that the minimum running clearances provided within the pump are adequate to avoid seizure of the rotor element when the pump is subjected to:

- Hydrodynamic loads
- Pipe connection loads
- Seismic loads
- Transient temperature modes, if significant

One type of hydrodynamic load is BWR LOCA-related blowdown loads, which are transmitted to the plant basemat or foundation that is common with the suppression pool. These hydrodynamic loads may be applicable to nuclear safety-related pumps depending on their location within the plant. Other hydrodynamic loads, either for specific BWR or PWR designs, may require consideration for dynamic qualification of nuclear safety related pumps. Further guidance in this area is given in Section 5. In general, however, the hydrodynamic loads which have been identified for nuclear safety related pumps are comparable with those handled by pumps in fossil power, petrochemical and industrial process applications. All pumps, regardless of application, are subjected to internal pulsations caused by hydraulic interaction between the pump impeller and casing. Performance testing and routine surveillance testing of pumps should verify the adequacy of the basic hydraulic design.

Pipe connection loads are usually analyzed as part of the pressure boundary calculations of the pump casing and nozzle. Pump configurations with forged or cast casings generally can withstand these pipe connection loads because the pump casing and nozzle wall thickness are greater than the pipe thickness. Some pumps, such as the vertical multi-stage units, have a cylindrical shell for a casing which may be more susceptible to deformation from pipe connection loads. Pipe connection loads should be reviewed to ensure that they will not adversely affect pump operability. Analysis is the appropriate qualification method for evaluation of pipe connection loads. The basic approach in dealing with excessive pipe connection loads is to further restrain the connecting piping (to reduce the loads) or to add structural support to the pump nozzle.

Seismic loads are considered in the design evaluation of all safety-related pumps. The operability considerations which typically are addressed analytically are the shaft stress; the deflections at the shaft seal, at the impeller wear ring areas and at the shaft supporting bearings; and the bearing loads. The seismic qualification methodology is to perform a static or dynamic analysis to determine the resulting stresses, bearing loads and deflections when the pump is subjected to externally applied loads. Since the ASME code only governs the allowable stresses in pressure boundary parts, the allowable stresses for non-pressure boundary parts and their relationship to safety factors are established by sound engineering practice and manufacturer's guidelines. Momentary bearing loads for seismic, which are higher than the bearing manufacturer's continuous load rating, are acceptable (See Appendix A). If the deflection calculation shows a momentary loss of running clearance and therefore a rub between metallic parts, a materials review should be conducted to assure that the combination of materials is not susceptible to galling. The pump manufacturer should be consulted for guidance in this area.

In some safety-related applications, pumps may be subjected to significant transient temperature cycles when they are required to perform a safety-related function. These transient temperature cycles will not exist during surveillance testing. Therefore, a transient thermal analysis should be performed to verify that running clearances are maintained throughout the duration of the transient temperature cycles and that non-metallics would survive thermal loading without impairing operability. In some cases it may be necessary to supplement the thermal analysis with shop testing to determine the effect of transient temperature modes on pump running clearances. During such shop tests, the pump is operated when it is subjected to a series of transient temperature cycles which simulate the expected accident-induced thermal cycles. An example of transient temperature cycle tests is discussed in Reference (67).

In addition to surveillance and thermal cycling tests, in-situ tests can be employed to supplement the qualification of pumps. In-situ testing may be employed to:

- Identify structural resonances and damping for use in seismic analysis
- Establish similarity between pumps (so as to qualify one pump by work done on another pump or class of pumps)

In-situ testing may demonstrate that a pump is rigid and can be qualified by static analysis methods. This is generally the case for horizontal pumps. However, for vertical pumps, flexibility of the rotor shaft is often desirable and must be accounted for in the seismic analysis. The placement of transducers is generally limited to the exterior of the pump casing and motor housing, particularly for pumps which are already installed. Because of this restriction, the dynamic characteristics of installed vertical pumps are usually established on the basis of similarity with another pump which has already been in-situ tested and seismically analyzed.

7.9 DIESEL ENGINES

Diesel engines are used at nuclear power plants primarily as drivers for emergency electrical generators. In addition some diesel engines are used as drivers for pumps, such as diesel driven fire pumps. The diesels used in nuclear plants generally fall in the capacity range of 2000 KW to 9000 KW.

Diesel engines similar to those used in nuclear plants have been used for many years in other applications, e.g., hospital and airport emergency power supplies, marine propulsion, military applications, pumping stations, utility base load and peaking stations and locomotives. Frequently, the testing and operating conditions for these non-nuclear applications are more harsh than for the nuclear applications. Examples of this are the weekly quick start/load testing of hospital units, shock testing of military units, pitch and roll in marine applications, and shock loads in dredging or dragline service. Historically, the performance of diesel engines which have been subjected to seismic conditions has been good (31, 39).

In nuclear applications, diesel engines are normally located in individual compartments away from other equipment and piping (e.g., in the auxiliary or turbine building). The environmental conditions are mild in these compartments in both normal operating and accident situations.

The diesel engine consists of a massive engine structure and internals along with a wide assortment of auxiliary pumps, valves, filters, and heat exchangers. This auxiliary equipment provides the diesel engine with starting power, fuel oil, lubrication, combustion air intake and exhaust removal, cooling and speed control. The engine structure and internals are designed for the dynamic loads incurred during startup and normal operation. Also, the engine and auxiliaries are designed for the engine vibrations which normally occur during startup and operation.

Since these diesel engines are important to the safe operation of the plant, extensive surveillance testing and preventive maintenance programs are performed. As a minimum, surveillance tests are performed once each month and simulate an emergency by quick starting the engine, synchronizing the generator to the grid and then rapidly loading the diesel generator. In addition, during refueling outages the diesel-generator assembly is put through a cold, quick start and load test to demonstrate its ability to provide power in an emergency.

In most non-nuclear applications of diesel engines, maintenance schedules are based primarily on hours of engine operation. However, for nuclear plants, the preventive maintenance programs applied to diesel engines are based primarily on elapsed time due to the use of these engines in stand-by service. These programs address inspection and maintenance of the engine and auxiliary systems components. Typical areas covered by these programs are: inspection of lube oil, cooling water and fuel systems for leaks, replacement or cleaning of filters and strainers, inspection of air start system, inspection of auxiliary pumps, lubrication of linkage, and inspection of engine internals.

In general, extensive teardowns of these engines are not performed unless a problem occurs. Experience in the nuclear and other industries has shown that reliability is best maintained by not disturbing equipment which is operating properly unless a good reason exists for suspecting incipient failure or excessive wear of specific components.

As a result of the mild environment, environmental degradation of diesel engine components is not a concern if an adequate maintenance and surveillance test program is followed.

In most cases the operating design conditions have resulted in a diesel engine and auxiliaries which will withstand the plant design earthquake without failure. However, some components may require further examination, testing or analysis to demonstrate full qualification for seismic loading. One area which deserves particular attention is the engine support system. Some engines have been mounted on isolation pads or support frames which result in a dynamic response frequency of the engine which is within the seismic range of less than 33 Hz. If the engine support system results in this type of response, modifications to the supports or analysis of the engine for these response conditions will be necessary.

A description of a typical diesel engine and its operation are given below. Although the diesel engines vary between two-cycle and four-cycle, in-line and vee, and opposed piston or piston/cylinder head arrangements, the basic description of the engine, auxiliaries and operation applies generally to all types.

A typical 2500 KW engine has 20 cylinders with a bore and stroke of nine inches (229 mm) and ten inches (254 mm), respectively, operates at 900 rpm and is turbocharged. During standby the engine coolant and lubricant are heated with immersion heaters and circulated continuously to maintain the bearings ready for the quick start and loading which is a requirement for these engines.

One or more systems are sometimes used for pumping, cooling and cleaning the lube oil when two systems are used. A scavenging oil system takes oil from the engine sump and pumps it through the main oil filter and oil cooler to a holding tank. The engine lube oil pump then takes suction from this holding tank and pumps pressurized oil to the engine bearings, turbocharger, governor and other components. On some engines additional lube oil pumps may be used to circulate oil during standby or for other applications.

The fuel pump takes fuel from the day tank and pumps it through the fuel filter to the fuel injectors. The day tank is filled by a fuel transfer pump from the main fuel tank.

The engine cooling system consists of one or two pumps which pump water through the turbocharger aftercooler and then through the engine cooling passages. The engine heat is removed from the coolant either by a radiator or heat exchanger. The remainder of the cooling system consists of a water tank, instrumentation and related piping. Some engines also use the coolant to remove heat from the lube oil.

Engine starting is accomplished by multivane air motors which drive the engine ring gear. (Some diesels use air delivered directly to the cylinders for starting. In this case an air distributor is used to control the air flow to each cylinder.) The compressed starting air is held in an accumulator near the engine. When a start signal is received a solenoid valve is energized allowing air to flow from the accumulator. The air flows to the air motors and engages the pinion gears with the engine ring gear. Once the pinion gears are engaged, the air pressure opens the main air start valve. This allows air to flow through the strainer and air line lubricator to drive the air motors. In other cases, the air flows, through a distributor valve, sequentially to some of the engine cylinders to turn over the engine.

The air accumulator is maintained within an acceptable pressure range by an air regulator and compressor. Since the accumulator contains sufficient air to allow several engine starts, the compressor is not a safety related component.

The lube oil, fuel, cooling and starting systems described above are the active auxiliary systems which must be considered in the qualification of diesel engines. Except for the components identified as not safety related, all of these systems must function properly for diesel engine operation.

In addition to these systems, diesel engine operation is dependent on the service water system. The service water system is needed to provide cooling water to the engine oil cooler and to the engine cooling water heat exchanger, unless a radiator is used.

Engine speed is controlled by a governor. Correct governor operation is critical to engine performance, particularly during generator loading and load shedding. Governor malfunction could result in an engine overspeed trip or too large of a voltage and frequency drop during loading.

The engine protective devices and the associated electrical systems and components are also necessary for proper unit operation. These electrical devices are addressed in electrical qualification programs.

Specific non-metallic materials which are used in diesel engines and typical applications of these materials are listed in Table 7-7. Each of these materials must be shown to be acceptable for the environmental conditions listed in the following section.

TABLE 7-7

DIESEL ENGINE MATERIALS

MATERIAL	TYPICAL APPLICATION
Asbestos/Paper with elastomeric binder	Piping gaskets, Cover gaskets
Asbestos with metallic binder or retainer	Cylinder head gaskets Exhaust pipe gaskets
Rubber elastomers	O-rings, cylinder liner seals, component seals, shafting and cylinder seals, control device diaphragms and seals

Environmental and Seismic Conditions

The diesel engines are located outside of containment in separate compartments. The diesel compartments are normally free of high energy lines and away from sources of radioactivity. The environmental conditions for typical PWR and BWR plants are slightly different, but the enveloping conditions for PWR and BWR, normal and accident are as follows:

Temperature	65 to 132°F
Pressure	-0.125 to 0.5 in. W.G.
Relative Humidity	10 to 100%
Radiation	10^4 R

Due to the location of the diesel engines in dedicated compartments away from sources of radioactivity, the impact of accidents on the diesel engine environment is insignificant. Because of this, any degradation of components resulting from these environmental conditions would be identified and repaired during the normal surveillance testing and preventive maintenance.

The diesel engine compartments are located at or below grade elevation. Therefore, the seismic excitation at the diesel approximates the site ground acceleration spectra. For most nuclear plants in the United States this means vertical and horizontal zero period accelerations of 0.3 g or less with higher accelerations for components in the frequency range of 2 to 33 Hz.

The stresses resulting from these accelerations and resultant forces are insignificant in comparison to the normal operating loads on the main internal engine components. However, seismic loads are significant and must be considered for some of the auxiliary components. This is addressed in a later section.

Failure Modes

Because of the complexity of the diesel engine and its support systems and components, a wide array of failure modes can be postulated. Some of the more likely failure modes are listed in Table 7-8. The failure modes are listed in two categories: engine failures and auxiliary failures. Electrical failures are not addressed here.

The failures listed are the primary non-electrical failures which would prevent the diesel engine from providing its safety function. In an emergency situation some of the failure modes listed (low oil pressure, loss of engine cooling, and hose and tubing leaks) would not immediately shut down the engine. However, after a short duration these modes could lead to other conditions (i.e., bearing failure or fuel or lubricant fire) which would interrupt engine operation.

The primary function of the diesel engine surveillance testing and preventive maintenance programs is to identify operational problem areas and to prevent minor problems from developing into failures.

TABLE 7-8

DIESEL ENGINE FAILURE MODES

TYPE OF FAILURE	POSSIBLE CAUSE
<u>Engine Failure</u>	
Wiped bearings	Inadequate lubrication, Air or dirt in lubricant
Turbocharger failures	Inadequate lubrication Oil leakage Fire Overheating Impeller failure
Fuel injection failures	Binding or corroded linkage Clogged injectors Water entrainment Degraded fuel oil
Piston Seizure	Liquid leakage into cylinder
<u>Auxiliary Failures</u>	
Low Oil Pressure	Plugged filter Pump wear Pump bearing clearance Water in oil Oil sloshing
Governor failures	Low oil level Damaged linkage
Loss of engine cooling	Valve failures Pump failure Pump air inleakage Cooler fouling
Hose, tube & pipe leaks	Loose Fittings (Vibration) Cracks in components Gasket failure
Inadequate starting air	Obstruction in air line Plugged air filter Air start valve or distributor failure
Air start motor failure	Broken vanes Dirt or moisture in air

Most of the failure modes listed in Table 7-8 can be addressed by thorough inspection and surveillance testing in conjunction with a good preventive maintenance program. However, others may require additional engine or component qualification.

Qualification Methodology and Criteria

The metallic and non-metallic components of the diesel engine and its auxiliaries are not subject to any unusual temperature, pressure, humidity or radiation conditions because of their use in a nuclear power plant environment since the diesel engines are located away from sources of radioactivity and away from high energy pipe lines. Surveillance testing is an adequate method of on-going environmental qualification of diesel engines. This testing and inspection will identify component degradation due to material aging.

Dynamic loads on a diesel engine are associated with engine operation and are not unique or of greater magnitude because of the application in a nuclear power plant. Diesel engine dynamic loads are exclusively associated with the dynamics of a reciprocating engine.

Diesel engines are designed with consideration of vibration, fatigue and wear; and the surveillance testing, inspection and preventive maintenance imposed on the nuclear plant diesels is intended to identify excessive wear or vibration damage. Although fatigue damage is not usually identifiable prior to component failure, it is not a concern in nuclear applications of diesels. This is due to the short total operating time of these engines. Diesel engines of this type are designed for long term operation at heavy loads. Typically, diesel engines at nuclear plants will operate less than 100 hours per year. For a 40 year life this is a total of only 4000 hours of actual operating time. Diesel engines of this size have a design life of approximately 100,000 operating hours. In summary, dynamic load qualification for engine vibration is most appropriately performed by surveillance testing.

Seismic qualification of diesel engines involves evaluation of the engine structure and internals, engine auxiliary components, and anchoring systems. Each of these is addressed below.

The basic engine structure is a massive casting or fabricated structure that is designed to be rigid in order to withstand the engine operating forces without permitting harmful deflections. The engine internals, such as the crank shaft, connecting rods, pistons, bearings and camshaft, are all designed for the operating loads with limiting torsional or linear deflections and limiting stresses considering fatigue over the engine lifetime. Since the diesel engine structure and internals are rigid, and the engines are located at grade elevation, the seismic loads will be low. In comparison with the operating loads these seismic loads are insignificant and the engine structure and internals are qualified for the seismic loads. One possible region which may not have received adequate attention is the portion of the engine structure in the region of the engine to building or skid anchors. This will be addressed later in the discussion of anchoring.

Auxiliary components, such as lube oil pumps, air starters and piping located both on and off the engine, are generally designed to be rigid. On older engines, seismic design analysis of these components has generally consisted of static analysis of the supports and bolts assuming that the components and supports are rigid. Recently, some impedance testing of auxiliary components has been performed to establish response frequencies. Although most auxiliary components have been found to be rigid, some modifications to supports have been required to increase response frequencies to acceptable levels. For older engines, this may mean that some modifications to auxiliary component supports will be necessary. In-situ testing can be used to establish the response frequencies of the auxiliaries.

During initial engine assembly design, anchor bolts from the engine to the skid/building are usually analyzed statically assuming a rigid engine structure. Although the overall engine structure is undoubtedly rigid, the local region where the anchor bolts attach to the engine must also be analyzed to determine local stresses and rigidity. Excessive local deflections or inadequate torqueing of anchor bolts could affect the overall response of the engine, thereby increasing the seismic loads applied to the anchor bolts.

A summary of the recommended qualification methodology for diesel engines is presented in Table 7-9. The area that requires qualification beyond surveillance testing and maintenance is seismic qualification.

Qualification of the auxiliary components, engine mounts and anchoring is best demonstrated by the use of in-situ testing to identify response frequencies, and, depending on the component frequency obtained, either static or dynamic analyses to ensure that stresses and deflections are within allowable limits.

The engine operating loads on the engine structure and internal components far exceed seismic. Since the engine is designed for long term operation under the operational loads, seismic qualification of the structure and internals is not a concern.

TABLE 7-9

DIESEL ENGINE QUALIFICATION RECOMMENDED PRACTICES

QUALIFICATION AREA	QUALIFICATION METHODOLOGY	COMMENTS
<u>Environmental</u>:		
Pressure Temperature Humidity	Surveillance Testing, Inspection & Preventive Maintenance	Diesel engine components have extensive operating history in more severe conditions. Preventive maintenance programs must address periodic replacement of parts subject to degradation
Radiation	Surveillance Testing, Inspection & Preventive Maintenance	The diesel radiation exposure due to normal and accident conditions is less than 10^4 Rads. All diesel materials will withstand this exposure without unacceptable degradation
<u>Dynamic</u>:		
Vibration	Surveillance Testing, Inspection & Preventive Maintenance	Inspection & preventive maintenance programs address checks for loose fittings and bolts on a regular schedule
Wear	Surveillance Testing, Inspection & Preventive Maintenance	Areas subject to accelerated wear due to nuclear applications (e.g., many quick starts & loads) should be addressed in inspection & preventive maintenance programs

TABLE 7-9

DIESEL ENGINE QUALIFICATION RECOMMENDED PRACTICES

QUALIFICATION AREA	QUALIFICATION METHODOLOGY	COMMENTS
Fatigue	None	Engine design includes consideration of fatigue due to operating loads for greater than 1,000,000 hours of operation
<u>Seismic</u>:		
Engine Structure & Internals	Surveillance Testing, Inspection & Preventive Maintenance	Engine structure and internals are rigid and are designed for operating loads which far exceed seismic loads. Engines have been tested for railroad humping & military shock loads. Engines have been subjected to actual humping loads during shipment & to actual seismic loads. Some impedance testing to determine response frequencies has been performed.
Auxiliary Equipment	Analysis with In-situ Testing	Engines have been tested for railroad humping & military shock loads. Engines have been subjected to actual humping loads during shipment & to actual seismic loads. Some impedance testing to determine response frequencies has been performed. When the component response frequency is not known, testing should be performed to

TABLE 7-9

DIESEL ENGINE QUALIFICATION RECOMMENDED PRACTICES

QUALIFICATION AREA	QUALIFICATION METHODOLOGY	COMMENTS
		determine frequency. If necessary, modify supports to provide additional rigidity. Then perform static structural analysis. Original engine design includes analysis of component support bolts for seismic loads, assuming rigidity.
Anchorage	Analysis with In-situ Testing	Anchor bolts from engine to building have been analyzed statically assuming engine rigidity. Analysis must be extended to engine structure at anchor location to verify rigidity and determine stress/load distribution.

7.10 MECHANICAL DRIVE TURBINES

Mechanical drive turbines are used for safety related applications in nuclear power plants to drive high head pumps such as Reactor Core Isolation Cooling (RCIC) and High Pressure Coolant Injection (HPCI) - both BWR - and Auxiliary Feedwater (AFW) - PWR.

Mechanical drive turbines similar to those used in nuclear power plants have been used for many years in a variety of industrial applications. The primary difference in assuring operation of mechanical drive turbines in nuclear safety related service is that during normal plant operation they are in a standby mode.

Since these mechanical drive turbines are important to the safe operation of the nuclear plant, extensive surveillance and preventive maintenance programs are performed to assure that they will operate as required in an emergency. As a minimum, surveillance tests are performed once every three months by cold quick starting the turbine to simulate an emergency. Preventive maintenance programs are based on experience from both industrial and nuclear applications.

Typical areas covered by surveillance and maintenance programs include:

- Turbine performance
- Excessive vibration/noise
- Excessive leakage (steam, water, oil)
- Lubrication

In general, complete disassembly of the turbine is not performed unless a problem occurs. Experience in industrial and nuclear applications has shown that reliability is best maintained by not disturbing equipment which is operating properly unless a good reason exists for suspecting incipient failure or excessive wear of specific components.

<u>General Description</u>

These turbines are typically of the solid wheel type to provide for rapid starting and operation under potentially high moisture steam conditions. They are fitted with a Kingsbury-type thrust bearing to take axial thrust from the pump and tilting pad journal bearings to prevent any possibility of oil whirl problems occurring during rapid starting.

Steam enters through the Multi-valve Governor Steam Chest and is fed into a steam ring which is cast as an integral part of the casing. The steam issues from the steam ring through the expanding nozzles at high velocity and enters the wheel buckets, where its direction is reversed. The buckets are semi-circular and are milled in the periphery of the wheel. The nozzles are positioned so that the steam enters the bucket tangentially and is reversed 180^{o}, leaving the wheel bucket at the opposite side. A cast stainless steel reversing chamber fastened to the casing at each nozzle position causes the steam to re-enter the wheel several times until practically all available energy is utilized. This principle allows the efficient use of the steam in a single-piece, almost indestructible wheel.

The work is done by the impulse of the steam at the base of the bucket and the partitions guide the steam in its path through the buckets. All expansion takes place in the nozzles; therefore, the horizontal case joint and both glands are subjected only to exhaust pressure and temperature.

The turbine control system consists of a flow controller and an electronic-hydraulic-mechanical turbine governor. The flow controller senses the pump discharge flow and outputs a variable milliampere signal to the turbine governor system. A limiting device on the flow controller restricts the signal range which can be transmitted to the turbine governor where the signal input is converted into hydraulic-mechanical motion for proper positioning of the turbine governor valve. The system is calibrated so that the signal range corresponds to the required speed range of the turbine.

Other active auxiliary equipment required for turbine operation include:

- Turbine Shaft Driven Oil Pump
- Motor Driven Auxiliary Oil Pump
- Turbine Stop Valve
- Vacuum Pump (Gland Condensing System)
- Condensate Pump (Gland Condensing System)

Typical Environmental and Seismic Conditions

Safety-related mechanical drive turbines are located outside containment, usually in an auxiliary building. Typical environmental conditions are listed below:

	BWR	PWR
Temperature		
Normal	85-105°F (30-40°C)	65-105°F
Accident	212°F (100°C)	304°F (151°C)
Pressure		
Normal	-0.4 in. W.G.	Atmospheric
Accident	2 psig	8.6 psig
Relative Humidity		
Normal	20-100%	0-100%
Accident	100%	100%
Radiation		
Normal	$3x10^5$R	$1x10^3$R
Accident	$1x10^7$R (HPCI)	$1x10^4$R
Accident	$3x10^5$R (RCIC)	

The difference in accident radiation doses between HPCI and RCIC reflects the fact that RCIC is intended to perform only when the primary reactor coolant system remains intact, i.e., no LOCA has occurred.

Most safety-related mechanical drive turbines are located at or below grade elevation. Therefore, the seismic excitation at these turbines is the same as the site ground acceleration spectra. At other locations appropriate building floor response spectra should be used. The seismic information presented in Table 7-10 is derived from plants with an SSE horizontal ground motion on the order to 0.2g.

TABLE 7-10

SEISMIC ACCELERATIONS FOR SAFETY-RELATED
MECHANICAL DRIVE TURBINES

LOCATION	DIRECTION	ZPA(f>=33Hz)	PEAK (β=2%)
Basemat	Horizontal	0.38 g	1.8 g
	Vertical	.25 g	1.2 g
Other Locations	Horizontal	0.50 g	1.9 g
	Vertical	.33 g	1.3 g

Failure Modes

The failure modes that can lead directly to failure of a mechanical drive turbine to perform its active safety-related function are summarized in Table 7-11. Also included are the most probable causes of failure, loading conditions and the appropriate qualification methods to eliminate systematic deficiencies in design and operation.

The failure modes are grouped under three main headings:

- Rotating Element Problems
- Control System Problems
- Auxiliary Device Problems

Rotating element problems can become severe enough to result in significant damage to the turbine. Control system problems can lead to the loss of active function of the turbine. Auxiliary devices are also important to the active function of the turbine and potential problems reflect the complexity of the turbine as an assembled unit. Since the electrical components required to assure the active function of the turbine are considered to be within the scope of on-going Electrical Equipment Qualification programs, they are outside the scope of this report.

The primary contributor to rotating element problems is the misalignment of rotating parts. Normal wear of rotating and stationary parts is also a contributor to rotating element problems.

TABLE 7-11

APPROPRIATE QUALIFICATION METHODS
FOR
MECHANICAL DRIVE TURBINES

FAILURE MODE/ PROBABLE CAUSE	LOADING CONDITION	APPROPRIATE QUALIFICATION METHODS
Rotating Element Problems		
Bearing Overload	Normal operational environment Seismic	Surveillance Maintenance Analysis
Lubrication Inadequate	Normal operational environment Radiation $>10^7$R	Surveillance Maintenance Materials Review
Shaft Deflection	Dynamic Seismic	Analysis Surveillance
Thrust Collar Wear	Normal operational environment	Surveillance Maintenance
Coupling Wear	Normal operational environment	Surveillance Maintenance
Control System Problems		
Mechanical Overspeed Trip Failure	Normal operational environment Seismic	Surveillance Maintenance Analysis Test
Governor Failure	Normal operational environment Seismic	Surveillance Maintenance Analysis Test
Auxiliary Device Problems		
Hydraulic Fluid Loss/Degradation	Normal operational environment Radiation $>10^7$R	Surveillance Maintenance Materials Review
Servomechanism Failure	Normal operational environment Seismic	Surveillance Maintenance Analysis Resonance Test
Auxiliary Valve Failure	Normal operational environment Radiation $>10^5$R Seismic	See Section 7.1
Auxiliary Pump Failure	Normal operational environment Radiation $>10^5$R Seismic	See Section 7.8
Gland Condenser Leakage	Normal operating environment Radiation $>10^5$R	Surveillance Maintenance Materials Review

The primary contributor to control system problems is the normal operating environment. The contributors to auxiliary device problems vary with the type of device. The probable causes of these failure modes are shown in Table 7-11.

Gland condenser degradation does not necessarily lead to an active function failure of the turbine. Gasket leakage is significant for turbine qualification only if

- Turbine performance becomes inadequate
- Fluid leakage affects adjacent safety-related equipment

The gland condenser is provided as a subatmospheric pressure collection point for turbine and valve leak-offs. Turbine operation with a leaking gland condenser can cause the escape of potentially radioactive gases and liquids into the area of the turbine and its auxiliaries.

Appropriate Qualification Methods

Routine surveillance and maintenance (S&M) is the appropriate qualification method to address many of the potential failure modes for mechanical drive turbines. Surveillance testing confirms the integrity of the rotating element assembly, including proper alignment/clearances for rotating parts, as well as proper functioning of the overspeed trip mechanism. Analysis, materials review and testing are also appropriate qualification methods, as summarized in Table 7-11.

The metallic and non-metallic parts of nuclear safety-related mechanical drive turbines are subjected to pressure, temperature and humidity conditions which are common to many industrial applications. The maximum temperature of the steam which drives the turbine is 560°F, a condition comparable to other applications.

In PWR applications - Auxiliary Feedwater - the steam is not highly radioactive. In BWR application - High Pressure Coolant Injection and Reactor Core Isolation Cooling - the steam can be more highly radioactive, with a typical integrated dose for an HPCI turbine on the order of 10^7R. The RCIC system is not intended for operation when the primary system is losing inventory via a pipe break and thus is exposed to lower radiation doses when its function is required. The RCIC turbine, however, could - in practice - be qualified to similar environmental conditions as the HPCI turbine since the non-metallic parts which could potentially be exposed to radioactive steam are limited to gaskets and lubricating oil. Environmental qualification for mechanical drive turbines, in addition to normal surveillance and maintenance, is a design review of non-metallic materials to resist exposure to the calculated total integrated radiation dose. In addition to the turbine itself, mechanical control systems and auxiliary devices contain non-metallic materials which require surveillance, maintenance and, perhaps, design review to assure adequate environmental qualification. Justification for not requiring qualification of non-metallic materials in the gland condenser, as discussed above, should be provided. Details of the environmental qualification process are given in Section 4.

Mechanical features which are critical to turbine operability are those which could lead to rotating element or control system problems or to those auxiliary device problems which require qualification to withstand seismic or dynamic loading. The evaluation consists of analysis or testing or a combination of analysis and testing, as shown in Table 7-11.

No unusual dynamic loads have been identified in the review of mechanical drive turbines. The quick-start loads on the turbine exist in all surveillance tests involving quick start. For this reason, surveillance tests are considered an adequate and appropriate method of qualifying mechanical drive turbines for the dynamic loads associated with their application in nuclear plants.

Pipe connection loads are usually analyzed as part of the pressure boundary calculations of the turbine casing and nozzle. Since the turbine casing and nozzle wall thickness are greater than the attaching pipe thickness, the turbine can generally withstand applicable pipe connection loads. Turbine operability when subjected to pipe connection loads is confirmed during routine surveillance testing.

The seismic qualification of a drive turbine encompasses several considerations.

1. The turbine wheel and shaft and the journal and thrust bearings form a rugged, simple design of the turbines rotating element, an assembly that has a high natural frequency and is subject only to ZPA accelerations.

2. The turbine contains several auxiliary devices which are mounted to the turbine base plate. These are devices which must function in order to maintain a functioning turbine, and include:

    (a) The control and stop valves.

    (b) Governor servo and linkage to the control valve.

    (c) Shaft and motor driven oil pumps.

    (d) oil filters

    Oil filters are included to emphasize that certain passive components are critical to the active functioning of mechanical drive turbines. The provision of filtered oil in a hydraulic control or a lubrication system cannot be assured without properly functioning oil filters.

    As-mounted condition of these auxiliary devices must be determined. A combination of analysis and in-situ testing is a practical method to accomplish this. The approach recommended is to determine the natural frequency and to stiffen as necessary to increase the as-mounted natural frequency to 33 Hz or greater where the excitation forces will be at the ZPA level and amplification of these forces will be nil.

In summary, the seismic qualification of a mechanical drive turbine should <u>not</u> focus primarily on the main rotating element mounted in its bearing system, but rather on the operability of auxiliary components. The qualification of auxiliary components involves the following steps:

- Inventorying the auxiliary components.
- Determining structural resonances.
- Adjusting, as required, to override resonances.

The above steps will require a combination of analysis and testing. An analytical study with subsequent tests to demonstrate the controllability of a nuclear auxiliary feed pump service mechanical drive turbine, when subjected to seismic forces, was reported at the ANS/ASME Conference in Portland, Oregon on August 8, 1984 (132). This study showed no resonances, speed or structural, during the resonance search of the mechanical hydraulic governor. Proof tests of the governor under steady state, load drop and quick start conditions were performed with excellent results (132).

7.11 FANS

Fans and fan coolers are used in safety related applications in nuclear power plants for ventilation and room cooling, including reactor containment heat removal. The Reactor Containment Fan Coolers (RCFC) are considered in this section as exemplary for the development of appropriate qualification methods.

Fan coolers are used in many industrial applications where ambient temperature control within buildings and structures is required. Because fans are motor-driven they have received extensive evaluation in various Electrical Equipment Qualification (EEQ) programs, particularly for fans - such as RCFC - which are located in harsh environments.

Since RCFC operation is important to the safety of the nuclear plant, surveillance testing is conducted each month to verify that each fan starts and operates for at least 15 minutes and also to verify that cooling water flow is adequate. Other surveillance during normal plant operation is limited to checking for excessive water leakage since the containment is not accessible. When the plant is shutdown for refueling, repairs or maintenance, preventive maintenance can be performed to assure that the fans are functioning properly. For safety related fans which are accessible during normal operation, surveillance testing can include direct observation of fan performance.

General Description

Typical fans which are used in safety-related application, such as the PWR Reactor Containment Fan Cooler (RCFC) System, are of an axial flow configuration consisting of an outer casing, vanes downstream of the fan rotor, and a means for mounting and supporting the motor in the center of the outer casing. A typical cross-section is shown in Figure 7-16.

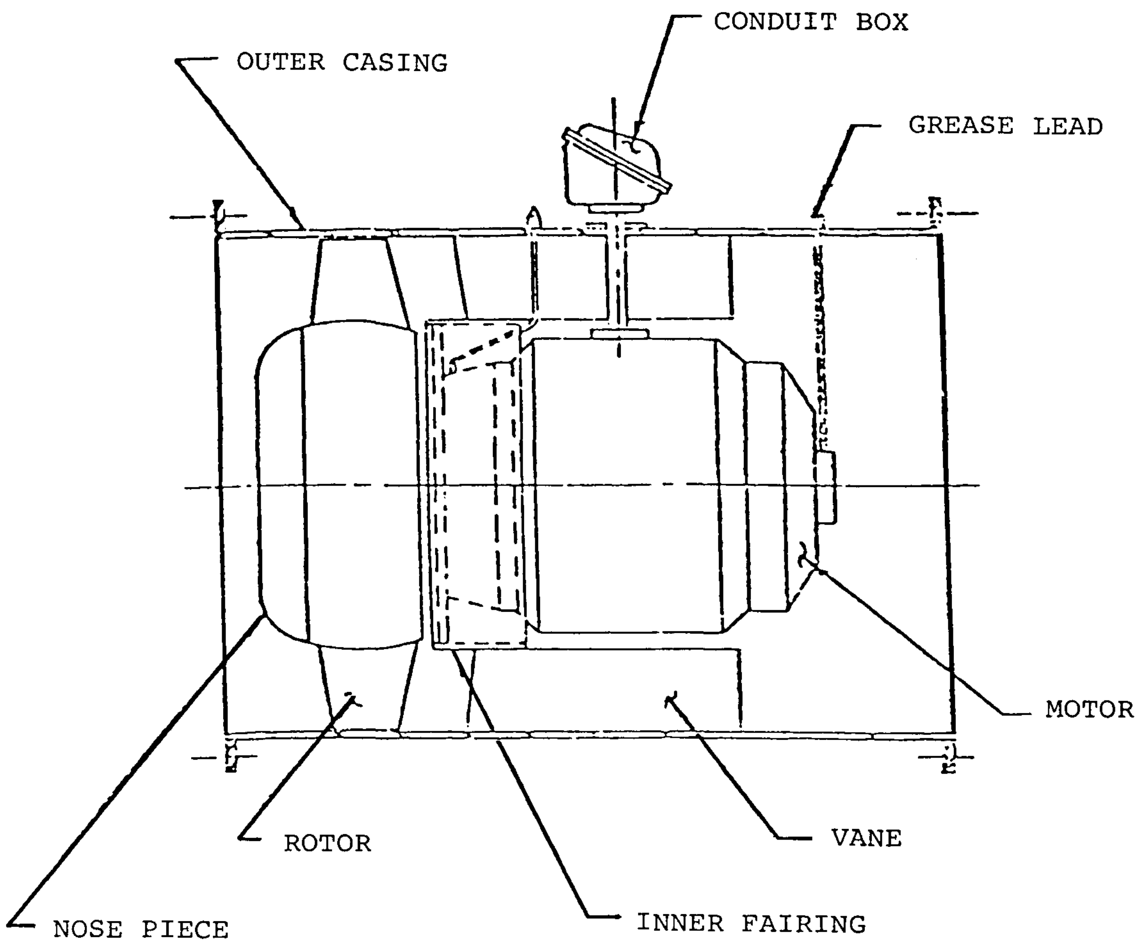


FIGURE 7-16: TYPICAL CROSS SECTION
OF AXIAL FAN

Environmental and Seismic Conditions

The safety-related fan which is subjected to the most severe environment is located inside PWR containment in Reactor Containment Fan Cooler System Service. Typical environmental conditions are listed below.

Temperature

Normal	65-120°F
Accident	320°F

Pressure

Normal	-0.1 to 0.3 psig
Accident	50 psig

Relative Humidity

Normal	0-80%
Accident	100%

Radiation

Normal	$3.5x10^6$R
Accident	$2x10^8$R

The seismic excitations (see Table 7-12) on these fan cooler units are derived from plants with an SSE horizontal ground motion on the order of 0.2 g.

TABLE 7-12

SEISMIC ACCELERATIONS FOR RCFC

DIRECTION	ZPA (f>=33Hz)	PEAK (β=2%)
Horizontal	0.4 g	1.9 g
Vertical	0.27 g	1.3 g

Resonant searches using sine sweep have shown the lowest natural frequency of any consequence for typical units is 40 Hz (See for example Reference 128).

Failure Modes

The mechanical failure modes that can lead directly to failure of a fan to perform its active safety-related function are summarized in Table 7-13. Also included are the most probable causes of failure, loading conditions and the appropriate qualification methods to eliminate systematic deficiencies in design and operation.

The failure modes are grouped under two main headings:

- Rotating Element Seizure/Drag
- Loss of Rated Flow/Head

Rotor element drag can become severe enough to result in excessive slowdown, stall or trip of its prime mover. Loss of rated flow/head is primarily the result of excessive internal bypass.

The primary contributor to rotating element seizure is the misalignment of rotating parts. Normal wear of rotating and stationary parts is the primary contributor to loss of rated flow/head.

Appropriate Qualification Methods

Routine surveillance and maintenance (S&M) is the appropriate qualification method to address many of the potential failure modes for fans. Surveillance testing confirms proper alignment/clearances for rotating parts. Excessive noise or vibration during surveillance testing may require disassembly to determine corrective measures. Correct alignment is properly addressed in maintenance activities during fan reassembly. Analysis and materials review are also appropriate qualification methods, as summarized in Table 7-13.

The metallic and non-metallic components of nuclear safety related fans are subjected to temperature, pressure, humidity, chemical spray and radiation conditions which have already been evaluated in EEQ programs.

Metallic parts which are critical to fan operability include the rotor, shaft and bearings. The appropriate qualification methodology is to perform a static or dynamic analysis to determine the resulting stresses, bearing loads and deflections when the fan is subjected to externally applied loads. The allowable stresses for these parts are established by sound engineering practice and manufacturer's guidelines. Momentary bearing loads for seismic, which are higher than the bearing manufacturer's continuous load rating, are acceptable - see Appendix A. If the deflection calculation shows a momentary loss of running clearance and therefore a rub between metallic parts, a materials review should be conducted to assure that the combination of materials is not susceptible to galling.

Resonance testing to determine the lowest natural frequency (shaft frequency) of the fan to make sure it is above 33Hz - if this has not already been done - may be useful in demonstrating the static analysis methods which are appropriate for fans used in nuclear safety related applications.

TABLE 7-13

APPROPRIATE QUALIFICATION METHODS FOR FANS

FAILURE MODE/ PROBABLE CAUSE	LOADING CONDITION	APPROPRIATE QUALIFICATION METHOD
<u>Rotating Element Seizure/Drag</u>		
Bearing Overload	Seismic	Analysis
Lubrication	Normal operational environment Radiation $>10^7$R	Surveillance Maintenance Materials Review
Shaft Deflection	Dynamic Seismic	Analysis Surveillance
<u>Loss of Rated Flow/Head</u>		
Rotor/Blade/Vane Wear	Normal operational environment	Surveillance

7.12 SNUBBERS

Snubbers are dynamic restraints which are used in nuclear power plants to provide restraint for piping and components when this equipment is subjected to abnormal dynamic loads. While the snubbers must provide restraint for abnormal loading conditions, they must at other times allow normal pipe and component movement, such as results from thermal expansion.

Two general types of snubbers have been developed to provide this dynamic restraint: hydraulic and mechanical. Figures 7-17 and 7-18 illustrate two types of hydraulic snubbers and Figure 7-19 shows a mechanical snubber. Although the principles of operation are completely different for hydraulic and mechanical snubbers, both provide the same function of restraining abnormal component motion, thereby preventing damage to the component. A discussion of snubber design and qualification/acceptance test recommendations is presented in EPRI NP-2297 (117).

A typical arrangement for a snubber installation is shown in Figure 7-20. One end of the snubber is connected to a fixed support, such as a wall or beam, the other end is connected to the pipe or component. Extension pieces are commonly used between the snubber and equipment to accommodate the varied installation requirements.

A substantial amount of testing has been performed on snubbers and snubber components. The results of these tests are largely reported in various unpublished and/or proprietary vendor and other reports (129, 130, 131). These and other such reports are not available without special arrangements with the snubber vendors. These tests have demonstrated that the snubbers (both mechanical and hydraulic) currently in use are highly resistant to environmental degradation and perform satisfactorily under anticipated dynamic loadings.

Description

To perform satisfactorily in an installation on a pipe or other component, a snubber must:

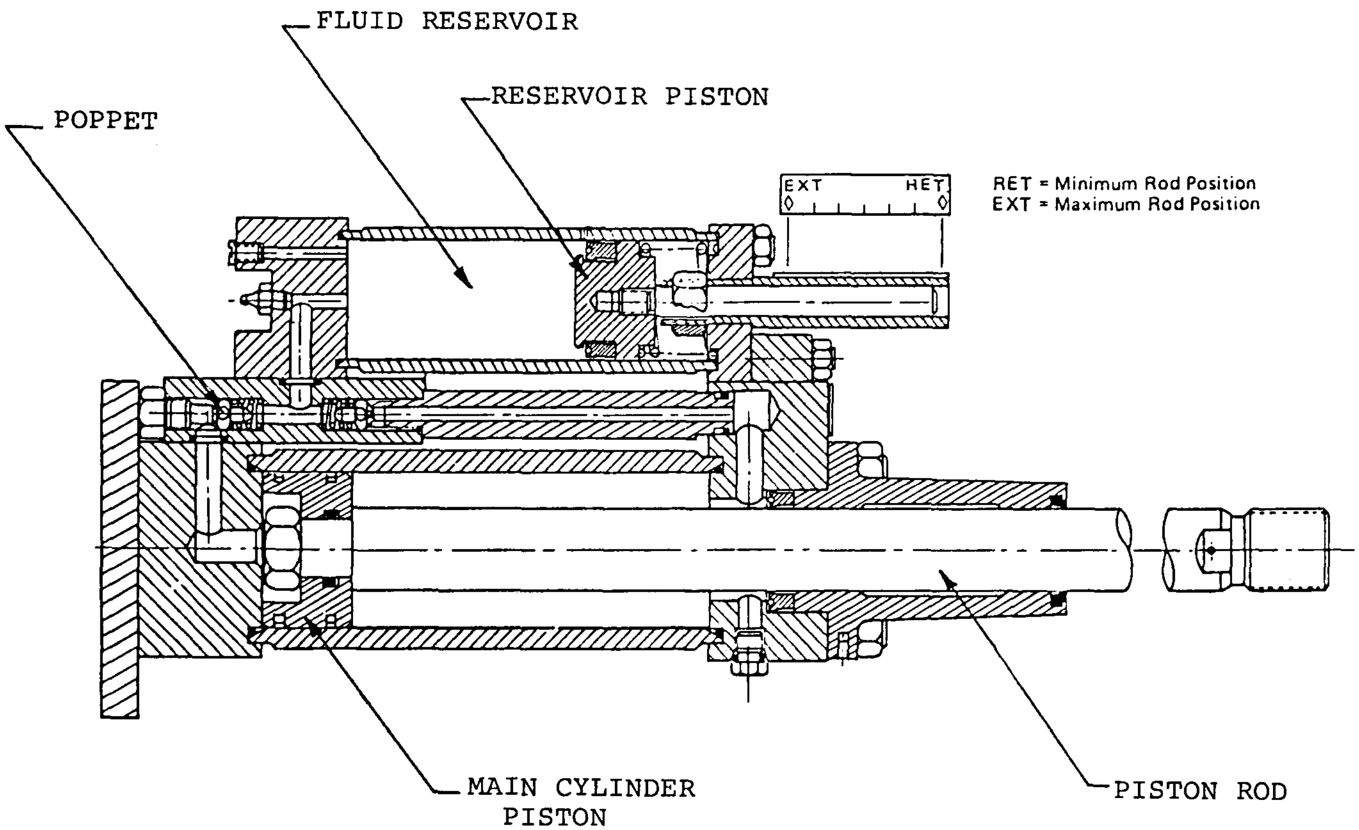


FIGURE 7-17: HYDRAULIC SNUBBER ASSEMBLY

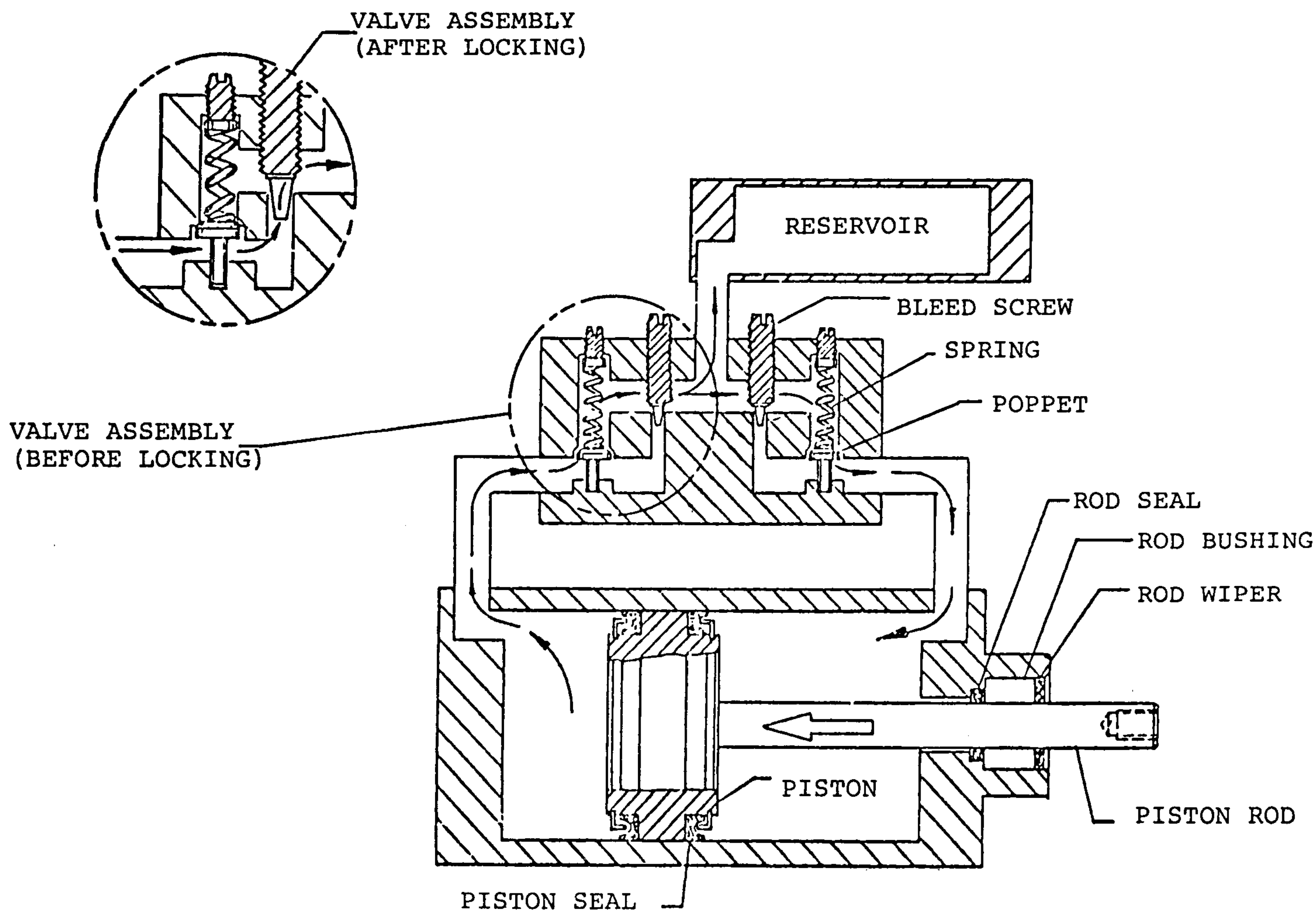


FIGURE 7-18: HYDRAULIC SNUBBER ASSEMBLY

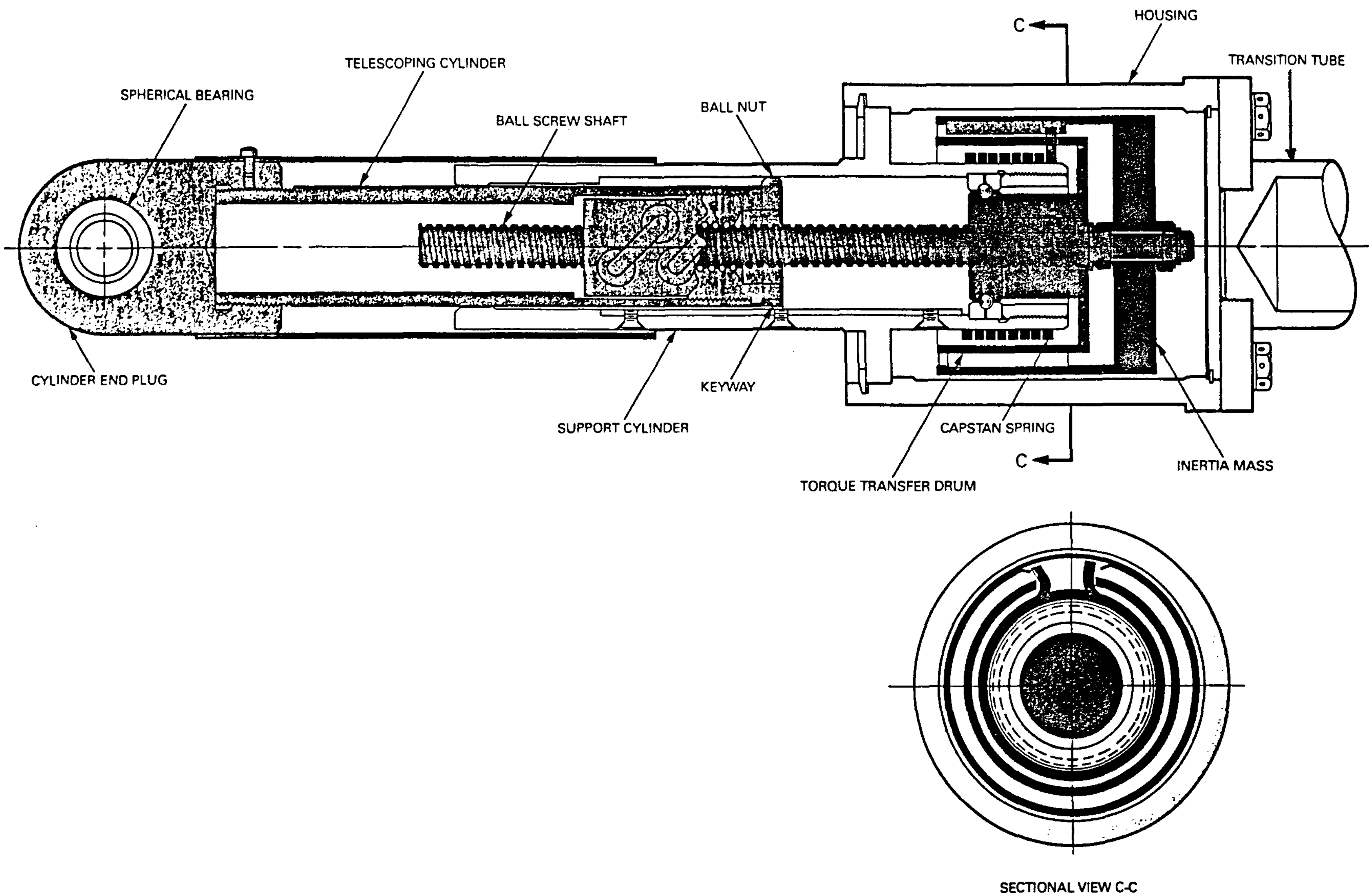


FIGURE 7-19: MECHANICAL SNUBBER ASSEMBLY

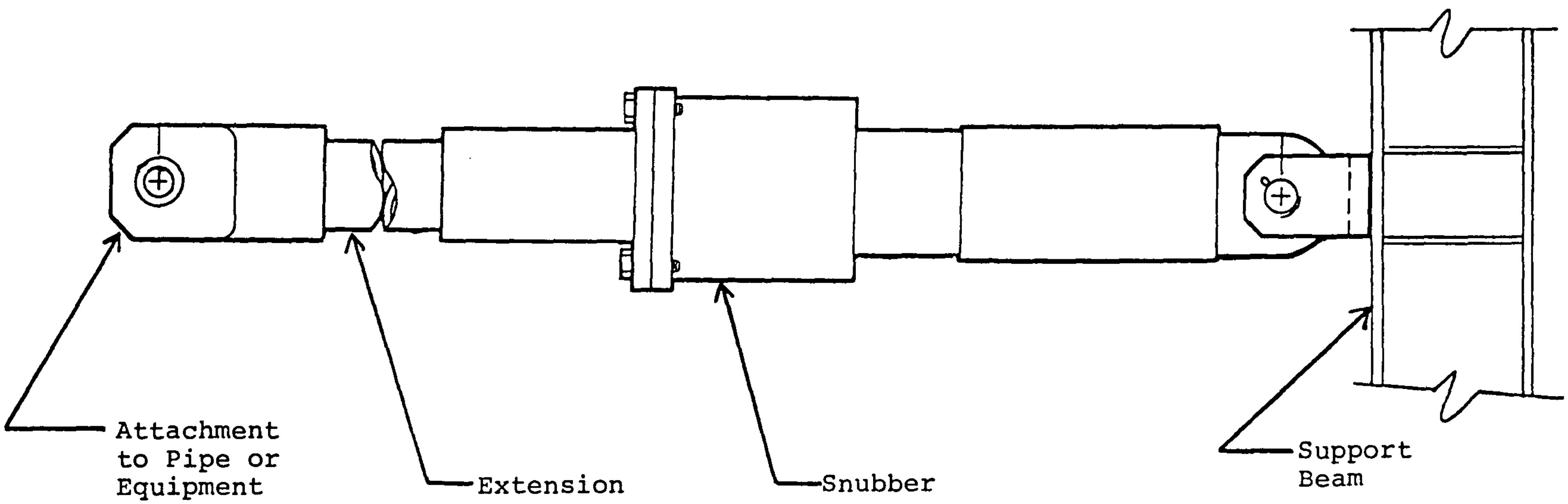


FIGURE 7-20: TYPICAL SNUBBER INSTALLATION

- Allow normal movement due to thermal expansion
- Prevent excessive (abnormal) movement resulting from seismic and dynamic loads

Failure to provide either of these functions constitutes failure of the snubber and may result in failure of the pipe or component. The means used by hydraulic and mechanical snubbers to provide these functions are different.

As can be seen in Figures 7-17 and 7-18, hydraulic snubbers consist of a hydraulic cylinder, piston, piston rod, poppet valves and a reservoir (which may be pressurized as in Figure 7-17 or gravity flow as in Figure 7-18). These snubbers allow gradual pipe movement and resultant piston rod movement by flow of the hydraulic fluid past the poppet valve to the reservoir and/or to the opposite side of the piston. At low piston and fluid velocities, the pressure differential across the poppet is small, the poppet remains open and main piston movement is not restrained. In response to abnormal loads, such as those resulting from seismic or water hammer disturbances, the restrained pipe will drive the piston at a higher velocity. This high piston velocity results in a high fluid velocity which increases the pressure drop across the poppet. When this pressure drop becomes sufficient to compress the poppet spring, the poppet closes and flow is blocked except for the bleed flow. (The Figure 7-18 snubber uses an adjustable bleed screw. The Figure 7-17 snubber has bleed slots built into the poppet.) When the poppet closes, pipe movement is restrained. The fluid bleed provides the appropriate snubber release rate. (Release rate is the rate of piston movement under load after the snubber has been activated.) With the snubber in this activated condition, the pipe load is transmitted through the piston rod to the hydraulic fluid and then through the piston cylinder to the snubber support.

Hydraulic snubbers are velocity sensitive, since they activate based on the velocity input to the piston rod and the resultant velocity of the fluid flowing past the poppet. Therefore, these snubbers are set to activate at a specific piston velocity.

Non-metallic components in hydraulic snubbers consist of the hydraulic fluid and various seals. The fluid and seals must perform properly both during standby and activation of the snubber for satisfactory snubber performance. Deterioration of the fluid or seals could result in excessive drag force during normal pipe movement, failure to activate as required or improper release rate.

Figure 7-19 shows one type of mechanical snubber. Although other mechanical snubber designs do exist, most mechanical snubbers currently in use are of the type shown in Figure 7-19. Movement of a pipe or component connected to the cylinder end plug results in axial movement of the ball nut which produces rotation of the ball screw shaft. The torque transfer drum is keyed to the shaft and rotates with it. As the drum rotates it contacts one of the capstan spring tabs. Since a narrow clearance exists between the inside diameter of the capstan spring and the outside of the support cylinder, the spring is free to rotate. As it rotates the other spring tab contacts the tab on the inertia mass. With the slow rotation which results from normal pipe movement, only a small force is required to start the inertia mass moving. So the capstan spring is compressed very slightly and the inertia mass begins to rotate. This process allows gradual movement of the pipe without activation of the snubber.

In the case of rapid acceleration of the pipe, the above process occurs until the capstan spring tab contacts the inertia mass. In order to accelerate the inertia mass rapidly, a large force is required. This force compresses the capstan spring, resulting in a reduction of the spring inside diameter. The diameter reduction eliminates the spring to cylinder gap and the spring grabs the cylinder, thereby preventing rotation. As the inertia mass will then move, the spring will unlock, and then lock again as it contacts the inertia mass. This provides a function similar to the bleed valve on a hydraulic snubber.

Unlike hydraulic snubbers, this type of mechanical snubber does not activate at a certain velocity. The mechanical snubber shown in Figure 7-19 will activate at a pre-determined acceleration. Because of the differences in other manufacturers' mechanical snubber designs, they may be either velocity or acceleration sensitive.

Mechanical snubbers do not use non-metallic seals and gaskets. Some mechanical snubbers do use small amounts of lubricants. However, degradation of the lubricant will not impair snubber operation. Vendor tests have shown that irradiation of the snubber causes the lubricant to become somewhat tacky, and this results in activation of the snubber at lower accelerations. The post-irradiation activation acceleration was well above the lower allowable limit.

Environmental and Seismic Conditions

Snubbers are located throughout the plant - both inside and outside the containment. Because of the wide dispersion of snubbers throughout the plants, snubbers are exposed to a wide variety of environments. The conditions listed below are the bounding limits for PWR and BWR plants including normal and accident conditions.

Temperature	$30^{o}F$ ($-1^{o}C$) to $340^{o}F$ ($171^{o}C$)
Pressure	-2 PSIG (-13.8kPa) to 50 PSIG (345kPa)
Relative Humidity	Steam
Radiation	$2x10^{8}R$

In addition, snubbers may be exposed (either during snubber storage, shipping, plant construction or operation) to sand, dust or salt spray.

Since the intended purpose of snubbers is to restrain pipes and components when they are subjected to seismic and other dynamic loads, the snubbers are designed to be capable of withstanding these dynamic loads. This capability for some configurations has been demonstrated by both analysis and testing. Most analysis and testing has concentrated on axially imposed loads on the snubbers, because snubbers provide only an axial restraint function. However, dynamic loads may also act on the snubber mass in the transverse direction. The transverse loads are far less than the axial loads, since the transverse loads involve only the snubber mass and acceleration. However, the transverse loads may cause some damage directly or may result in a reduction in axial capacity (e.g., a reduction in snubber buckling load).

In addition, some snubbers may be installed in locations which are subject to low amplitude, high frequency vibrations. An example of this could be a snubber attached to a pipe near a pump discharge. If a specific snubber will be subject to this type of vibration during its lifetime, then qualification by testing of the snubber for the vibration is necessary.

Failure Modes

Other than some fluid/seal incompatibility problems which have been resolved, most snubber failures have been the result of manufacturing flaws, installation damage or damage incurred during plant construction or maintenance. This discussion of failure modes is based on the assumption that the snubbers have been properly installed and tested. Damage to snubbers may occur as a result of improper snubber use (e.g., standing on a small installed snubber) or accidental impact of a snubber with a heavy object, but prevention of this type of damage is dependent on adequate plant construction and operating personnel knowledge and training. It is not amenable to practical solution by snubber design modification nor qualification to these loads.

Table 7-14 lists various failure modes for both hydraulic and mechanical snubbers. As mentioned earlier, snubbers must perform two functions: allow normal movement and prevent excessive movement. In the table, each failure mode is identified as to which type of failure may result. The "excessive drag force" category also includes failures which might contribute to activation of the snubber at a velocity or acceleration low enough to prevent normal pipe movement.

TABLE 7-14

SNUBBER FAILURE MODES

	EXCESSIVE DRAG FORCE (PREMATURE ACTIVATION)	FAILURE TO ACTIVATE
Hydraulic Snubbers		
Dirt in Poppet Valve		X
Poppet Spring Failure	X	
Damaged or Worn Poppet	X	X
Failed Piston or Rod Seals	X	X
Bent Piston Rod	X	
Loss of Hydraulic Fluid		X
Mechanical Snubbers		
Dirt in Ball Nut or Spring	X	
Spring Tab Failure		X
Bent Ball Screw Shaft	X	

Qualification Methodology and Criteria

Qualification and acceptance testing of snubbers have been addressed by the NRC in a draft regulatory guide, Reference (52). In addition, ASME has published Reference (53) which addresses inspection and operability of installed snubbers. The General Electric specification for snubber qualification, Reference (54), is also commonly used by utilities and vendors.

The above documents provide a well defined basis for snubber qualification. The snubber vendors have performed extensive testing of snubber components and complete snubber assemblies to demonstrate the adequacy of their components. This testing has been performed as a part of vendor sponsored development and qualification programs as well as in response to NRC requests and specific client requirements.

These tests cover the areas of degradation due to temperature, pressure, humidity, salt spray, dust and sand. Tests have also been performed to determine snubber lifetime cycles at various loads as well as testing for a 40 year service life. Some testing has been performed to simulate transverse seismic loading by application of a transverse load at the snubber center of gravity while the snubber is subjected to axial cyclic loading at rated capacity.

In general the snubbers have performed well during this testing. Some specific findings from the tests are described below.

Radiation exposure of a Pacific Scientific mechanical snubber to 4×10^9 Rads resulted in the NRR grease 159 becoming tacky (129). This caused the capstan spring to lock onto the cylinder at lower accelerations than pre-irradiation. The results were still within acceptable limits.

Radiation testing of ITT Grinnell snubbers with ethylene proplyene seals to 1.5×10^7 resulted in a minor, but unacceptable, leak from one manufacturer's compound. Retesting with another compound produced acceptable results (131).

The General Electric silicone fluid SF1154, which is used in most hydraulic snubbers, has been shown to be serviceable at 10^8R (131).

As illustrated by the above examples, hydraulic and mechanical snubbers can generally be demonstrated to be qualified for their intended service based on existing tests and analyses. Information on these tests and analyses should be available from snubber manufacturers, NSSS vendors or various utilities which have sponsored previous qualification efforts. Some additional qualification testing may be required for special conditions such as low level vibration, and some hydraulic snubber components may require replacement if they are not made of a qualified material. The snubber manufacturers, in combination with independent test laboratories, have facilities which can provide the required testing.

Section 8
EXAMPLES OF EQUIPMENT QUALIFICATION

The qualification of active mechanical equipment involves an evaluation of operating experience, of the design and materials of construction, and periodic maintenance and testing of the equipment throughout the life of the equipment. This process is illustrated in this section by tracing a mechanical component through the seismic and environmental qualification process and the periodic testing and maintenance process that maintains its qualification. The equipment chosen as the example is a vertical in-line pump used for Residual Heat Removal (RHR) service. The pump is shown in Figure 7-12 and its features are described in Table 8-1.

Qualification of active mechanical equipment includes:

1. Qualification by surveillance testing
2. Seismic qualification
3. Environmental qualification
4. Qualification by preventive maintenance

Each is discussed in this section for the example pump.

This section also contains an example that illustrates the use of in-situ testing to aid the seismic qualification of a diesel engine (Section 8.5). The engine was chosen because of its complexity requiring the methodology of in-situ testing and because of the availability of information.

TABLE 8-1

EXAMPLE PUMP PARAMETERS

<u>PERFORMANCE</u>

Rated Flow:	8500 GPM
Rated TDH:	250 ft.
Rated Speed:	1800 RPM
Maximum Suction Pressure:	150 psig
Maximum Discharge Pressure:	150 psig
Process Fluid Temperature:	350°F

<u>PHYSICAL FEATURES</u>

Closure Gasket:
Spiral wound stainless steel chevrons coated with asbestos.

Mechanical Seal:
Rotating Seal face: Tungsten carbide
Stationary seal face: Graphite
O-rings: Ethylene-propylene

Gear Box:
Lubricant: Oil
Oil Lip Seals: Nitrile rubber

Motor Bearing:
Radial, Deep groove ball bearing, 200 Series, Number 215 per Table 1, Chapter 6, Reference (58)

Motor Stand:
Two 60° access windows from shaft coupling

8.1 QUALIFICATION BY SURVEILLANCE TESTING

All safety-related systems and components are required by Plant Technical Specifications (Tech Specs) to be surveillance tested to demonstrate operability. The plant Tech Specs define a component as being operable when it is capable of performing its specified function(s) (80).

The operability requirement of the Tech Spec for a component is usually stated as an overall functional requirement and addresses only the safety-related function. One Tech Spec (80) addresses the safety-related LPCI mode of the RHR system by the following:

"4.5.3.2 Each LPCI subsystem shall be demonstrated OPERABLE:

a. At least once per 31 days by:

   1. Verifying that the system piping from the pump discharge valve to the system isolation valve is filled with water,

   2. Verifying that each valve (manual, power operated or automatic) in the flow path that is not locked, sealed, or otherwise secured in position, is in its correct position, and

   3. Verifying that the subsystem cross-tie valve is closed with power removed from the valve operator.

b. At least once per 92 days by verifying each pair of LPCI pumps discharging to a common header can be started from the control room and develops a total flow of at least 17,000 gpm (for two pumps in parallel operation or 8500 gpm/pump [Editor's comment]) against a system head corresponding to a reactor vessel pressure of >= 20 psig.

c. At least once per 18 months by performing a system functional test which includes simulated automatic actuation of the system throughout its emergency operating sequence and verifying that each automatic valve in the flow path actuates to its correct position. Actual injection of coolant into the reactor vessel may be excluded from this test."

The Tech Spec requires that the RHR pumps used as the LPCI pumps in the safety-related mode be tested at rated flow once a quarter. The Tech Spec requirement is a general requirement that must be translated into a plant test procedure. The capability to frequently test a standby pump is usually provided in the plant design by such means as a test loop that allows recirculation of flow from the pump discharge back to the suction source.

A plant test procedure will usually include the following:

1. Start the pump from the control room and operate the pump until equilibrium temperature in the lubricating oil (if oil is used) is achieved.

2. Measure pump head and flow. The Tech Spec requirement to "develop a total flow of at least 17,000 GPM against a system head corresponding to a reactor vessel pressure of >= 20 psig" is usually correlated to flow and head measured at a flow meter near the discharge of both pumps.

3. Measure motor current or coastdown time as an indication of a change in internal friction.

4. Check for unusual vibration. Pump shaft vibration can be monitored by permanent or portable vibration meters.

5. A visual check for leakage.

In summary, the periodic surveillance test required by plant Tech Specs, translated into specific measurements in the plant test procedure, allows aging effects such as wear, corrosion and clearance of components to be monitored and tracked throughout the life of the equipment. The Technical Specifications issued for all operating plants require that all active safety-related components, including diesel engines, mechanical drive turbines, pumps and valves undergo periodic surveillance test.

8.2 SEISMIC QUALIFICATION

Figure 7-12 shows the RHR pump and the model of the pump that will be used in the example of seismic qualification by analysis. This pump has no bearings in the pump proper. The bearings are in the motor. The plant location is subjected to a peak ground acceleration of 0.2g.

The seismic qualification will follow the process shown in Figure 6-4 and the discussion of Section 6.6.

<u>Is the Equipment Qualified by Seismic Experience</u>?

<u>Comment</u>. The example pump appears to fall outside the limits of the SQUG/SSRAP recommendation (<u>124</u>) for vertical pumps. Although the pump is vertical, the plant spectrum is below the Type A bounding spectrum, and the pump shaft is short, the shaft <u>does not</u> have a bottom bearing support. This falls outside a literal interpretation of the SSRAP limits on vertical pumps, which states:

"Vertical turbine pumps, i.e., deep well submerged pumps with cantilevered casings up to 20 feet in length and with bottom bearing support of the shaft to the casing are well enough represented to meet the bounding criteria."

Is the Equipment Characterized by Failure Modes that are Structural or Quasi-structural?

Comment. Yes. The potential failure modes of the pump are listed in Table 8-2. These are all failure modes that are structural or quasi-structural since they involve bending of structural elements such as shafts or response of inertia loads to seismic accelerations.

TABLE 8-2

EXAMPLE PUMP SEISMIC QUALIFICATION SUMMARY

FAILURE MODE	ACCEPTANCE CRITERIA	RESULTS*
Rotor Element		
Pump Bearing	Bearing Pressure 1600 psi	Acceptable - Bearing Pressure= 37 psi
Motor Bearings	Rated bearing life under seismic load well in excess of seismic event duration	Acceptable - Rated life under seismic load - 400 hours
Motor Standard Deflection	No permanent distortion; stress 25,000 psi (yield)	Acceptable - Max. stress = 1250 psi

* See Appendix D for Calculations

Can Equipment be Qualified by Static Methods?

Comment. Yes. The pump is characterized by two relatively heavy masses, the impeller and motor, supported on a rigid pump body by a flexible pump shaft and motor stand.

Is Equipment Rigid?

Comment. No. The analysis contained in Appendix D shows the natural frequency of the motor/motor stand to be below 33Hz. Thus, qualification by the Static Equivalent Method must be used. The analysis in Appendix D is based on this method.

Is Equipment Qualified?

Comment. Yes. See Table 8-2.

8.3 ENVIRONMENTAL QUALIFICATION

The pump shown in Figure 7-12 has several components that have non-metallic materials: the pump seal, the motor bearing lip seal and the motor bearing lubricant. Since the bearing is a mechanical component and vital to the operation of the pump - although it is properly a part of the motor - it will be included in the analysis as part of the pump qualification for completeness.

Environmental Qualification Process

The qualification process described in Section 4.4 and Figure 4-1 is shown in Table 8-3 for the example pump.

The process shown in Table 8-3 identified the need to qualify the non-metallic parts because accumulated radiation is greater than 10^5 Rads.

Radiation Qualification Process

The qualification process described in Section 4.4 and Figure 4-2 is shown in Table 8-4 for the example pump. The conclusion is that the materials used are suitable for the intended service, with periodic replacement of ageable parts.

8.4 QUALIFICATION BY PREVENTIVE MAINTENANCE

Periodic maintenance provides a means to maintain the qualification of mechanical equipment by measuring the condition of parts and replacement of parts that are susceptible to aging, such as wear, loosening of parts held in place by light friction fits or set screws, or environmental deterioration of some non-metallic parts.

TABLE 8-3

ENVIRONMENTAL QUALIFICATION PROCESS
PER FIGURE 4-1

1. Is this equipment scheduled for periodic maintenance that addresses parts susceptibility to aging and vibration?

   Comment: Yes. Maintenance is discussed in Section 8.4.

2. Is the equipment subject to an adequate periodic surveillance test?

   Comment: Yes. Surveillance testing is discussed in Section 8.1.

3. Are accident ambient or equipment temperatures significantly greater than surveillance test conditions?

   Comment:
   a. The ambient temperature during surveillance test will most often be less than 100°F, while the accident ambient temperature may be as high as 148°F. This is not considered to be significant for either the metallic or non-metallic components.
   b. During a LOCA, the process fluid source is the ECCS sump with a temperature less than 212°F. During normal operation when the RHR is used in the shutdown cooling mode, the process fluid temperature is as high as 350°F. Thus, the pump is operated at a temperature exceeding LOCA process fluid temperatures and additional thermal analysis is not considered necessary.
   c. Both the seal water and the bearing oil are cooled by heat exchangers using RHR service water. Thus neither the lubricating oil or seal components will be subjected to high ambient or process fluid temperature.

4. Is the equipment exposed externally or internally to chemical solutions?

   Comment: The RHR pump in BWR service will be exposed to demineralized water and the RHR pumps in PWR service will be exposed to dilute solutions of boric acid. Both fluids require that the materials of construction consider corrosion. A materials review shows all materials in contact with the coolant are stainless steel, thus qualifying the pump for both boric acid and demineralized water service.

5. Is the equipment exposed to accumulated radiation $>10^5$ Rads?

   Comment: Yes. See Table 8-4 for the radiation qualification evaluation.

RADIATION QUALIFICATION PROCESS
PER FIGURE 4-2

1. Does this equipment use oil or grease exposed to $>10^7$ Rads?

   Comment: No. The maximum radiation is 10^7 Rads. This value is in the pump itself mainly due to radioactivity in the process fluid. No analysis of the lubricant is required.

2. Does this equipment use non-metallic bearings?

   Comment: No

3. Does this equipment use seals whose failure could result in functional failure?

   Comment: Yes

   a. The impeller side of the pump, which is exposed to radioactive process water, uses a mechanical shaft seal of the type shown in Figure 7-11. Leakage from the seal along the pump shaft could impinge on the pump motor. The rotating seat is tungsten carbide and the stationary seat is graphite. A metallic spring and bellows assembly is employed to maintain pressure and prevent leakage between the rotating seat and the stationary seat. Exposure to 10^7 Rads has no detrimental effect on either tungsten carbide or graphite. O-rings are discussed below.

   b. The pump motor uses lubricating oil and requires a radial lip seal at the end of the bearing housing to prevent oil leakage. The lip seals are fabricated from Nitrile rubber which is recommended for use in lubricating oil. Nitrile rubber retains satisfactory properties for this application at the maximum radiation dose of 10^7R (14). The Nitrile seals should be replaced on a 5 to 10 year schedule because of ordinary aging mechanisms.

4. Does this equipment use diaphragms or non-metallic piston rings to perform its active function?

   Comment: No

RADIATION QUALIFICATION PROCESS
PER FIGURE 4-2

5. Does this equipment use O-rings or gaskets whose failure could result in a loss of active function?

Comment: Yes. O-rings are used to prevent leakage between the pump shaft and the mechanical seal sleeve, and between the seal sleeve and bellows. These O-rings are stationary and are exposed to water environment. They are fabricated from Ethylene-Propylene which has adequate resistance to withstand a 10^7R radiation dose. The O-rings of Ethylene-Propylene should, however, be replaced at 5-10 year intervals (14).

Flange gaskets and other gaskets are of the "Flexitallic" type which consist of a spiral wound stainless steel chevron coated with asbestos. These gaskets are not detrimentally affected by 10^7 R radiation dose.

The extent of preventive maintenance depends on the design and application of the specific mechanical equipment being considered. Examples are the extent non-metallic parts are used, the number of auxiliary mechanical devices required to operate in order to achieve the safety-related function, the results of surveillance testing, and the extent changes and trends go undetected by surveillance testing. Some factors that should be considered during preparation of a preventive maintenance program are:

1. Parts such as organic based seals, gaskets and diaphragms should be periodically replaced at fixed intervals to assure no adverse effects when and if the equipment is required to provide a safety-related function.

2. Parts prone to loosening or fatigue by ambient vibration or the vibration or loads associated with operation and testing of the equipment should be checked frequently, and alternate methods of maintaining tightness and preventing fatigue should be provided, if failures occur. Examples that may need monitoring are mechanical control linkages clamped by set screws and small diameter air, fuel and hydraulic lines servicing the equipment.

3. Large mechanical components such as metal bearings, pump wear rings and engine cylinders, pistons and rings should not have to be measured or replaced except at intervals recommended by the manufacturer or when surveillance test results indicate an adverse trend in performance, such as increasing vibration or reducing head with time.

In many cases the manufacturer will provide recommended maintenance and replacement intervals. These may have to be reviewed, coordinated with surveillance testing and supplemented to assure that all parts prone to aging are checked, refurbished or replaced to maintain qualification.

With the above as maintenance guidelines, the following is a possible preventive maintenance schedule for the example pump:

1. Every year

   - Change the lubrication oil or perform an analysis of a sample drawn from the reservoir.

2. Every five to ten years

   - Replace or refurbish the mechanical shaft seal and the motor bearing oil lip seals.

3. Perform heavy maintenance such as replacement of wear rings or bearings at the frequency recommended by the manufacturer or when observations taken during surveillance testing show adverse trends in measured head/flow or vibration. Replace stainless steel/asbestos gaskets only when closures are opened for other maintenance purposes.

The above schedule avoids unnecessary disassembly of major equipment parts and relies, instead, on surveillance test observations to determine the need for major overhaul. The schedule does however recognize the need for regular, periodic sampling, refurbishment and replacement of ageable, non-metallic materials and parts.

8.5 SEISMIC QUALIFICATION UTILIZING IN-SITU TESTING

In-situ testing, by itself, does not provide complete evidence of component qualification. However, in-situ testing can be a valuable portion of a complete seismic qualification program. In order to illustrate how in-situ testing can be combined with analysis and shake table testing to demonstrate seismic qualification of mechanical equipment, this section presents a case study of a diesel engine seismic qualification.

The particular case to be discussed involved a diesel generator set used for standby power generation in an existing nuclear power plant. The output capacity of the system is approximately 9500 KW provided by a 20-cylinder diesel engine. The diesel/generator unit is approximately 9 ft. x 35 ft. (2.7m x 10.7m) in plan with a height of approximately 15 ft. (4.6m) and a weight of 400,000 lb. (181,000 kg.). A number of engine auxiliary systems are mounted on a separate auxiliary skid. Additional engine components are mounted on an adjacent air-starting skid, and the jacket water heat exchanger is located in a separate building location suitable for a cooling-tower-type structure. This is an extremely large and complex item of equipment. Since it involves the qualification of the large engine structure and many smaller components such as pumps, valves and filters, this provides a good illustration of the wide application of in-situ testing. In this case study only the qualification efforts relative to the engine and engine mounted auxiliaries will be described. The qualification efforts relative to the generator, auxiliary skid and air-starting skid components were similar to those described herein, and their inclusion in this section would not add any significant contribution.

The approach used in the qualification procedure for the diesel was to first use in-situ testing to dynamically excite the engine and associated auxiliaries, and then to use the test data to construct an analytical model of the system. This model was then used to establish vibration levels for shake table tests of small auxiliary components. The actual in-situ testing described was performed at the diesel manufacturers plant. Because of this the engine/skid mounting arrangement was not the same as the foundation and mounting at the actual power plant location. When testing is done in this manner, the effect of the factory mounting in relation to actual power plant mounting must be thoroughly evaluated. Also, after installation of the equipment in the power plant, the response of the equipment, as mounted, should be determined using in-situ testing methods. For this diesel generator mounted in the manufacturer's plant, response modes corresponding to engine rocking on the foundation were identified at frequencies as low as 9.8 Hz. The more rigid foundation provided in the plant eliminated these low frequency responses.

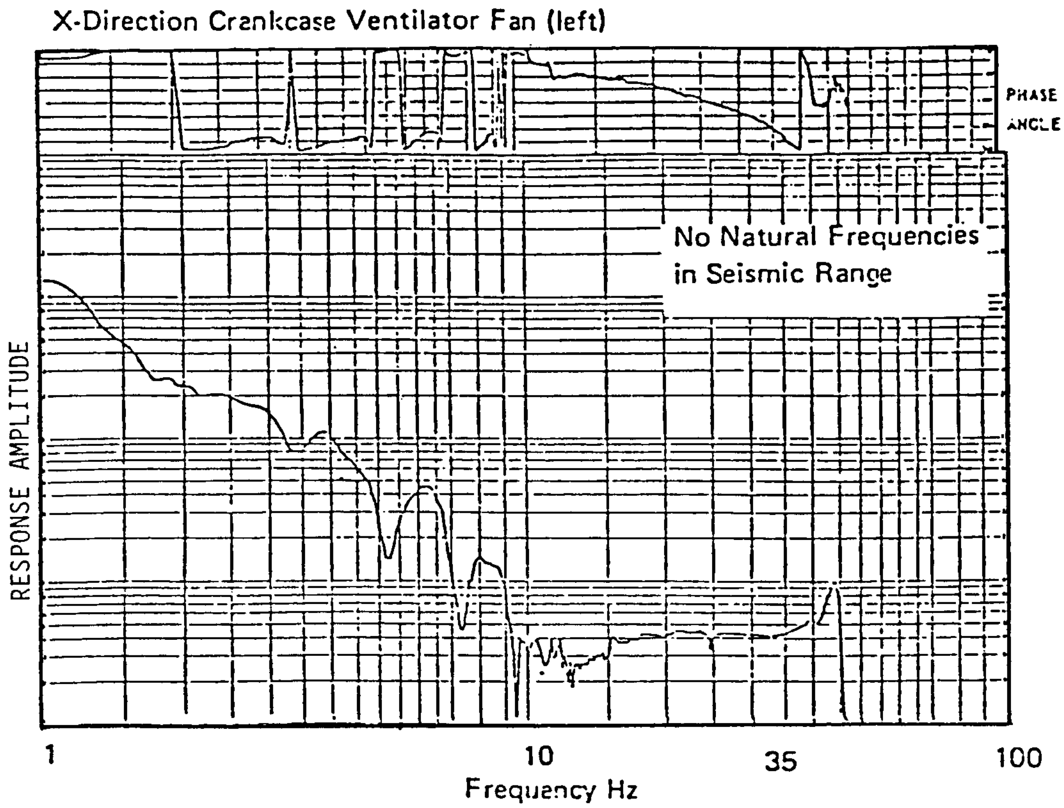
Impact hammer testing was performed on selected engine appendages and auxiliaries to determine if the items had natural frequencies in the seismic range (i.e., <33 Hz). Typical output data obtained using this method are shown in Figure 8-1. This figure shows the response of a typical component in the X and Z directions. The plots shown are not transfer functions, but rather uncalibrated Fourier transforms or spectra of the component response to the impact hammer blow. These plots are referred to as Bode plots, and their purpose is to identify variations in the component response at different frequencies. No ordinate scale is shown because the value of a specific response amplitude is dependent on how hard the impact hammer strikes the component and is, therefore, not relevant. Only the variation in response with frequency is of concern. In

Figure 8-1 the lowest response mode is identified as a peak at 43 Hz. The phase angle shown at the top of each chart is used in interpreting the output data.

Some components were found to have modes within the seismic range. For these components the support arrangements were modified, and the modified components were retested using the same impact hammer testing method. As an example, the top chart in Figure 8-2 shows the identification of prelube filter response modes at 13 Hz and 22 Hz. Since these modes are within the seismic range, additional bracing was added to the filter. After the bracing modifications, the filter response output was as shown in the lower chart where no modes are found within the seismic range.

This procedure of improving component supports was followed until all practical modifications had been implemented. At this point, the entire engine/generator and auxiliary skid assembly were excited in-situ with a 5000-lbf hydraulic actuator. For the particular anchorage and foundation used for the diesel/generator system, this actuator arrangement was sufficient to cause a 0.0015 inch (0.038 mm) motion at the top of the engine when the force was applied laterally at the crankcase centerline. The actuator was driven with a random noise drive signal (white noise), and the resulting transfer functions (output/input) determined at various locations on the engine with emphasis on component mounting bracket attachment points.

Utilizing the results of the above described in-situ testing, together with the known mass distribution of the engine, a lumped mass analytical model was developed for the engine. This analytical model was used both to (1) provide the seismic forces that the engine mounting bolts and anchorage must resist and (2) provide the response motion anticipated at the appendage mounting points. Since the tests were performed at the manufacturer's factory, it was necessary to ignore the low frequency response due to the test anchorage and to model the diesel as though it was mounted in the nuclear plant. The model of the nuclear plant anchorage was then verified by in-situ testing after installation of the diesel engine.



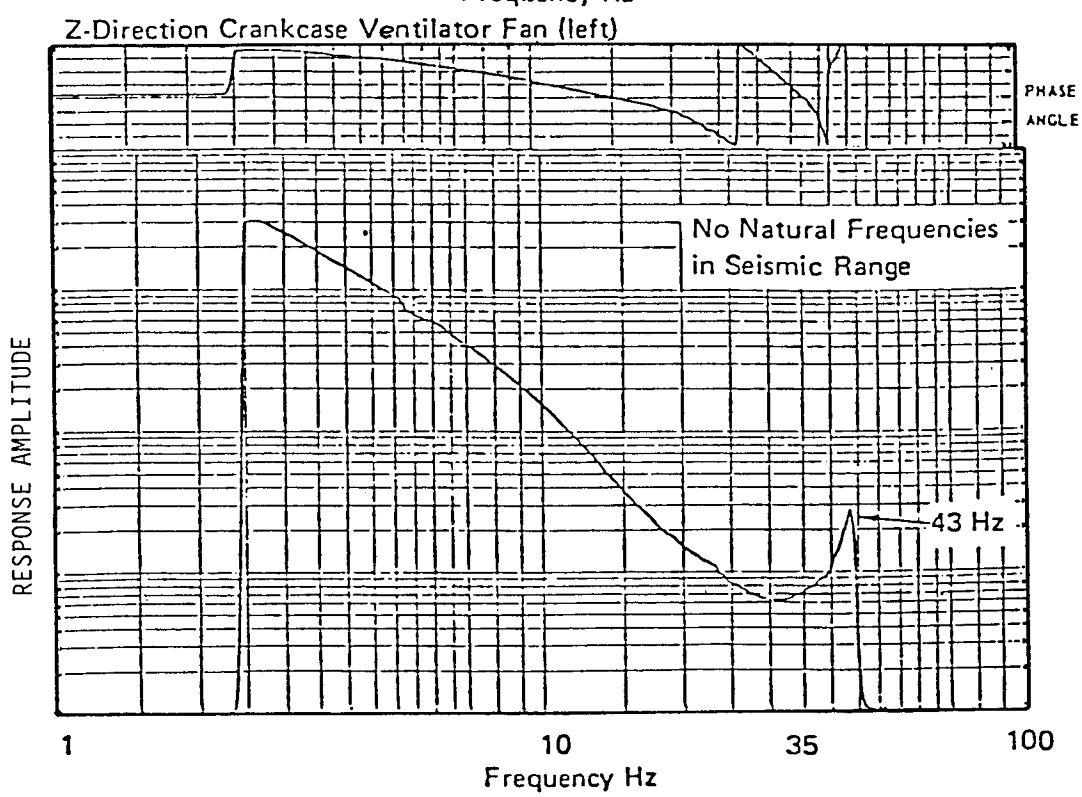
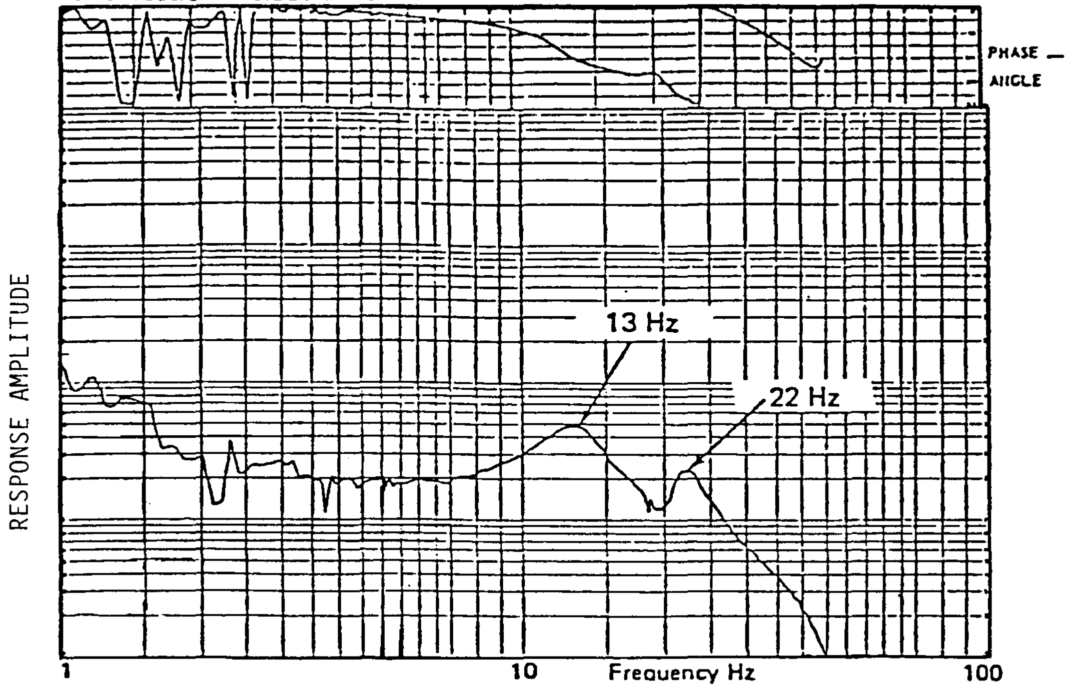


FIGURE 8-1: TYPICAL IN-SITU TEST OUTPUT

X-Direction Prelube Filter



X-Direction Prelube Filter after Add Bracket @ Top

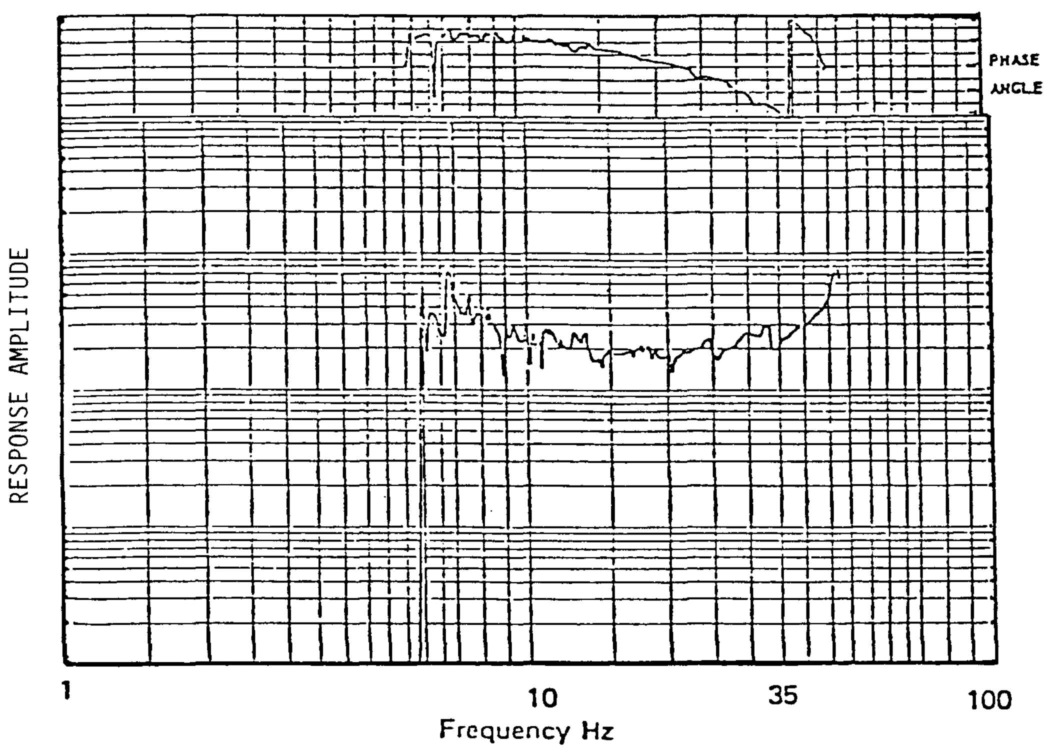


FIGURE 8-2: VERIFICATION OF SUPPORT MODIFICATION ADEQUACY

For the relatively small auxiliary components (particularly those which provide an active function) shake table testing of individual components was utilized. The input vibration levels for these components were established using the analytical model.

As can be seen from the above description, for a complex piece of mechanical equipment such as a diesel engine, it may be necessary to use a combination of in-situ testing, analysis and shake table testing to provide a thorough seismic qualification. Table 8-5 lists the individual diesel engine components and the appropriate qualification methods. Each component was qualified using a combination of in-situ and analysis or in-situ and shake table testing. The components which were qualified by in-situ and analysis consisted primarily of the large rigid components such as the engine structure and the smaller, relatively simple brackets and piping. Most of the active components such as pumps and valves were qualified by shake table testing.

The applications of in-situ testing as described in this section can be widely applied in seismic qualification of other mechanical components. In-situ testing can simplify qualification by demonstrating that the response of a component is outside the seismic range, thereby allowing a simple static analysis to demonstrate structural adequacy. For components which do have response modes within the seismic range, in-situ testing can be used for validation of modifications to supports which may move the component response out of the seismic range. When such modification is impractical, the testing can be valuable in the preparation and validation of analytical models, and in establishing the response motion at components supports (for use in analysis or shake table testing).

TABLE 8-5

DIESEL ENGINE QUALIFICATION METHODS

COMPONENT	QUALIFICATION METHOD			
	IN-SITU TESTING		ANALYSIS	SHAKE TABLE TESTING
	IMPACT HAMMER	RANDOM INPUT		
Foundation & Anchorage (mounting bolts)		x	x	--
Engine Structure (base, frame, block, head)		x	x	--
Turbocharger	x		--	x
Turbocharger - Engine Attachment Brackets		x	x	--
Turbocharger - Intercooler Attachment Bracket		x	x	--
Intercooler	x		--	x
Engine - Driven Lube Oil Pump	x		--	x
Engine - Driven Jacket Water Pump		x	--	x
Jacket Water Piping	x		x	--
Engine - Driven Fuel Oil Booster Pump	x		--	x
Fuel Oil Filters	x		--	x
Fuel Oil Strainer	x		--	x
Exhaust Pipe and Supports	x		x	--
Crankcase Ventilator Fan	x		x	--
Overspeed Trip with Shutdown Valving	x		--	x
Governor	x		--	x
Solenoid Operated Air Start Valve		x	--	x
Starting Air Check Valve		x	--	x

Section 9
QUALIFICATION DOCUMENTATION

Part of the process of qualification is the documentation of evidence that the equipment is qualified to perform its safety-related function and that this qualification is maintained throughout the plant lifetime. The documentation should support the qualification process presented in this report:

Equipment and Loads Identification (Section 3)
Environmental Qualification of Mechanical Equipment (Section 4)
Dynamic Loads Qualification (Section 5)
Seismic Qualification of Mechanical Equipment (Section 6)

In addition, the guidance contained in Section 7 should be used to determine the specific details of each equipment item that require attention in the qualification process. The engineering judgment used in applying experience data should be documented to support the qualification of mechanical equipment. The documentation should include evidence that the mechanical equipment has been properly designed and proof tested at the appropriate ambient and elevated process conditions including operational dynamic loads. In addition, the effects of aging, LOCA and seismic loads should be documented to show that they do not negate the equipment's proven capability.

The evaluation of these effects involves the use and maintenance of the following types of documentation:

- Equipment Specifications
- Load Specifications
- Vendor Instructions/Recommendations
- Radiation Effects Data
- Seismic Experience Data
- Seismic Analysis Reports

- Seismic Testing Reports
- Maintenance Records
- Surveillance Testing Records

TABLE 9-1

QUALIFICATION DOCUMENTATION

	DOCUMENTATION
<u>SERVICE CONDITIONS/ FUNCTIONAL REQUIREMENTS</u>	
Pressure	Certificate of Conformance
Temperature	Certificate of Conformance
Humidity	Certificate of Conformance
Chemistry	Certificate of Conformance
Radiation	Materials Review
Dynamic Loads	Analysis & Test Records Seismic Experience Data
Seismic	Analysis & Test Records Seismic Experience Data
Active Function	Analysis & Test Records
<u>MAINTENANCE</u>	
Wear	Plant Procedure and Records, Materials Review
Ageable Parts	Plant Procedure and Records, Materials Review
<u>PREOP/SURVEILLANCE TESTING</u>	
Active Function	Plant Procedure and Records

Table 9-1 is a summary of the types of documentation which are used in the qualification process. Guidance on the specific types of documentation which are appropriate for each type of mechanical equipment can be found in Section 7. Some features of the documentation package are:

Service Conditions/Functional Requirements

This is the set of equipment service conditions and functional requirements placed on the equipment. These are derived from the system performance requirements, the plant licensing basis and the plant's design basis events. These requirements are usually placed on the vendor through purchase documentation, but can also be documented separately as part of the qualification documentation package.

The types of documentation used to demonstrate conformance with these requirements are:

1. Certificate of Conformance. This is a document certifying that the equipment furnished meets the requirements of the specification and can be provided without further supporting documentation. A certificate of conformance is often considered sufficient for those specification parameters that are not a significant departure from those that are applicable to the other, non-nuclear applications of the equipment, such as temperature, pressure, humidity and water chemistry.

2. Materials Review. Specification conditions such as radiation are unique to the application of the equipment in a nuclear plant. Materials review involves the comparison of the calculated radiation exposure with the capability of the non-metallic material to perform under these radiation conditions.

3. Seismic Experience Data. Documentation of seismic experience data applies in two cases: (a) complete seismic qualification can be verified by direct comparison to the data base; (b) partial seismic qualification can be established because the equipment is included by type in the data base although one or more parameters (e.g., size, mounting or elevation) is outside the limits of the experience data.

4. Analysis or Test. Specification conditions such as radiation, dynamic loads and seismic loads are generally unique, resulting from application of the equipment in a nuclear plant and are often beyond the normal experience of the equipment vendor. In this case, there should be a documented analysis or test demonstrating that the equipment is qualified for these conditions.

5. Performance Test. All active mechanical equipment is required to perform a motion to fulfill its safety function. In general, the test to demonstrate this is performed at the factory and may be accompanied by analysis to the extent that the test cannot simulate

actual specified service conditions. The factory test may be deferred in lieu of a field test in its as-installed, system configurations.

Maintenance

The proper documentation of the monitoring, replacement, and refurbishment, as required, of parts subject to aging are maintenance procedures. Maintenance records document the results of maintenance activities. In some cases, a materials review may supplement the maintenance records, particularly when material substitutions are made.

Preop/Surveillance

Records of periodic testing conducted in accordance with plant technical specifications can demonstrate equipment operability by assuring the equipment is correctly assembled following maintenance.

Section 10

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Appendix A
BEARING CAPACITY

Bearings are used in mechanical equipment to support and locate rotating and sliding components. Bearings transfer axial and radial loads between rotating or sliding and stationary members while allowing relatively free movement and minimizing friction. Two types of bearings are used in mechanical equipment, fluid film bearings and rolling contact bearings.

Both fluid film and rolling contact bearings require lubrication to properly perform their function. Since the qualification of lubricants is addressed elsewhere in this report, this section is concerned only with bearing load capacity. All bearings have a basic load rating that is based on the assurance of long bearing life. However, bearings have the capacity to withstand momentary loads far greater than the rated load without damage to the bearing.

This momentary load capacity can be taken into consideration when analyzing the effects of momentary loads which may result from abnormal occurrences, such as seismic events.

FLUID-FILM BEARINGS

Fluid-film bearings, according to their function, may be:

1. Journal bearings that are cylindrical in shape and support radial loads from a rotating shaft.
2. Thrust bearings that prevent axial motion of a rotating shaft.
3. Guide bearings that guide a machine element in its axial motion, with or without rotation of the element.

Figure A-1 (59) shows a journal bearing and the notation used to describe bearing performance. Lubrication depends on the viscosity of the lubricant and its adhesion to the surfaces of the journal and bearing. The radial clearance provided in the bearing forms a wedge-shaped film between the journal and the bearing. The oil is entrained between the journal and the shaft. A hydrodynamic pressure is created in the film. It is this pressure that prevents metal-to-metal contact and attempts to center the journal in the bearing.

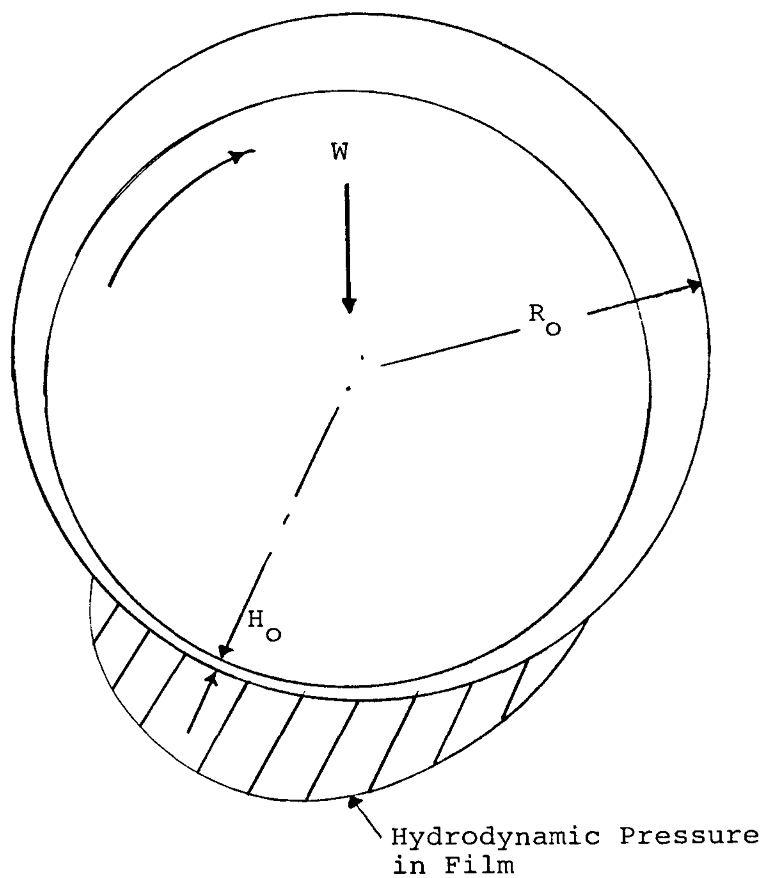

FIGURE A-1: HYDRODYNAMIC BEARING

The allowable bearing pressure inside of a hydrodynamic bearing depends on the application and the degree of finish of the surfaces. Table A-1 (59) lists the current practice in mean non-pressurized bearing pressures. The pressure inside the bearing is very much a function of the eccentricity. The minimum film thickness, h_o, determines the closest approach of the journal and bearing surfaces while maintaining complete lubrication. In practice, h_o = 0.00075 inches (0.019 mm) is commonly used in rotating equipment with steel shafts and babbitted bearings.

TABLE A-1

CURRENT PRACTICE IN MEAN BEARING PRESSURES

TYPE OF BEARING	PERMISSIBLE PRESSURE, PSI, OF PROJECTED AREA
Diesel engines, main bearings	800-1,500
Marine Diesel engines, main bearings	400-600
Steam engines, main bearings	150-500
Steam turbines and reduction gears	100-220
Air compressors, main bearings	120-240
Centrifugal pumps	80-100

Consider a 3-inch shaft that might be used in a pump. Such a bearing might have the following properties.

L/D = 1.0

m = 0.0025

h_n = 0.00075 (0.019 mm)

N = 1800 RPM = 30 RPS

Z = 23.4 centipoises (SAE 20 oil at 105°F)

The load capacity of a journal bearing is given by Equations 29, 30 and 31 of Reference (58). Using these equations and the above data, the unit load capability of the bearing at recommended minimum film thickness is 1600 psi. This capability is considerably higher than the mean bearing pressure normally applied to journal bearings. This indicates that higher pressures from events such as seismic can be accommodated by margins in operating bearings.

In some cases, bearings will act under non-full-film conditions due to high loads squeezing out the film below a minimum film thickness.

In other cases, a bearing type surface, such as wear rings, not normally operating as a bearing will come into contact and operate as a bearing with rubbing wear. Operation under these conditions is often evaluated by the PV (pressure times velocity) or rubbing-factor method (58). The rubbing-factor analysis is not intended to answer the question of how a bearing will perform at any given operating condition. It is primarily intended to provide the limiting conditions under which the bearing will operate satisfactorily. Reference (58) describes the rubbing factor analysis method and relates dry bearing operation (the limiting case) to two important criteria, allowable operating temperature and allowable wear rate. The criteria based on allowable wear rate under dry bearing type of operation is usually governing. The final criteria, based on wear is:

$$PV = \text{Constant}$$

where

$$P = W/DL$$

$$V = \pi DN/12$$

or

$$PV = \pi WN/12L$$

and

W = Load, lbs.

L = Dry bearing length

N = Revolutions per minute

References (58) and (87) provide recommended values of PV for various materials. Manufacturers often have permissible values of PV for materials they use in bearings, wear rings, and elements under sliding contact.

ROLLING CONTACT BEARINGS

Most failures of rolling contact bearings result from lack of lubrication, contamination with foreign material or improper mounting or installation. For a properly designed and installed bearing which is well lubricated and shielded from foreign material, failure of the bearing will eventually occur due to fatigue. Rolling contact bearings are typically selected to provide an adequate fatigue life based on the design loading.

Within certain limits as identified below, the effect of a change in load is limited to a change in the fatigue life of the bearing. This load versus life relationship is expressed as:

$$L = (C/P)^k$$

For a detailed discussion of this topic, the reader should see one of the many references on rolling contact bearings, such as Reference (58). The discussion presented here assumes all loads are radial. The reference addresses the incorporation of thrust loads into the load and life calculations.

When a bearing is subjected to fluctuating loads during its life, the effect on the bearing fatigue life can be determined by finding an equivalent load to use in the life equation. For "a" different loads the equivalent load is:

$$P = \left(\sum_{i=1}^{i=a} P_i{}^k N_i \right)^{1/k}$$

Where

N = Total number of revolutions

N_i = Number of revolutions at Load P_i

This equation for fluctuating loads is valid as long as the load does not exceed the bearing C rating. For abnormal loads, such as shock or seismic loads, the fluctuating load equation can be used if the load and duration are known. Due to the short duration of seismic loads it is apparent that the effect of the increased load on bearing life will be insignificant, as long as the seismic load is less than C. A sample calculation illustrating this is shown at the end of this section.

If the bearing C rating is exceeded by seismic loads, a careful evaluation of the bearing damage is necessary. Excessive loads imposed on the bearing may result in permanent deformation of bearing parts and a reduction in the smooth running properties of the bearings. This can lead to a significant reduction in bearing life.

Rolling bearings are capable of withstanding higher loads when they are rotating than when they are static. Each bearing has both static and dynamic basic load ratings, and the dynamic rating is greater. In a static bearing position excessive loads may result in flat spots on the rolling elements or raceways. When the bearing is rotating, any deformation will be uniformly distributed and will not affect the smooth running properties of the bearing as greatly. Because of this characteristic, the addition of operating loads to the seismic loads may be no worse than the application of seismic loads on the non-operating component.

SAMPLE CALCULATION FOR SEISMIC LOADING

Assume a ball bearing with a normal operating load of 0.25C or 25% of its load rating and a seismic load equal to C and with a duration of 1,500 bearing revolutions (30 seconds at 3000 RPM).

Without an earthquake the rating life is:

$$L \quad = \quad (C/P)^k = (C/0.25C)^3 = 64 \times 10^6 \text{ revolutions}$$

With the seismic load:

$$P \quad = \quad \left[\frac{1}{64 \times 10^6}\left((0.25C)^3 \times 64\text{x}10^6 + C^3 \times 1500\right)\right]^{1/3}$$

$$P \quad = \quad 0.2502C$$

$$L' \quad = \quad (C/0.2502C)^3 = 63.85\text{x}10^6 \text{ Revolutions}$$

The difference between L and L', which is due to the seismic load, is obviously insignificant. Therefore, within the limit of the design load rating C, the imposition of seismic loads on equipment rolling bearings need not be a concern in qualification of the equipment.

TABLE A-2

BEARING NOMENCLATURE

<u>FLUID FILM BEARINGS</u>

C	=	radial clearance = bearing radius - journal radius
D	=	journal diameter
h_n	=	minimum film thickness
L	=	axial length of journal
N	=	speed, rps
Z	=	viscosity
P	=	contact pressure in sliding bearing
p	=	fluid bearing pressure in fluid film bearing
V	=	sliding velocity
W	=	load

<u>ROLLING CONTACT BEARINGS</u>

L	=	rated life in millions of revolutions
C	=	basic load rating which produces a rating life of one million revolutions
P	=	equivalent load actually imposed on the bearing
k	=	3 for ball bearings
k	=	10/3 for roller bearings

Appendix B

SEISMIC EXPERIENCE

A survey of recent world-wide earthquakes has been made using reports cited in the references. The survey is limited to those sites that may have some power generation or processing industry.

1. Two studies (31, 48) reported no failure of active mechanical equipment during the Imperial Valley, California, 1979; Miyagi-Ken-Oki, Japan, 1978; and San Fernando, California, 1971 earthquakes.

2. The data for qualification of mechanical components at many of the other sites is limited by the following considerations:

   --The lack of ground acceleration data and reports as to pre-seismic and post-seismic operability of active mechanical equipment. Most of the available reports have reported Richter scale intensity and have focused on the damage to residential and commercial building structures and the response of civil authority to the crisis.

   --The lack of equipment of sufficient similarity to steam power plant equipment. Many of the seismic locations are remote and undeveloped. Others, including Alaska in 1965, rely on small hydro, gas turbines and diesel-generators for furnishing electric power which, except for diesel generators, have little similarity to nuclear power plant equipment.

3. In addition to the California earthquakes (31) and the Miyagi-Ken-oki Japan earthquake (45) there are other sites that may have equipment with similarity that was subjected to strong motion. This conclusion, inferred from the industries based in the location, would need verification as to type of plants in the area and availability of strong motion recorders. These sites are:

   --Campania-Basilicata, Italy 1980
   --Friuli, Italy 1976
   --Romania, 1977
   --Central Greece, 1981

4. An approach that could provide data during future seismic events is, by agreement with a host country, install strong motion recorders in critical locations in their power plants. Prime candidates are Japan and Taiwan which are in high seismic zones and have large, modern fossil power, nuclear power and process plants.

5. Finally, the nuclear industry could form a team of power plant and electrical distribution experts that could travel to and investigate a seismic event soon after it occurs. It may be possible for this team to work in cooperation with the Earthquake Engineering Research Institute which organizes and dispatches an investigative team of soil and structural engineers to the site as soon as practical after a major seismic event has occurred and prepares both reconnaissance and engineering reports.

Table B-1 lists the major earthquakes that have occurred since 1964. In order to be useful for active mechanical equipment qualification, the seismic data must have the following characteristics:

1. The seismic site must include power plants and/or processing plants that have mechanical equipment handling fluids and gases at high temperature and pressure.

2. The plants at the candidate seismic site should have records describing the performance of mechanical equipment during and after the seismic event.

3. Ground acceleration measurements should be available and should be capable of being used to ascertain accelerations at the mechanical equipment locations.

At a practical level, the seismic events must have occurred at locations where there is a sufficiently industrialized society requiring the generation and distribution of electric power or that has a process such as oil refining that involves high temperature and pressure fluid processing. The seismic location must also be in a country that enjoys sufficiently good relations with the U.S. to permit an investigative team to visit the site and review records and conduct interviews.

Several of the recent major earthquakes have been investigated by the Earthquake Engineering Research Institute (EERI) and the National Science Foundation (NSF). In addition, a special study of California earthquakes has been sponsored by the Seismic Qualification Utility Group (31). Except for the SQUG review the focus of the EERI and NSF has been to report on structural damage to public building and residential structure and the response of civil authority to minimize casualties. Very little is reported on lifeline facilities such as water and power.

The aforementioned EERI and NSF reports, listed in References (32) through (48), are summarized below with respect to their potential source of qualification data to supplement the work done by SQUG.

Montenegro, Yugoslavia Earthquake of 1979

Reference (33) reported that this event registered magnitude 6.6 on the Richter scale and that maximum ground accelerations of 0.46 g N-S and 0.49 g V were recorded. The reference has no discussion of data from power plants or other processing plants. It does note that the Montenegro area is "rich in hydro power." Presumably there are no fossil generating plants or process plants in the area.

Oaxaca, Mexico Earthquake of 1978

Guerrero, Mexico Earthquake of 1979

Reference (34) describes the Oaxaca event as 7.8 and the Guerrero event as 7.6 on the Richter scale and notes that a strong motion instrument recorded 0.2 g N-S and .15 g E-W near Guerrero and 0.01-0.05 g in Mexico City during the event.

This reference does not report any power or process plant data in the Oaxaco or Guerrero areas, but notes that they are both in remote areas east of Mexico City where transmission line failures occurred during the Guerrero event.

Thessaloniki, Greece Earthquake of 1978

Reference (35) reports this to be magnitude 6.5 and a peak of 0.15 g near the city of Thessaloniki. This city with a population of 700,000 reported a loss of 50 lives and relatively minor damage to property. Reference (35) contains no data on industrial damage.

Muradize-Calderan, Turkey Earthquake of 1976

Reference (36) reports this to be magnitude 7.3 with the epicenter near Van, a city of 50,000. An accelerograph recorded 0.07 g at Van. The reference states that there is no industry in the area.

Reference (37) reports that Turkey has experienced 46 earthquakes of magnitude 5 to 8 between 1925 and 1976.

Alaska Earthquake of 1964

Alaska experienced an earthquake of Richter magnitude 8.2 in 1964, the strongest earthquake within the U.S. in recent years. The event is discussed in several references (38-40). Reference (38) states that the Eklutna hydro-electric station remained operable and was put back on-line after failed circuit breakers were jumpered. Anchorage, the largest city in Alaska, experienced damage. The city power was supplied by two 15 Mw gas turbines and a diesel generator. One turbine, in operation before the event, tripped due to circuit breaker trips but was back on line after electrical grid integrity was restored. The other turbine, on turning gear when the seismic event occurred, warped and had to be repaired. The references indicate that the sources of power generation in other Alaska cities are either hydro or diesel-generator. The fact that most "frontier" and third-world countries are limited to gas turbine and diesel engines, not steam plants or process plants, will limit their usefulness for seismic qualification of steam power plant mechanical equipment.

Managua, Nicaragua Earthquake of 1972

This event was reported in Reference (41, 42). They report that the Managua Power Plant, consisting of one 40 Mwe and two 15 Mwe gas turbines, was less than one mile from one of the major faults. Turbine pedestal damage occurred and required repair. There was no report of measured ground acceleration.

Campania-Basilicata, Italy Earthquake of 1980

Reference (44) reports this seismic event as having an intensity of 6.8 to 7.2, a ground motion of 0.35 g measured at Stuno and an estimated peak ground acceleration of 0.6 to 0.7 g at the epicenter. It was estimated that a 20 km by 50 km area experienced accelerations greater than 0.2 g. There was no report on industrial damage, but there may be industry, including power generation facilities since this is in a populated area.

Friuli, Italy Earthquake of 1976

Reference (45) describes this event as about magnitude 6.5 with two aftershocks of about magnitude 6.0. The report describes considerable damage to old structures constructed with thick masonry walls utilizing large rocks and rubble stone bound together with weak mortar. The report contains discussions of the damage that occurred to industrial buildings and modern condominiums in the Friuli area. The existence of industry and modern residences implies the existence of an electric power grid and power generation facilities in the area. However, Reference (45) does not discuss power generation facilities or process plants. Also, it does not discuss ground accelerations.

Romania Earthquake of 1977

Reference (46) reports a magnitude of 7.1 at the epicenter 170 km NE of Bucharest. Ground accelerations were reported.

Epicenter	-	0.23 g
Bucharest	-	0.2 g ground
		0.3 g on roof of 11-story building

There was no report covering operation of power generation or process plants during and following the earthquake. The proximity of this event to Bucharest implies that relatively modern industrial and power facilities may exist in the vicinity of the epicenter.

Central Greece Earthquake of 1981

Reference (47) reports the epicenter to be 60 km from Athens, near Corinth and that the magnitude was 6.7 with aftershocks ranging from magnitude 5 to 6.2. A peak ground acceleration of 0.28 was recorded at the Corinth Telecommunications Center.

Reference (47) does not report on the operation of power generation or process plants during and following the seismic event.

Miyagi-Ken-Oki, Japan Earthquake of 1978

This seismic event is reported as a magnitude of 7.5 (32). This is an important seismic event as its location contained a fossil-fueled power plant and the Fukushima Nuclear Power Plant site where six units were operating or under construction at the time of the event.

Reference (48) reports on the condition of industrial facilities in the area and notes the following of interest.

1. There are six nuclear units at the Fukushima Nuclear Power Plant complex. Units 1 through 5 were operating and Unit 6 was under construction at the time. The only damage reported was a broken ceramic insulator on the output transformer of Unit 1. Units 1 and 6 are heavily instrumented, and more than 20 strong-motion records were obtained. The maximum recorded ground acceleration was approximately 0.12 g; the maximum peak response acceleration of the structures was about 0.25 g. The duration of the recording was somewhat longer than 30 seconds.

2. The new Sendai Power Plant consists of two oil-fired boilers, one is 350 Mwe, the other is 600 Mwe. Both units suffered damage to tubing inside of the boiler and structural support damage to the suspended boiler units. There was no report of failure of active mechanical components such as pumps and valves. The plant seismic alarm, located at the level of the turbine operating floor, was triggered at approximately 0.15 g.

3. The Sendai Refinery was shutdown at the time of the seismic event. Structure failure occurred to three large storage tanks, spilling approximately 68,000 kl of oil, inundating much of the refinery area.

In summary, this was a severe earthquake occurring in a highly industrialized area.

TABLE B-1

MAJOR EARTHQUAKES SINCE 1964

YEAR	LOCATION	MAGNITUDE	REMARKS
1964	U.S.A.: Prince William Sound, ALASKA	8.4	Known as the Good Friday earthquake. Severe damage to Anchorage & other cities. Landslides. Great tsunami damaged many coastal cities in Alaska and in California.
1964	JAPAN: Niigata	7.5	Considerable liquefaction and subsidence caused much building damage. Large tsunami caused coastal flooding.
1965	CHILE (Central)	7.5	Extensive damage.
1966	TURKEY (Eastern)	7.1	
1966	PERU: Huacho	7.5	
1967	VENEZUELA: Caracas	6.5	Many buildings damaged. Several high-rise buildings collapsed.
1967	INDIA: Koyna Dam	6.4	Caused by filling of the reservoir. Village of Koyna Naga heavily damaged.
1968	SICILY (Western)	6.1	Seventeen earthquakes with magnitudes 4.1 to 6.1 from 1/14 to 2/6.
1968	JAPAN: Hachinohe (off the coast)	8.6	Known as the Tokachi-Oki earthquake. Damage to many buildings and port facilities from tsunami.
1968	NEW ZEALAND: Inangahua, Nelson Provincial District	7.1	Widespread damage
1968	IRAN (Eastern) Khorasan Province	7.3	

MAJOR EARTHQUAKES SINCE 1964

1969	INDONESIA: Celebes	6.9	
1969	CHINA: Guangdong Province	5.9	
1970	TURKEY: Gediz	7.3	Many buildings collapsed.
1970	PERU: Chimbote	7.8	Greatest earthquake disaster in the Western Hemisphere.
1971	U.S.A.: San Fernando, California	6.6	Several buildings and highway bridges collapsed. Many instrumental records obtained.
1971	TURKEY: Bingol	6.7	Many villages damaged.
1971	CHILE: Illapel	7.5	Tsunami at Valparaiso.
1972	TAIWAN	7.5	Minor damage
1972	IRAN: Qir	7.1	City destroyed.
1972	U.S.A.: Sitka, Alaska	7.6	Minor damage.
1972	PHILLIPINE ISLANDS	7.4	Tsunami.
1972	NICARAGUA: Managua	6.2	Extensive building damage.
1973	MEXICO: Michoacán Coast	7.5	Heavy damage.
1973	CHINA: Sichuan Province	7.9	Casualties & damage.
1973	MEXICO: Northern Oaxaca	7.2	Many houses destroyed.
1974	CHINA: Yunnan Province	7.1	
1974	PERU: Lima	7.6	Extensive damage in Lima.
1974	KASHMIR	6.0	Villages destroyed.
1975	CHINA: Liaoning Province Haicheng	7.3	Earthquake was successfully predicted. Evacuations took place. Heavy damage but many lives saved.
1975	NORTH ATLANTIC OCEAN	8.1	Slight damage on Madeira Islands.

MAJOR EARTHQUAKES SINCE 1964

1975	TURKEY: Lice	6.7	
1976	KERMADEC ISLANDS	8.0	Tsunami.
1976	GUATEMALA	7.5	Extensive damage to adobe-type buildings. Numerous landslides.
1976	ITALY: Friuli Region (near Gemona)	6.5	Extensive damage; many buildings destroyed.
1976	NEW GUINEA (West)	7.1	Villages destroyed. Landslides
1976	INDONESIA: Island of Bali	6.5	
1976	CHINA: Hebei Province Tangshan	7.8	Major industrial city totally destroyed. Four aftershocks on same day of magnitudes 6.5, 6.0, 7.1, & 6.0.
1977	PHILLIPPINE ISLANDS: Moro Gulf	8.0	Many buildings damaged. Large tsunami.
1976	TURKEY (Eastern)	7.3	Many buildings collapsed in the towns of Muradiye & Caldiran.
1977	ROMANIA: Vrancea Region	7.2	Many buildings collapsed in Bucharest.
1977	IRAN (Southern): Khorgu	7.0	Many villages heavily damaged.
1977	SOLOMON ISLANDS	8.1	Landslides
1977	INDONESIA (south of Sumbawa Island)	8.0	Tsunami destroyed several villages.
1977	ARGENTINA: San Juan Province Caucete	7.4	Extensive damage to village.
1977	IRAN (Central)	5.8	Much damage to villages.
1978	JAPAN: Sendai	7.5	Some buildings damaged in this modern city.
1978	GREECE: Thessaloniki	6.5	Much damage to buildings.
1978	IRAN (Central): Tabas	7.7	

MAJOR EARTHQUAKES SINCE 1964

1978	MEXICO: Oaxaca	7.8	Many buildings damaged.
1978	IRAN: Izeh	6.3	
1979	IRAN (Eastern): Mashad	6.7	
1979	MEXICO: Guerrero	7.6	Many buildings damaged.
1979	YUGOSLAVIA: Southern Montenegro	7.0	Near the Adriatic Coast. Extensive damage.
1979	AUSTRALIA	6.4	Northwest of Perth. Minor damage.
1979	NEW GUINEA (West)	8.1	Located near island of Yapen.
1979	IRAN (Eastern)	6.5	South of Mashhad.
1979	COLOMBIA (Southwestern coast)	7.7	Located off the coast. Large tsunami.
1980	AZORES ISLANDS	7.2	Much damage to homes.
1980	ALGERIA: El Asnam	7.3	Large fault scarps. Many buildings collapsed. 200,000 people homeless. El Asnam 60% destroyed.
1980	ITALY (Southern)	7.0	Several large shocks. Great damage to homes built of stone masonry in Calabritto and nearby towns.
1981	NEW GUINEA (West)	6.7	
1981	GREECE (Eastern Gulf of Corinth)	6.7	Several buildings collapsed in Loutraki, northeast of Corinth. Minor damage in Athens. Many aftershocks.
1981	IRAN (Southwestern): near Kerman	6.9	Town of Golbaf severely damaged; 50,000 people homeless.
1981	SAMOAN ISLANDS	7.9	Largest earthquake in 1981. No damage. Small tsunami.
1981	KASHMIR	6.1	Many houses damaged.
1981	MEXICO: Southern Michoacán	7.2	Extensive damage.

MAJOR EARTHQUAKES SINCE 1964

1982	CANADA: New Brunswick	5.7	Slight damage.
1982	MEXICO: Guerrero	7.2	A second earthquake of magnitude 7.0 on same day. Damage as far away as Mexico City.
1982	EL SALVADOR	7.0	Rural villages damaged.
1982	SANTA CRUZ ISLANDS (Southwest Pacific Ocean)	7.2	No damage.
1982	GUATEMALA-HONDURAS BORDER	5.3	Damage to many dwellings.
1982	YEMEN: East of San'a	6.0	Many villages destroyed. Landslides in mountainous regions.
1982	AFGHANISTAN: North of Kabul	6.8	Extensive damage in Hindu Kush region.
1982	TONGA ISLANDS	7.7	Largest earthquake in 1982. No damage.
1983	COLOMBIA	5.5	Widespread damage in Popayàn. Many buildings collapsed. Thousands homeless.
1983	COSTA RICA	7.1	Near Panamanian border.

Appendix C

ESTIMATE OF MAXIMUM SEISMIC LOADS FOR PIPE MOUNTED EQUIPMENT

In general, the dynamic environment for seismic qualification of pipe mounted equipment such as valves comes from the dynamic analysis of the piping system. In most cases, this analysis would have been performed independently of the building analysis and would have used the results of the building analysis as input to the piping anchor points. The method of analysis would have been either time history or response spectrum. If the mass of the pipe mounted equipment were large it would have been included in the piping analysis. If there were a large extended mass such as a large valve operator, both the mass and its attachment flexibility may have been included in the model. Output of the piping dynamic analysis might include a time history, a response spectrum or possibly only the maximum acceleration (no frequency information) for the location of the equipment. The method chosen for qualification of the pipe mounted equipment would depend, in part, on the type and extent of dynamic information available.

When pipe mounted equipment was originally qualified, and today when this or related equipment is being re-evaluated or requalified, the results of the piping system dynamic analysis with respect to seismic acceleration on pipe mounted equipment are often not available. Consequently, what was originally done, and can be done today, is to estimate the maximum acceleration that the equipment is likely to see and qualify the equipment at that level. This Appendix provides a method of making this estimate.

Review of completed LWR seismic analyses gives some guidance as to maximum acceleration that pipe mounted equipment could experience during a seismic event. Examination of the results of seismic analysis of 76 PWRs was carried out some time ago and reported in the literature (29). The plants studied had safe shutdown earthquake

(SSE) ground acceleration of 0.1 to 0.5 g's and included significantly different building, basemat and soil conditions. The predicted amplification between basemat and the operating floor for these plants was found to have a mean value of 2.4 with a standard deviation of 1.4. Using the mean plus one standard deviation as a conservative estimate of building amplification for all light water reactors, it is unlikely that piping supports would see accelerations greater than 3.8 times the ground levels. (This is reasonable as the differences between BWR and PWR plant designs are not much greater than differences between different PWR designs.)

The amplification in the piping system can be estimated by noting that piping dynamic response is generally dominated by the first mode of vibration which can be characterized with a single degree of freedom model. If the piping were responding to a sinusoidal input at its natural frequency, the amplification would be $1/2\xi$ where ξ is the system damping. On the other hand, if it were responding to a random input the amplification would be $(1/2\xi)^{1/2}$.

In reality, the response of the building (which is the input to the piping) is neither sinusoidal nor random. Consequently, a reasonable compromise is to assume that amplification is approximately $(1/2\xi)^{2/3}$. This results in an amplification factor between the resonant and random cases and when used with conservative damping values, should also be conservative (29).

Figure C-1 shows the maximum expected accelerations for pipe mounted equipment as a function of SSE ground level accelerations. These curves are based on building amplification of 3.8, piping amplification of $(1/2\xi)^{2/3}$ and damping for piping <12 inch dameter and 3% for piping >= 12 inch diameter (30).

Figure C-1 can be used as a conservative estimate of pipe mounted equipment maximum acceleration if other information is not available. If the results of the piping analysis were available, it of course would be used. If the building analysis is available, predicted building amplification can be used in place of the 3.8 number. Also, if more realistic damping values can be justified, they could be used in the calculation.

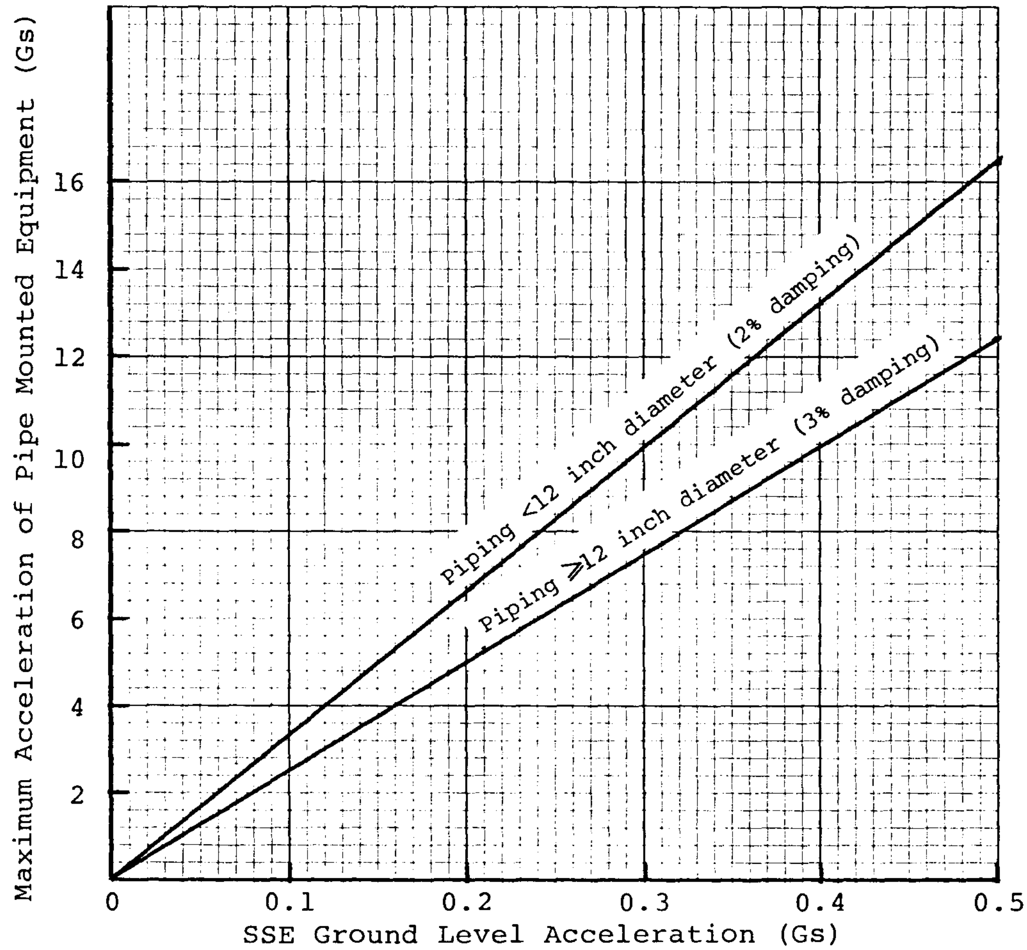

FIGURE C-1: ESTIMATION OF PIPE MOUNTED EQUIPMENT MAXIMUM ACCELERATION LEVELS BASED ON BUILDING AMPLIFICATION OF 3.8 AND PIPING AMPLIFICATION OF $\left(\frac{1}{2\xi}\right)^{2/3}$

There are a number of reasons why the approach proposed here for estimating maximum acceleration for line mounted equipment is conservative. First, with the lack of any frequency information, it is necessary to assume that this acceleration level applies to the entire seismic frequency range. If the pipe mounted equipment had a significant resonance in this range, one would have to assume it were excited by the piping motion. Second, the higher acceleration levels shown in Figure C-1 would physically not be achievable at low frequencies. The displacements associated with a high g, very low frequency response could simply not be accommodated in a piping system which is qualified for seismic service. Third, it is unlikely that the pipe mounted equipment would be mounted at the piping location which would experience the maximum acceleration. Fourth, the damping values used to estimate the pipe mounted equipment acceleration levels are known to be conservative. And finally, while it is true that ground level random excitation produces a narrow band response in piping, its amplitude fluctuates in a random manner. This observation has led to a recommendation to test line mounted equipment with a narrow band input but with the RMS level set to 70% of that specified for a sine beat test at the maximum acceleration (51).

Appendix D
EXAMPLE SEISMIC ANALYSIS CALCULATIONS

This Appendix contains the analysis used to support the example seismic qualification contained in Section 8.2. The definitions and parameters used in the analysis are given at the end of Appendix D.

NATURAL FREQUENCY CALCULATION

A natural frequency analysis does not demonstrate qualification of the pump, but is performed to determine if qualification should proceed in accordance with the Static Method or the Static Equivalent Method. In addition, the calculation is shown since the type of calculation is commonly used for pumps and valves with extended motors. It should be noted that the natural frequency of the motor/motor stand could also be determined using in-situ testing methods discussed in Section 6.

The pump/motor assembly shown in Figure D-1 is a simplified, single degree of freedom system. Such a simplified model can be used to calculate the lowest fundamental frequency of the system where the maximum modal participation is taking place.

The natural frequency for a cantilever beam with a concentrated mass at the free end is calculated by (82):

$$f = \frac{1.732}{2\pi}\sqrt{\frac{EIg}{WL_z^{\,4}}}$$

Where I is the beam moment of inertia and W is mass. For simplicity and conservatism only the stiffness of the motor stand is included in the frequency calculation. Because of the access windows in the motor stand, the moment of inertia in the two orthogonal axes will be different. These moments of inertia, as shown in Figure D-1, are

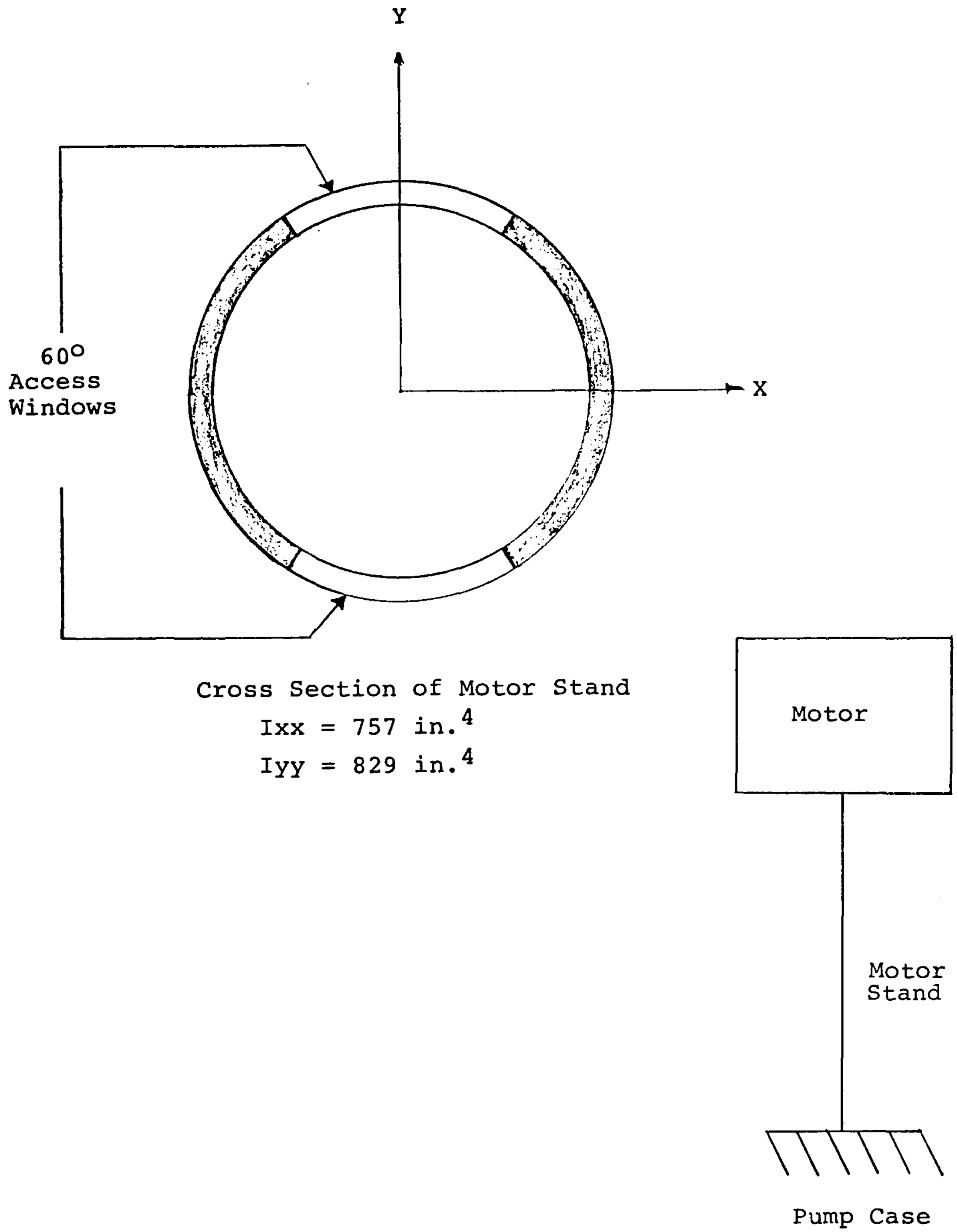

FIGURE D-1: SIMPLIFIED MODEL FOR RESONANT FREQUENCY CALCULATION conservatively assumed to apply to the entire length of the motor stand, i.e., the stiffer non-access window cross-sections are not considered.

Now,

$$W = \text{weight of motor} + 2/3 \text{ weight of shaft} + 2/3 \text{ weight of motor stand}$$

$$= 1000 + 2/3 \times 80 + 2/3 \times 200 = 1187 \text{ lbs.}$$

$$f = \frac{1.732}{2\pi}\sqrt{\frac{27\text{x}10^6 \times 757 \times 32 \times 12}{1187 \times 49^4}} = 9.3 \text{ Hz}$$

The above analysis shows a resonant frequency of about 9 Hz. This is below 33 Hz, thus requiring qualification by analysis using the Static Equivalent Method. Consequently, horizontal acceleration value of 1.7g is selected for the seismic evaluation of the pump. This is based on multiplying the horizontal acceleration of Figure D-2 by $\sqrt{2}$. This factor is justified on the basis of the SRSS method of combining the directional effect of the earthquake (83). The higher modes of simple structures are generally well separated (114). Therefore, no closely spaced mode effect needs to be considered for this pump and the factor of 1.5 often used with the Static Equivalent Method is not applied to the example.

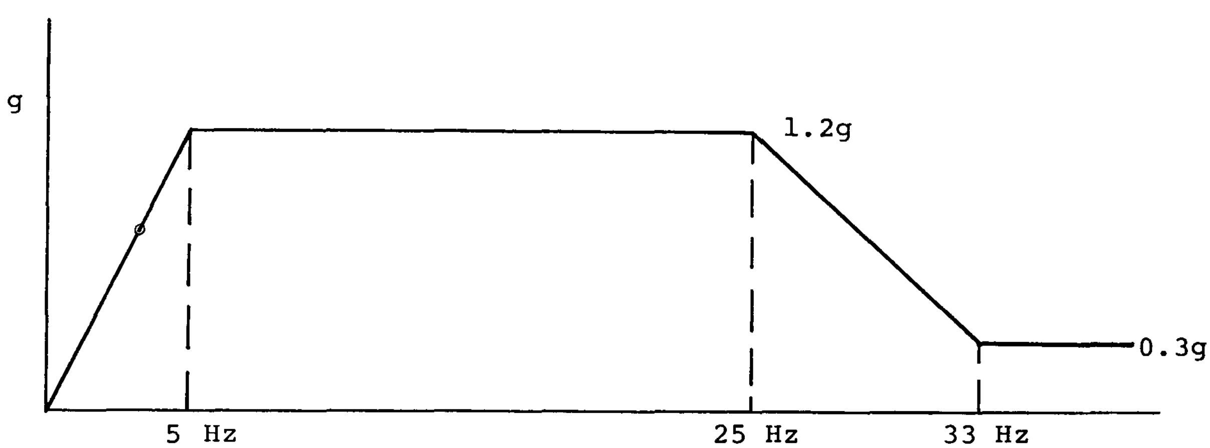

FIGURE D-2: PUMP SEISMIC RESPONSE SPECTRUM

PUMP BEARING LOADS

The shaft and impeller will be subjected to seismic forces which will result in load on the journal bearing located immediately above the impeller. This load is estimated to be:

$$W = 1.7\ g \times (\text{weight of impeller} + 2/3 \text{ weight of shaft} + \text{hydrodynamic mass})$$

$$= 1.7 \times (250 + 2/3\times100+60) = 640 \text{ lbs.}$$

The pump bearing is 3 inches in diameter by 3 inches long. The bearing load is calculated to be:

$$P_b = \frac{W}{LxD} = \frac{640}{3x3} = 70 \text{ psi}$$

This bearing load is well within the capability of 1600 psi estimated in Appendix A for a bearing of this size. The seismic load is low enough that the evaluation does not have to consider the internal loads associated with operation which are estimated to be less than 100 psi based on information contained in Appendix A.

MOTOR BEARING LOADS

The motor bearings in the example pump must accommodate the horizontal motor rotor seismic load as well as the vertical rotor and pump loads as the pump bearing does not have thrust capability. The bearings are ball bearings, one supporting each end of the motor shaft.

The horizontal load on the motor will be due to the horizontal acceleration of the motor rotor, a portion of the pump shaft and the rotor loads acting under the influence of the magnetic field if the motor is energized. The latter is usually less than 1/10 of the bearing rating to provide a rating life to 10,000 hours (See Table 17, Chapter 6, Reference 58).

For a 3-inch, 200 Series ball bearing,

$$C = 11{,}400 \text{ lbs.}$$

$$F_r = 1/2\ (1.7\ [W_s+W_r] + 2 \times 0.1\ C)$$

$$= 1/2\ (1.7\ [100+1000] + 2 \times 0.1 \times 11{,}400)$$

$$= 1/2\ (1870+2280) = 2075 \text{ lbs. for each bearing}$$

The seismic load is less than the operating load.

The vertical load F_a on the motor can be calculated by:

$$F_a = F_1 + F_2 + F_3$$

Where

F_1 = the vertical seismic load

F_2 = the vertical internal load due to internal fluid pressure

F_3 = vertical hydraulic thrust of the impeller

The vertical seismic load, F_1, can be calculated from the vertical seismic coefficient acting on the pump shaft and impeller and the motor rotor.

$$F_1 = g_v\ (W_s+W_p+W_r)$$

$$= 1.0\ (100+250+1000) = 1350 \text{ lbs.}$$

The vertical upward thrust due to internal fluid pressure, F_2, can be calculated from the internal working pressure times the shaft cross-sectional area.

$$F_2 = P \times A_{ps}$$

$$= 250 \times 7 = 1750 \text{ lbs.}$$

The vertical hydraulic thrust, F_3, varies with impeller design and must be obtained from the manufacturer. For vertical pumps, the thrust at rated flow is downward, opposite the vertical upward pressure from internal pressure. In the absence of specific thrust data from the vendor, a conservative assumption is to neglect the hydraulic thrust and the downward pull of gravity as offsetting loads and evaluate the bearing loads by the full upward vertical seismic coefficient and internal pressure loading.

$F_a \quad = 1/2\ (F_1+F_2) = 1/2(F_1+F_2) = 1550$ lbs.

Using Table 1, Chapter 6, Reference 58:

$C_o \quad = \quad 9250$ lbs.

$V \quad = \quad 1.0$ (inner ring rotates)

$F_a/C_o = \quad 1550/9250 = 0.17$

$e \quad = \quad 0.34$

$Y \quad = \quad 1.31$

$F_a/VF_r = \quad 1550/2075 = 0.75$

$X \quad = \quad 0.56$

$$P = XVF_r + YF_a$$

$$= 0.56x1x2075+1.31x1550 = 3190$$

Using Equation 27, Chapter 6 of Reference 58:

$$L = \frac{16,667}{n}\left(\frac{C}{P}\right)^R$$

$$= \frac{16,667}{1800}\left(\frac{11,400}{3190}\right)^3 = 423 \text{ hours.}$$

The rated bearing life of the pump motor bearings is well in excess of the duration of the seismic event. Thus, the seismic event will have negligible effect on the rated bearing life under normal loads that will exist before and after the seismic event has occurred.

MOTOR STAND DEFLECTIONS

The pump motor stand is subject to a certain amount of deflection as the internal mass of the motor and motor stand responds to the seismic accelerations of the pump casing. Operation of the pump may be impaired if this deflection causes permanent misalignment of the motor and pump shafts. This is generally avoided if the stresses in the motor stand stay well below the yield strength of the motor stand.

The induced bending and shear stresses in the motor stand can be calculated by:

$$S_o = \frac{MC_{ms}}{I}$$

$$S_s = \frac{F_s}{A_{ms}}$$

Where the moment of inertia, I, is taken as the lower value shown in Figure D-1.

$$I = I_{xx} = 757 \text{ in.}^4$$

The overturning moment, M, is:

$$M_b = g_n L_z \times (W_m + W_{ms})$$

$$= 1.7 \times 49 \times (1200+200) = 116{,}620 \text{ in-lbs.}$$

and the bending stress,

$$S_b = \frac{99{,}960 \times 9.5}{757} = 1460 \text{ psi}$$

The shear force, F_s, is

$$F_s = 1.7(1200+200) = 2380 \text{ lbs.}$$

and the shear stress,

$$S_s = 2380/19.4 = 125 \text{ psi}$$

These stresses are well below the yield stress for steels, usually in the range of 25,000 psi. Thus, deformation of the motor stand will not occur and there will be no impairment of the alignment.

ANALYSIS DEFINITIONS AND PARAMETERS

A_{ps} = Cross-sectional area of pump shaft = 7.0 in.^{2}

A_{ms} = Cross-sectional area of motor stand = 19.4 in.^{2}

E = Modulus of elasticity of motor stand = 27×10^6 psi

L_z = Effective length of motor stand being beam connecting motor center of mass to valve body = 49 inches

I_{xx} = Bending moment of inertia of motor stand about x-axis, in.^{4}

I_{yy} = Bending moment of inertia of motor stand about y-axis, in.^{4}

I = Bending moment of inertia of motor stand, in.^{4}

W_d = Weight of motor stand = 200 lbs.

W_h = Hydrodynamic mass of water in pump = 60 lbs

W_r = Weight of motor rotor = 1000 lbs.

W_p = Weight of pump impeller = 250 lbs.

W_s = Weight of pump shaft = 100 lbs.

W = Weight, lbs.

g = Acceleration of gravity = 386 in./sec.^{2}

P_b = Pump bearing pressure, psi

W_m = Total weight of motor = 1200 lbs.

F_a = Thrust load on motor ball bearing, lbs.

F_r = Radial load on motor ball bearing, lbs.

C = Motor ball bearing basic load rating, lbs.

C_o = Motor ball bearing basic static load rating, lbs.

d = Pump discharge design pressure, 250 psi

g_h = Peak horizontal seismic acceleration = 1.7 g

g_v = Peak vertical seismic accleration = 2/3 g_h = 1 g

V = Ball bearing rotation factor per Table 1, Chapter 6 (58)

e = Ball bearing reference valve per Table 1, Chapter 6 (58)

x = Ball bearing radial factor per Table 1, Chapter 6 (58)

Y = Ball bearing thrust factor per Table 1, Chapter 6 (58)

P = Ball bearing equivalent load, lbs.

k = Ball bearing rating exponent = 3

n = Ball bearing raceway speed, RPM

S_b = Motor stand bending stress, psi

S_s = Motor stand shear stress, psi

M = Bending moment on motor stand, in-lb.

C_{ms} = Radius of motor stand = 9.5 inches

F_s = Shear load on motor stand, lbs.

Appendix E
RESPONSE SPECTRUM ANALYSIS METHOD

A continuous physical system such as a pump or a valve is mathematically idealized as an assemblage of one dimensional elements. Normally, beam elements with six static degrees of freedom and three dynamic degrees of freedom (translations only) are utilized. In the case of pump models, all journal bearings and thrust bearings are modelled as linear springs. In general, the mathematical model should characterize each relevant component of the equipment important to its safety function or structural integrity.

Upon constructing the mathematical model, the undamped natural frequencies and mode shapes of the equipment are obtained in the following way: first, the total stiffness matrix (K) is assembled by the superposition of all stiffness matrices of the various beam elements. Next, the diagonal mass matrix (M) for the entire system is assembled. The equation of motion for free vibration of an undamped system can be written as:

$$M\ddot{u} + ku = 0 \qquad (1)$$

Where

M = diagonal mass matrix

K = total stiffness matrix

$\ddot{u}$ = acceleration column vector

u = displacement column vector

Equation (1) can be rewritten as:

$$K^{-1} M\ddot{u} + u = 0$$

If the displacement is assumed to take the form of simple harmonic motion, the free vibration equation of motion can be further rewritten as:

$$K^{-1} M\phi = 1/\omega^2\phi \qquad (2)$$

Where ϕ is the mode shape and ω is the circular natural frequency of the simple harmonic motion assumed for the displacement u.

Equation (2) is an eigenvalue problem and can be satisfied with particular values of W and the associated ϕ. There are N eigenvalues corresponding to N dynamic degrees of freedom. There are many methods to solve an eigenvalue problem in the form of $AX = \lambda X$. The inverse iteration method is commonly used for large models to obtain the modes of vibration. The undamped natural frequencies obtained are almost the same as the damped natural frequencies for a system with reasonably small structural damping factors. As an example, for 10% damping, the damped natural frequency is approximately 0.995 of the undamped natural frequency.

The equation of motion for the dynamic response of a linear elastic system subjected to a base excitation $\ddot{u}_g$ is written as:

$$M\ddot{u} + C\dot{u} + Ku = -M\delta\ddot{u}_g \qquad (3)$$

Where

M	=	mass matrix
C	=	damping matrix
K	=	stiffness matrix
u	=	displacement vector
$\dot{u}$	=	velocity vector
$\ddot{u}_g$	=	ground acceleration
δ	=	a directional vector

This is a system of "n" 2nd order coupled linear differential equations describing the structure in its geometric coordinate or degree of freedom characteristics.

By performing a suitable transformation, a decoupled system of independent 2nd order linear differential equations in a generalized (normal) coordinate system can be obtained. This transformation is:

$$\{u\} = [\phi]\ \{q\} \tag{4}$$

Where the components of the vector g are the generalized coordinates. the matrix ϕ is the eigenvector which was obtained by solving Equation (2) as described earlier.

Substituting the above transformation in Equation (3) and premultiplying by ϕ^T one can obtain:

$$\phi^T M\phi\ddot{q} + \phi^T C\phi\dot{q} + \phi^T K\phi q = -\phi^T M\delta\ddot{u}_g \tag{5}$$

Considering the orthogonality property of normal modes and assuming that the damping matrix C is proportional to a linear combination of the mass and stiffness matrices such that

$$[C] = \alpha[M] = \beta[K] \tag{6}$$

The ith uncoupled system of 2nd order linear differential equations can be written as

$$\ddot{q}_i + 2\xi\,\omega_i\dot{q}_i + \omega_i^2\ q_i = -\Gamma_i\ \ddot{u}_g/M^* \tag{7}$$

Where

M^* = $\phi^T M\phi$ commonly called the generalized mass matrix

$2\xi\omega_i$ = $\phi^T C\phi\ /M^*$

ω_i^2 = $\phi^T K\phi\ /M^*$

ξ = percentage of critical damping

Γ_i = $\phi^T M\delta$= modal participation factor

ω_i = the ith natural frequency

The numerical solution of Equation (7) in the generalized coordinate system is obtained

$$q_i = \frac{\Gamma_i}{M^*\omega_i^2}\ (S_a)_i \tag{8}$$

Where $(S_a)_i$ is the maximum value of spectral acceleration for mode i given

$$(Sa)_i = \omega_i \int_0^t \ddot{U}g \; e^{-\xi\omega_i(t-\tau)} \sin\omega_i(t-\tau)d\tau \qquad (9)$$

Once q_i is determined, the solution in the original coordinate system (i.e., geometric coordinate system) is obtained using the transformation defined by Equation (4).

$$[U_{ij}] = [\phi] \; \{q\} \qquad (10)$$

Where U_{ij} is the displacement at mass point i in mode j.

SEISMIC LOAD CALCULATION

It should be noted that the dispacements U_{ij} as given by Equation (10) are computed for dynamic degrees of freedom only. Therefore, it is necessary to compute the displacements for all degrees of freedom prior to calculating the load (i.e., shears and moments). Otherwise, the calculated load will be erroneous. In addition, the modal values need to be combined in some statistical way as described below.

Corresponding to each mode shape $\{\phi_i\}$ there exists an internal load distribution in the system. Let $\{P_i\}$ be the vector of loads corresponding to the displaced configuration $\{\phi_i\}$. The terms in $\{P_i\}$ are computed from the first (axial load and torque), second (bending moments), and third (shears) derivatives of the displacement function (or mode shape) $\{\phi_i\}$. The individual modal loads are then summed in the same manner as for system total displacement to yield the internal loads for the complete structural system.

Appendix F
ACRONYMS

The acroynms used in this report are noted below for the convenience of the reader.

ADS	Automatic Depressurization System
AEC	Atomic Energy Commission
AFW	Auxiliary Feedwater
ANSI	American National Standards Institute
ASME	American Society of Mechanical Engineers
BWR	Boiling Water Reactor
CFR	Code of Federal Regulations
DBA	Design Basis Accident
ECCS	Emergency Core Cooling System
EERI	Earthquake Engineering Research Institute
EPRI	Electric Power Research Institute
FSAR	Final Safety Analysis Report
GDC	General Design Criteria
HELB	High Energy Line Break
HPCI	High Pressure Coolant Injection
IEEE	Institute of Electrical & Electronics Engineers
IGSCC	Intergranular Stress Corrosion Cracking
LOCA	Loss of Coolant Accident
LPCI	Low Pressure Core Injection
MSIV	Main Steam Isolation Valve
NPSH	Net Positive Suction Head
NSF	National Science Foundation
NSSS	Nuclear Steam Supply System
OBE	Operating Basis Earthquake
PORV	Power Operated Relief Valve
PSD	Power Spectral Density
PWR	Pressurized Water Reactor
R	Rad (a unit of radiation)
RCFC	Reactor Containment Fan Cooling
RCIC	Reactor Core Isolation Cooling
RHR	Residual Heat Removal
RG	Regulatory Guide
RRS	Required Response Spectrum
S&M	Surveillance and Maintenance
SEP	Systematic Evaluation Program
SQRT	Seismic Qualification Review Team
SQUG	Seismic Qualification Utility Group
SRV	Safety Relief Valve
SSE	Safe Shutdown Earthquake
SSRAP	Senior Seismic Review Advisory Panel
TRS	Test Response Spectrum
ZPA	Zero Period Acceleration