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Aging and Service Wear of Auxiliary Feedwater Pumps for PWR Nuclear Power Plants

Volume 1. Operating Experience and Failure Identification

M. L. Adams
E. Makay

Work Performed for
U.S. Nuclear Regulatory Commission
Office of Nuclear Regulatory Research
Under DOE Interagency Agreement No. 40-551-75
NRC FIN No. B0828

OPERATED BY
MARTIN MARIETTA ENERGY SYSTEMS, INC.
FOR THE UNITED STATES
DEPARTMENT OF ENERGY

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Superintendent of Documents
U.S. Government Printing Office
Post Office Box 37082
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and

National Technical Information Service
Springfield, VA 22161

NUREG/CR-4597
Volume 1
ORNL-6282/V1
Dist. Category RV

AGING AND SERVICE WEAR OF AUXILIARY FEEDWATER PUMPS FOR PWR NUCLEAR POWER PLANTS

Volume 1. Operating Experience and Failure Identification

M. L. Adams E. Makay

Manuscript Completed — June 24, 1986
Date Published — July 1986

Report Prepared by
Energy Research and Consultants Corporation
900 Overton Avenue
Morrisville, Pennsylvania 19067
under
Purchase Order No. 11X-28630V
for Oak Ridge National Laboratory

Work Performed for
U.S. Nuclear Regulatory Commission
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under Contract No. DE-AC05-84OR21400

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ACKNOWLEDGMENTS

The authors gratefully acknowledge the continuing support and counsel of the Nuclear Regulatory Commission Nuclear Plant Aging Research Program manager, J. P. Vora, in the planning and implementing of this study. W. L. Greenstreet of Oak Ridge National Laboratory directed the work on this report, which was substantially configured through his direction and advice.

The information, insights, conclusions, and recommendations contained in this document are based primarily on the design, troubleshooting, and redesign experience of Energy Research and Consultants Corporation in power plants for electric utility companies. The authors gratefully acknowledge the cooperation of several Electric Utility Companies, in particular, operating personnel at the Palo Verde nuclear plant of Arizona Public Service Company.

In addition, we wish to express our thanks to the following pump manufacturers for their openness in discussing their points of view: Ingersoll-Rand, Byron Jackson, Pacific Pump Division of Dresser Clark, and Bingham Pump Companies.

SUMMARY

The primary focus of this report is on factors that are important to aging and service wear of auxiliary feedwater pumps (AUXFPs). These pumps are from the generic family of multistage high-head centrifugal pumps and are basically small boiler feed pump designs used in small capacity, mostly older fossil fuel electric generating plants.

A general description of AUXFPs is provided and includes illustrations, construction and configuration details, defined equipment boundaries, functional requirements, and materials of construction. Technical specification requirements and operating experience topics are summarized. Operational stressors are categorized, and a detailed stressor list is provided, component-by-component.

Failure modes are defined as follows: (1) failure to operate, (2) failure to operate as required, and (3) external leakage. Failure causes are identified in general terms and subsequently described in more specific terms in a section on failure cause analysis. The single most important factor relevant to AUXFP potential failures is the presence of large hydraulic dynamic forces within such pumps, particularly at flow rates substantially different than the best-efficiency flow. Correspondingly, determining safe minimum AUXFP flow rates (i.e., bypass or recirculation flow) is a very important task. This topic is discussed at length in Sect. 6 and in Appendix D.

Methods for detecting failure modes and differentiating between failure causes are described. In addition, measurable parameters (including functional indicators) are identified for potential use in detecting and monitoring degradation and for tracking degradation trends.

Parameters for identifying failure causes of pumps include

Vibration	Clearance
Delivered flow	Rotor axial position
Rotational speed	Leakage rate
Bearing temperature	Appearance
Transmitted torque	Local shaft temperature
Noise	Bolt torque

The appropriateness and utility of these and other parameters will be addressed in subsequent phases of the AUXFP investigation.

At present, most AUXFP installations contain no monitoring devices except for flow and head measurement, in contrast to continuously running power plant equipment. Furthermore, present surveillance practice consists primarily of starting each AUXFP at bypass flow once every 1 to 3 months for a short-duration test to verify operational readiness. Thus, the establishment and correlation of AUXFP operating parameters (e.g., vibration and bearing temperatures) to wear and aging criteria are not addressed in nearly all installations as presently configured and instrumented. Actions are detailed to address these present weaknesses. These actions include (1) disassembly and inspection and component renewal at refueling intervals and (2) parameter monitoring and instrumentation retrofits. These and other important factors are addressed in Sect. 9.

Appendix D provides a major background section on engineering information and design factors critically important to AUXFP reliability. The Bibliography contains an extensive list of over 40 publications and reports, which have been prioritized to aid in selecting those that are most critical to understanding the potential operating problems and failure modes of AUXFPs.

AGING AND SERVICE WEAR OF AUXILIARY FEEDWATER PUMPS FOR PWR NUCLEAR POWER PLANTS

Volume 1. Operating Experience and Failure Identification

M. L. Adams* E. Makay*

ABSTRACT

This report was produced under the Detection of Defects and Degradation Monitoring element of the Nuclear Plant Aging Research Program. Typical auxiliary feedwater pump (AUXFP) configurations are described in terms of configuration details, materials of construction, operating requirements, and modes of operation. AUXFP failure modes and causes due to aging and service wear are identified and explained, and measurable parameters (including functional indicators) for potential use in assessing operational readiness, establishing degradation trends, and detecting incipient failures are given.

A series of measures to correct present deficiencies in surveillance, monitoring, and in-service testing practices is discussed. The main body of the report is supplemented by a number of relevant appendixes; in particular, a major appendix is included on engineering and design information useful to assess AUXFP operational readiness.

1. INTRODUCTION

1.1 Background

The Office of Nuclear Regulatory Research of the Nuclear Regulatory Commission (NRC) has instituted studies aimed at understanding the time-related degradation (aging) of nuclear power plant systems and equipment, assessing the effectiveness of methods of inspection and surveillance to monitor such degradation, and establishing guidelines for maintenance. This study is one in a series intended to provide technical bases to assess the ongoing operational safety of operating plants. The strategy[1] followed can be used by others interested in analyses of equipment in nuclear applications.

This report addresses time-related degradation of pressurized-water reactor (PWR) power plant auxiliary feedwater pumps (AUXFPs). Because failures of these components can reduce the amount of feedwater available

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for removing heat when the usual feedwater supply is unavailable, such failures can result in altered safety margins for PWR systems.

1.2 Project Scope

This report is Volume 1 of a three-part report to be prepared on AUXFPs. The contents of the three volumes are summarized below.

Volume 1 — Operating experience and failure identification (Phase 1)

1. Background information on AUXFPs — boundary of AUXFPs to be studied, types, functional requirements, and materials of construction;
2. Reviews of regulatory requirements, guides, and standards;
3. Summary of operational and environmental stressors;
4. Summary of operating experience;
5. Manufacturers' input; and
6. State-of-the-art aging and service wear monitoring and assessment.

Volume 2 — Aging assessments and evaluation of monitoring methods (Phase 2)

1. Results from completion of comprehensive aging assessment based on postservice examination and tests of aged components and in-situ assessments,
2. Identification of monitoring techniques and review of information produced, and
3. Evaluations of monitoring methods.

Volume 3 — Analysis and recommendations

1. Value impact analysis, and
2. Recommendations for guidelines for monitoring methods and maintenance.

One of the objectives of the Phase 1 research effort is to provide baseline information for use in subsequent phases of the study. To prepare this report, operating experience, manufacturers' information, and information derived from design, troubleshooting, and redesign experience were reviewed to identify (1) failure modes and causes resulting from aging and service wear of AUXFPs and (2) measurable parameters (including functional indicators) with potential for quantifying and tracking degradation. These parameters are to be applicable for detecting and establishing time-dependent degradation trends before loss of function.

1.3 Definitions

For the purpose of this report, the following definitions apply.

Failure mode — the way a component does not perform a function for which it was designed (e.g., fails to actuate or leaks to outside).

Failure cause — degradation (the presence of a defect) in a component that is the proximate cause of its failure (e.g., bent shaft, loss of lubricant, or loosening of a bolt).

Failure mechanisms — the phenomena that are responsible for the degradation present in a given component at a given time. Frequently, several failure mechanisms are collectively responsible for degradation (synergistic influences). One major failure mechanism, where identified, has been called the "root cause." Generic examples of failure mechanisms (and of root causes) include aging, human error, or seismic events.

Aging — the combined cumulative effects over time of internal and external stressors acting on a component, leading to degradation of the component, which increases with time. Aging degradation may involve changes in chemical, physical, electrical, or metallurgical properties, dimensions, and/or relative positions of individual parts.

Normal aging — aging of a component that has been designed, fabricated, installed, operated, and maintained in accordance with specifications, instructions, and good practice and that results from exposure to normal stressors for the specific application. Normal aging should be taken into account in component design and specification.

Measurable parameters — physical or chemical characteristics of a component that can be described or measured directly or indirectly and that can be correlated with aging. Useful measurable parameters are those that (1) can be used to establish trends of the magnitude of aging associated with each failure cause, (2) have well-defined criteria for quantifying the approach to failure, and (3) are able to discriminate between the degradation that leads to failure and other observed changes.

Inspection, surveillance, and condition monitoring (ISCM) — the spectrum of methods and hardware for obtaining qualitative or quantitative values of a measurable parameter of a component. The methods may be periodic or continuous, may be in-situ, or may require removal and installation in a test stand or disassembly and may involve dynamic or static measurements.

2. BACKGROUND INFORMATION

2.1 Principal Types of Auxiliary Feedwater Pumps

AUXFPs for PWR systems are basically small boiler feed pumps used in small-capacity mostly older, fossil fuel electric generating plants. Thus, they retain most of the design features, plus the potential operating and reliability problems inherent in the feedwater pumps. A comprehensive bibliography of publications and reports document extensive field troubleshooting experience and analyses that are most relevant to AUXFP machinery. This bibliography also represents the several years of experience upon which the contents of the report rest.

AUXFP configurations are all multistage (from 6 to 12) with low-specific-speed, high head-per-stage (HHPS) impellers, having either vane-diffuser or volute collection chambers at the discharge section of each stage. The overall construction is always an axially (horizontal) split outer casing with internal stage-to-stage connecting flow passages (see Figs. 1–3).

More specific details of configuration vary considerably from vendor to vendor. Even within the array of available designs from a single manufacturer, there can be a considerable variety of construction details. The main factor that determines these more specific configuration details is the design approach used in handling the very large axial thrust forces on the rotor, which arise as a natural by-product in HHPS centrifugal pumps. A second factor is the design approach used to avoid cavitation in the inlet (suction), that is, single-suction (Figs. 2 and 3) or double-suction (Fig. 1) inlet stage.

Some manufacturers use a configuration with all stages positioned in the same axial direction (Fig. 3, called "in-line" type). Other manufacturers use a configuration with the stages on one-half of the shaft axially opposed to those on the other half of the shaft (Figs. 1 and 2, called "opposed" type) for thrust cancellation. The basic difference between these two configurations involves the previously referenced different design approaches to handling the very large axial thrust loads inherent in multistage HHPS centrifugal pumps. Opposed type configurations (i.e., thrust cancellation approach) are often designed with only a thrust bearing. In-line type configurations are always designed with an additional thrust balancer of either "balancing drum" type or "balancing disk" type (see Appendix D).

Plants have traditionally been configured with two AUXFPs per pressurized-water reactor (PWR). In most plants, one AUXFP is steam turbine driven, and the other is electric motor driven. The steam-turbine-driven unit has the advantage of variable speed, and the maximum design operating speeds are generally in the 4000- to 5000-rpm range. The motor-driven unit is usually designed for two-pole induction motor speed and powered from either one of two standby diesel-driven generators. Newer plants may also have a third "backup" AUXFP that is electric motor driven either from main line power or switchable to the diesel-driven generator bus. Thus, to provide very high standby reliability of the AUXFP system, the number of pumps employed (now three) and the number of

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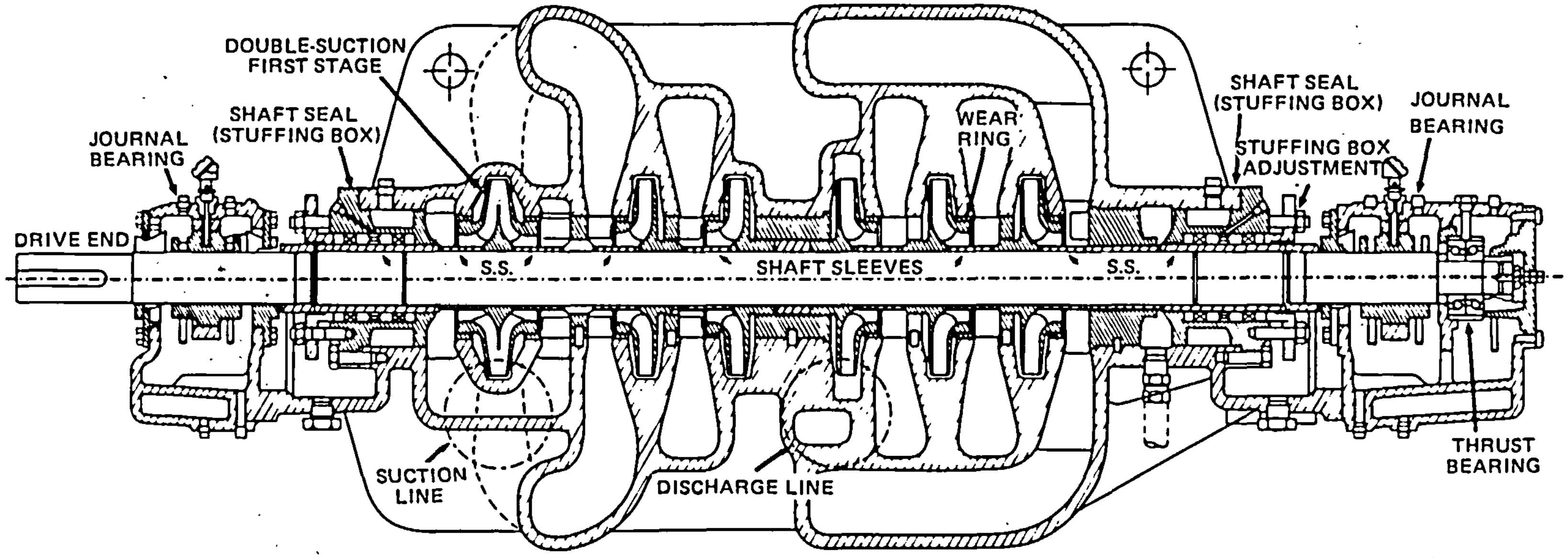

Fig. 1. Double-suction, opposed-impellers AUXFP.

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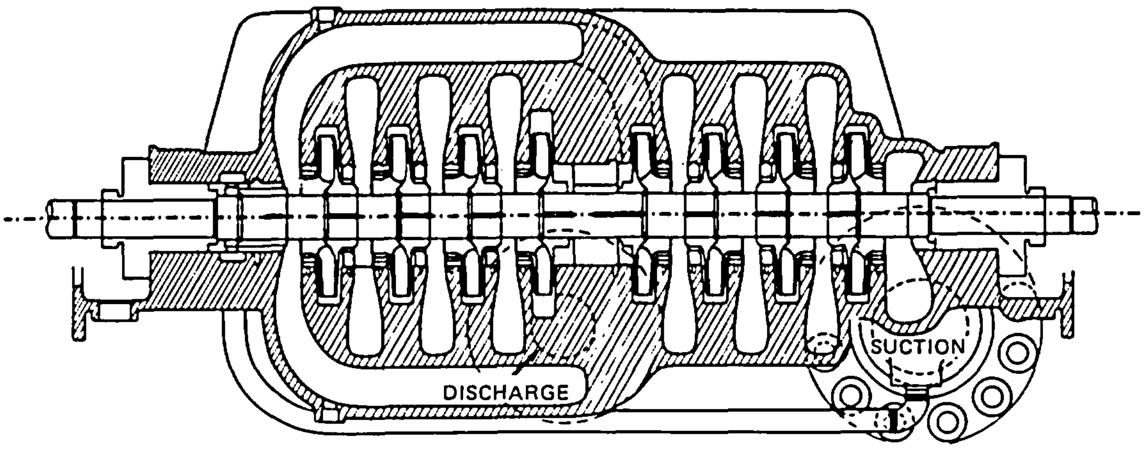

Fig. 2. Single-suction, opposed-impellers AUXFP.

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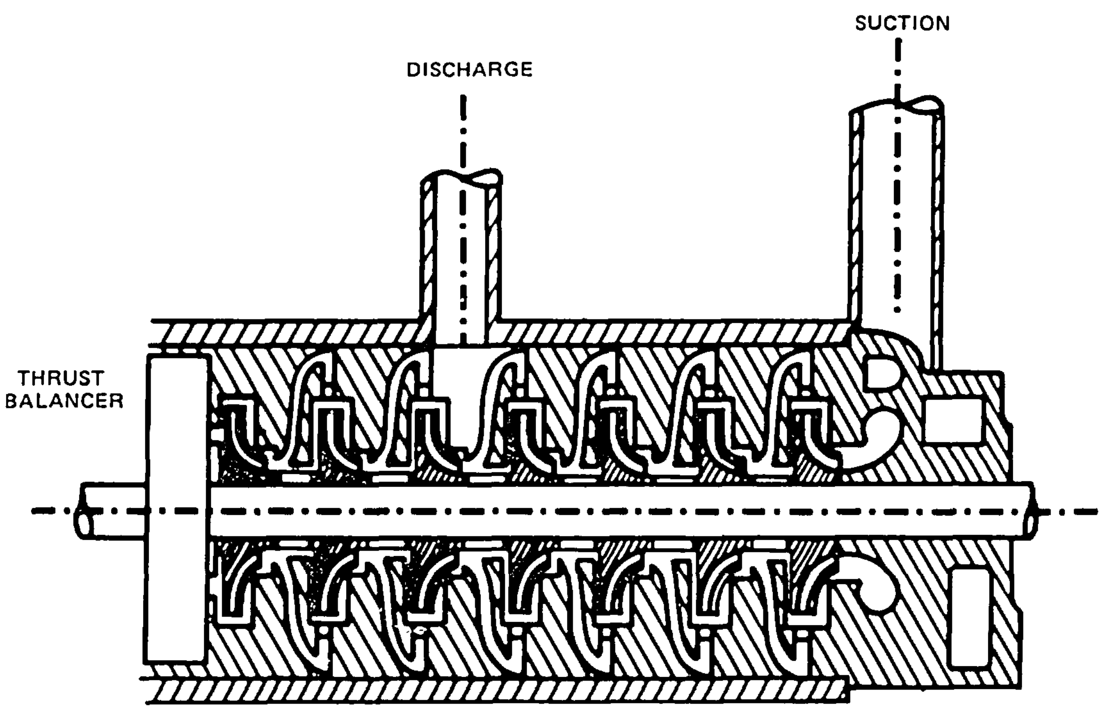

Fig. 3. Single-suction, in-line-impellers AUXFP.

separate power sources (i.e., steam, diesel-generator, main line) are triple redundant. The overall approach that now is emerging in the newest plants is to have the backup AUXFP be a non-safety-related, non-nuclear class (nonclass) pump (thus cheaper) and to use it for all normal plant startup, shutdown, and nonemergency service. However, the Plant Technical Specification requires readiness testing of nuclear class (N-class) AUXFPs at least every 3 months.

The rated discharge pressure for nuclear plant AUXFPs ranges from 1100 to 1500 psig. The rated capacity ranges from 200 to 1000 gal/min. However, rated head and capacity are generally pegged to the best-efficiency-point (BEP) flow at a given speed, and AUXFPs are required to operate over a wide range of percent BEP flow. They are typically required to operate from 10 to 140% of BEP capacity. This is considered a rather abusive operating requirement from the point of view of acceptable practice for main feedwater pumps.

2.2 Equipment Boundaries

For present purposes, the primary focus is on the pump per se, the drivers being outside the scope of this report. However, it is appropriate to give consideration to defining a system, that is, equipment boundaries.

The pump is mechanically coupled to its driver (steam turbine or electric motor). The pump-driver assembly is typically mounted on a common fabricated steel base that is stiffly connected to some structural "floor" section within the plant. Furthermore, the pump is hydraulically and mechanically connected to a fluid flow circuit through its inlet (suction) and discharge nozzles. For a turbine-driven pump, the turbine inlet is, of course, piped into a steam flow circuit and the outlet to a condenser. For a motor-driven pump, the motor is electrically connected to a 60-Hz, 3-phase power supply. Thus, the equipment boundaries will be taken to include the driver and pump on a common base and with the boundary taken to include all inlet and discharge nozzles as well as direct electrical connections. However, aging, wear, and failures of the driver and connecting steam or electrical power lines are not addressed in this report.

Failures outside this defined boundary will also not be considered in this report. The small steam turbines and the electric motors used to drive AUXFPs are standard designs used in many other applications as well and as yet do not appear to exhibit any aging or wear factors that are unique to the AUXFP application. Based on operating experience on similar pumping systems for other applications, driver failure can be caused by a massive failure of the pump. Primarily, this would be a pump failure resulting in loss of thrust balancer and thrust bearing load-carrying capacity. In such a situation, depending upon the specific pump and coupling design, there is a potential for subjecting the driver (i.e., its thrust bearing) to damaging load levels. Excessive pump vibration is another factor that could accelerate wear and aging of the turbine or motor driver.

2.3 Functional Requirements

AUXFP functional requirements may vary somewhat from plant to plant and also vary as a function of plant vintage. The following itemization of functional requirements includes the typical ranges for quantifiable parameters where appropriate.

2.3.1 General requirements

1. Supply feedwater to the steam generators under plant startup, normal shutdown, hot standby, and emergency conditions;
2. meet automatic starting requirements;
3. operate under normal and accident conditions of temperature, pressure, humidity, radiation, and available net positive suction head (NPSH); and
4. withstand seismic loadings without loss of function.

2.3.2 Specific requirements

1. Capable of producing the rated flow against the rated head within 20 s (typical) after actuation,
2. total of 1000 service-life operating cycles [includes all items in 2.3.1 (1)],
3. AUXFP operating cycle including up to 10 h of continuous pumping,
4. operation of pump at shutoff (upset condition) for up to 10 min,
5. a plant life of 40 years,
6. a steadily rising head-capacity characteristic with a shutoff head in the range of 115 to 130% of best-efficiency head at a given speed,
7. an efficiency characteristic commensurate with state-of-the-art hydraulic design, and
8. capable of operating at any point on the head-capacity curve, from minimum flow to full run-out condition (i.e., maximum pump flow possible against system resistance) during and following safe reactor shutdown at earthquake condition.

2.3.3 Miscellaneous

1. To be available for operation, except when taken off-line for maintenance or testing, and
2. to be supplied with appropriate protective devices such as overspeed trip.

2.4 Materials of Construction

Precise material designations vary somewhat from vendor to vendor. Also, in a number of AUXFP components, the current material or method of forming may not be the optimum for this application. The primary reason for this is that purchaser specifications are not always optimally explicit, potentially resulting in competing vendors using lower cost options.

Thus, the following paragraphs not only summarize current practice but also contain suggested alternatives that should be studied.

The two parts of the split casing, including inlet and discharge nozzles, are made of a cast carbon or stainless steel. The most common is the cast equivalent of 416 stainless steel, designated when cast as CA6NM.

The shaft is currently machined from 400-series stainless steel bar stock by all vendors. Typical material designations are 410, 414, and 416 stainless steel. A more durable alternative for consideration is to make the shaft from a forging.

Impellers are made of high-chromium alloy steels and are sand cast by all vendors. Commonly used alloys are 13-5 (i.e., 13% Cr, 5% Ni), 15-5, and 17-4 Ph. Here again, current practice may not have been optimally constrained by purchaser specifications. That is, impellers could be specified as precision cast to provide maximum quality both in terms of structural integrity and dimensional control of critical hydraulic passages.

The diffusers and stage-to-stage return channel components are also sand cast with the material varying from vendor to vendor, for example, stainless steel 440A, 440B, and 17-4 Ph. As in the case of the impellers, optimum dimensional control of critical hydraulic passages suggests that precision cast or milled diffusers and return channels should be considered.

The coupling (used to connect pump to driver) materials vary considerably with coupling type and manufacturer. The type of coupling used in nearly all installations is the gear type. However, tooth clearances in such couplings, always produce some measurable rotor unbalance that is not correctable. Furthermore, lubrication is required to prevent wearout in service. For these reasons, the dry flexible diaphragm type of coupling should be considered.

The main pressure-closure retaining bolts (studs) are made of high tensile strength Cr-Mo steels, suitable for the high prestressing necessary to ensure a leaktight and structurally sound closure at the casing joint.

Various shaft sleeves are made of the same steel alloys as the shaft. Shaft-sleeve retaining bolts are also made of similar steel alloys as the shaft.

Balancing disks, balancing drums, and wear rings are typically machined from stainless steel alloy material such as 416 and 420. A considerable enhancement to reliability can be ensured if the material is designated as 420F, which is a "free machining nongalling" stainless steel with a trace of phosphorus (0.17%) or sulfur (0.3%).

Oil-film journal bearings and thrust bearings are typically constructed of carbon steel shells with bearing surfaces lined with babbitt white metal (usually tin base for best results) that is cast into the preheated bearing shell and finished machined to the specified final dimensions. Rolling contact bearings, which are available to take radial and axial loads, are constructed of various specialty steels typically used in such bearings. These bearings are case hardened to provide very high strength outer (surface) properties and a more ductile inner core of the rolling contact elements.

3. TECHNICAL SPECIFICATION REQUIREMENTS*

Testing requirements for AUXFPs in nuclear power plants are described in the plant Technical Specifications that state the overall in-service inspection requirements for safety-related systems and components. In-service inspection and testing of ASME Class 1, 2, and 3 components are to be in accordance with Sect. XI of the *ASME Boiler and Pressure Vessel Code*.

The Sect. XI testing requirements for pumps involve measuring inlet pressure, pressure differential across the pump, flow rate, and vibration amplitude; all are measured while operating at specific design conditions. The test quantities are compared with reference values.

The *ASME Code* Sect. XI contains surveillance intervals, frequencies, and test requirements for Code Class 1, 2, and 3 systems. A review of Subsection IWP, In-service Testing of Pumps in Nuclear Power Plants, is given in Appendix A.

The purpose of Technical Specifications is not specifically to monitor degradation of performance but to ensure operability of components and systems within specified limits required to perform the desired safety function.

---

*Work performed by G. A. Murphy, ORNL Nuclear Operations Analysis Center.

4. SUMMARY OF OPERATING CONDITIONS AND STRESSOR INFLUENCES

4.1 Description of Typical Operating Regimes

4.1.1 Regimes

AUXFPs are used to supply feedwater to the steam generators under plant startup, shutdown, and emergency conditions. These pumps have a BEP flow at any operating speed, and if used in a continuous operating mode, normal delivered flow is between 50 and 120% of the BEP flow. However, the AUXFP application is by definition a transient operational mode, that is, startup,* shutdown,* and emergency. Thus, there are basically two operating regimes: (1) standby and (2) normal. Normal includes any flow from shutoff or bypass flow to full run-out flow. Alternately, if the term "normal operation" pertains to that regime in which the pump resides most of the time, then normal operation would be the standby condition, and any condition under which the AUXFP is pumping would be considered off-normal.

It is therefore more rational to categorize operating regimes in a manner that is most relevant to aging and wear factors. For that reason, the following are the defined operating regimes: (1) standby (i.e., 0 flow), (2) 0 to 50% BEP flow, (3) 50 to 120% BEP flow, and (4) 120% BEP flow to full run-out flow (typically 150% BEP flow). The relative importance and relevance of some operational stressors will be different for each of these defined operating regimes.

4.1.2 Pump operating information for each regime

The following operating parameter ranges are based on a representative number of plants listed in Table 1.

1. BEP flow, 200 to 1000 gal/min;
2. BEP discharge pressure, 1100 to 1500 psig;
3. available suction pressure, −2.6 to 60 psig;
4. speed, 3560 to 4400 rpm;
5. shutoff head, 115 to 130% BEP head; and
6. pumped water temperature, 60 to 125°F.

These BEP parameter values, when keyed with the operating regimes defined in the previous section, provide the essential quantitative information for each regime.

---

*In some plants, startup and normal shutdown are handled by a separate (nonclass) motor-driven pump, frequently referred to as the auxiliary feedwater backup pump.

Table 1. Examples of typical AUXFP BEP operating parameters

Power plant	Number of stages	Speed (rpm)	Flow (gal/min)	Suction (psig)	Discharge (psig)
TVA Sequoyah	5	3950	920	25	1120
TVA Sequoyah	9	3570	440	25	1252
North Anna 1 and 2	6	4200	735	−2.6	1216
North Anna 1 and 2	8	3560	370	−2.6	1216
TVA Watts Bar 1 and 2	6	3850	1000	10	1277
TVA Watts Bar 1 and 2	9	3577	500	10	1277
Seabrook 1 and 2	9	3577	500	10	1277
Shearon	9	4400	850	0	1270
Shearon	9	3550	425	0	1270
Ginna 1	10	3560	200	23	1475
Maine Yankee	5	4400	530	0	1095
Maine Yankee	8	3575	500	0	1095
Donald C. Cook Station	6	4350	900	25	1195
Donald C. Cook Station	8	3560	450	25	1195
Beaver Valley Station	8	3560	370	0	1165
Arkansas Power and Light Co.	9	3560	780	60	1172
Three Mile Island	8	3560	470	23	1128
Three Mile Island	6	4250	940	23	1128

4.1.3 Pump external environment

1. Indoor installation;
2. atmospheric pressure;
3. temperature, 60 to 105°F;
4. 40-year cumulative radiation, 200 rads;
5. pumped fluid not expected to be radioactive; and
6. maximum relative humidity, 100%.

4.2 Stressor Description

4.2.1 Mechanical

1. Torque transmitted loads (static and dynamic),
2. assembly (fastener) loads,
3. rotor-dynamic loads (e.g., unbalance),
4. piping forces,
5. seismic loads, and
6. vibration (for all sources).

4.2.2 Hydraulic

1. Hydraulic loads (static and dynamic),
2. fluid impingement,
3. internal pressure, and
4. cavitation.

4.2.3 Tribological*

1. Rubbing between rotating and nonrotating members [potentially more severe in the presence of 4.2.3 (7)],
2. bearing lubricant breakdown (viscosity and various chemical additive breakdowns),
3. surface fatigue (life limiting for rolling contacts and gears),
4. contamination and degradation of lubricant,
5. starts and stops,
6. fretting, and
7. surface oxide abrasive formation (see 4.2.4).

4.2.4 Chemical

Corrosion (oxidation) of 400-series stainless steels through chemical reaction with stagnant water can produce an oxide surface scale. Chlorides and other feedwater impurities have occurred from turbine condenser leakage and can increase the rate of this type of corrosion. Furthermore, this chemical process can result in atomic hydrogen diffusing into the metal surface to form molecular hydrogen that can cause surface "blisters."

Other chemical effects that may have influence are stress corrosion cracking, pitting, and crevice corrosion. Stress corrosion cracking may occur, depending on the impurities present in the water, but it is very unlikely to happen at the operating temperatures of these pumps. Pitting is possible, depending on the type of impurities in the water and their concentrations; potential pitting damage is projected to be small for AUXFPs, however. Crevice corrosion is another possible factor, but because of the low impurity content and the relatively low temperature of the water it is not likely to be significant.

4.2.5 Low-relevance factors

From the point-of-view of aging and wear considerations, the following stressor categories are not significant to AUXFPs:

1. thermal,
2. radiation, and
3. environmental [except for earthquakes, covered in 4.2.1 (5)].

4.3 Stressor Influence

4.3.1 Pump parts and components

The pump has been subdivided into five major segments: rotating elements, nonrotating internals, pressure-containment casing, mechanical

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*Tribology is now the unifying label for that field that encompasses friction, wear, lubrication, and machinery components affected by same.

subsystems, and support base. Each of these segment categories is further broken down into individual components listed below.

Pump segment	Parts	
Rotating elements	Shaft Impellers Miscellaneous spacers	Thrust runners* Fasteners
Nonrotating internals	Diffusers or volutes Return channels	Wear surfaces Fasteners
Pressure-containment casing	Upper casing Lower casing Fasteners	Suction and discharge nozzles
Mechanical subsystems	Thrust bearing Radial bearings Shaft seals	Thrust balancer Coupling Fasteners
Support	Base frame	Fasteners

The particular stressors that are significant to each part are enumerated in the following subsections, segment-by-segment. The relative importance of some stressors is a significant function of the particular operating mode. The single potentially important chemical stressor previously listed (Sect. 4.2.4) is carried through this document as a tribological stressor because its relevance to wear, aging, and failure modes stems from the abrasive surface oxides and blisters that might potentially be formed while the pump is on standby [see 4.2.3 (7)].

4.3.1.1 Rotating elements

Shaft

Mechanical stressors: Transmitted torque, fastener loads, rotor-dynamic loads

Hydraulic stressors: Hydraulic loads

Tribological stressors: Rubbing between rotating and nonrotating members, bearing lubricant breakdown, dirt in lubricant, starts and stops, surface oxide abrasives

*A thrust runner (also called a thrust collar) is the rotating part captured by a double-acting oil-film thrust bearing. In addition, the rotating part of a balancing disk assembly is called a thrust runner. In many AUXFPs, these two components are superceded by the use of a ball bearing for carrying axial thrust loads.

Impellers

Mechanical stressors: Transmitted torque, fastener loads, rotor-dynamic loads

Hydraulic stressors: Hydraulic loads, cavitation, fluid impingement

Tribological stressors: Rubbing between rotating and nonrotating members, fretting, surface oxide abrasives

Miscellaneous spacers

Mechanical stressors: Fastener loads, rotor-dynamic loads

Hydraulic stressors: Hydraulic loads

Tribological stressors: Rubbing between rotating and nonrotating members, fretting, surface oxide abrasives

Thrust runners

Mechanical stressors: Fastener loads, rotor-dynamic loads

Hydraulic stressors: Hydraulic loads

Tribological stressors: Rubbing between rotating and nonrotating members, bearing lubricant breakdown,* dirt in lubricant,* starts and stops, fretting, surface oxide abrasives

Fasteners

Mechanical stressors: Transmitted torque, assembly loads, rotor-dynamic loads

Hydraulic stressors: Hydraulic loads

Tribological stressors: Fretting

4.3.1.2 Nonrotating Internals

Diffusers or volutes

Mechanical stressors: Fastener loads

Hydraulic stressors: Hydraulic loads, fluid impingement, cavitation

Tribological stressors: Fretting

Return channels

Mechanical stressors: Fastener loads

Hydraulic stressors: Hydraulic loads, fluid impingement

Tribological stressors: Fretting

*May not apply since most, if not all, AUXFP thrust bearings are of the rolling contact type, not the hydrodynamic oil-film type.

Wear surfaces

Mechanical stressors: Assembly loads, rotor-dynamic loads

Hydraulic stressors: Fluid impingement

Tribological stressors: Rubbing between rotating and nonrotating members, starts and stops, surface oxide abrasives

Fasteners

Mechanical stressors: Assembly loads, vibration

Hydraulic stressors: Hydraulic loads

Tribological stressors: None

4.3.1.3 Pressure-containment casing

Upper casing

Mechanical stressors: Assembly loads, piping forces, seismic loads, vibration

Hydraulic stressors: Hydraulic loads, fluid impingement, internal pressure

Tribological stressors: None

Lower casing

Mechanical stressors: Assembly loads, piping forces, seismic loads, vibration

Hydraulic stressors: Hydraulic loads, fluid impingement, internal pressure

Tribological stressors: None

Suction and discharge nozzles

Mechanical stressors: Assembly loads, piping forces, seismic loads, vibration

Hydraulic stressors: Hydraulic loads, fluid impingement, internal pressure

Tribological stressors: None

4.3.1.4 Mechanical subsystems

Thrust bearing

Mechanical stressors: Assembly loads, vibration

Hydraulic stressors: Hydraulic loads

Tribological stressors: Surface fatigue, dirt

Radial bearings

Mechanical stressors: Assembly loads, rotor-dynamic loads

Hydraulic stressors: Hydraulic loads

Tribological stressors: Rubbing between rotating and nonrotating members, bearing lubricant breakdown, surface fatigue, dirt in lubricant

Thrust balancer

Mechanical stressors: Assembly loads, vibration

Hydraulics stressors: Hydraulic loads

Tribological stressors: Rubbing between rotating and nonrotating members, starts and stops, surface oxide abrasives

Shaft seals*

Mechanical stressors: Assembly loads, vibration, improper adjustment (overtightening) to correct for normal wear of packing

Hydraulic stressors: Internal pressure

Tribological stressors: Rubbing between rotating and nonrotating members, starts and stops, surface oxide abrasives

Coupling

Mechanical stressors: Transmitted torque, assembly loads, rotor-dynamic loads

Hydraulic stressors: None

Tribological stressors: Lubricant breakdown, surface fatigue, dirt in lubricant, fretting

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*Mechanical or stuffing-box seals.

Fasteners

Mechanical stressors: Transmitted torque, assembly loads, rotor-dynamic loads, vibration

Hydrodynamic stressors: Hydraulic loads

Tribological stressors: Fretting

4.3.1.5 Support

Base frame

Mechanical stressors: Assembly loads, piping forces, seismic loads, vibration

Hydraulic stressors: None

Tribological stressors: None

Fasteners

Mechanical stressors: Assembly loads, seismic loads, vibration

4.3.2 Operating regimes

The influence of stressors is described in Tables 2–6. Each of these tables pertains to a specific pump segment as previously defined in Sect. 4.3.1. Each segment is broken down into specific parts, and each part is briefly commented upon in regard to stressor influence for each operating regime. In Sect. 4.1.1, operating regimes were defined or categorized in a manner that is most relevant to aging and wear factors: (1) standby, (2) 0 to 50% BEP flow, (3) 50 to 120% BEP flow, and (4) 120% BEP flow to full run-out flow (typically 150% BEP flow).

The relevant stressor influence identified for the standby condition is abrasive surface oxide and blister formation. However, the potential wear mechanism resulting from this would arise when the pump is operated, not at standby per se. Also, this is the single significant chemical stressor identified in Sect. 4.2.4 but is subsequently herein referenced as a tribological stressor [see 4.2.3 (1) and (7)] due to its relevance primarily as a potential wear accelerator at close running clearances. Therefore, Tables 2–6 do not list this as a chemical stressor nor reference the standby condition.

The approach to AUXFP usage adopted in at least one of the newest plants (Palo Verde) is the use of an electric-motor-driven nonclass pump (of similar if not same design as AUXFPs) for startup and shutdown service. This approach renders the AUXFPs on standby all the time, except for occasional short duration testing. This approach, which clearly tends to preserve the AUXFPs for true emergency service, probably also reduces operating costs. That is, when comparing two pumps of the same design where one is purchased as an N-class pump and the other as a nonclass pump, the class pump and its related spare parts are priced considerably higher by vendors than the corresponding nonclass pump.

Table 2. Stressor influence on rotating elements

Part	Operating regime (% BEP flow)	Stressor influence and remarks[a]		
		Mechanical	Hydraulic	Tribological
Shaft	0–50	(1)	(3) Hydraulic dynamic loads	(3) Rubbing at close-running clearances due to hydraulic-load-induced rotor vibration
	50–120	(1)	(1)	(1)
	120–R.O.	(1)	(3) Hydraulic dynamic loads	(3) Same as 0–50
Impellers	0–50	(1)	(3) Cavitation	(3) Rubbing at wear surfaces
	50–110	(1)	(1)	(2) Same as 0–50
	120–R.O.	(1)	(3) Same as 0–50	(3) Same as 0–50
Miscellaneous spacers	0–50	(1)	(1)	(3) Rubbing at shaft seal sleeve due to rotor vibration
Thrust runners	0–50	(1)	(3) Hydraulic dynamic loads	(3) Rubbing due to hydraulic loads
	50–120	(1)	(1)	(1)
	120–R.O.	(1)	(3) Same as 0–50	(3) Same as 0–50
Fasteners	0–50	(1)	(2) Hydraulic dynamic loads	(1)
	50–120	(1)	(1)	(1)
	120–R.O.	(1)	(2) Same as 0–50	(1)

[a]To indicate a relative measure of importance of each stressor category, an approximate quantifier is shown as either low (1), medium (2), or high (3).

[b]R.O. = run-out flow.

Table 3. Stressor influence on nonrotating internals

Part	Operating regime (% BEP flow)	Stressor influence and remarks[a]		
		Mechanical	Hydraulic	Tribological
Diffusers or volutes	0–50	(1)	(3) Unsteady flow forces	(1)
	50–120	(1)	(2) Same as 0–50	(1)
	120–R.O.	(1)	(3) Same as 0–50	(1)
Return channels	0–50	(1)	(2) Unsteady flow	(1)
	50–120	(1)	(1)	(1)
	120–R.O.	(1)	(2) Same as 0–50	(1)
Wear surfaces	0–50	(1)	(1)	(3) Rubbing due to rotor vibration
	50–120	(1)	(1)	(2) Same as 0–50
	120–R.O.	(1)	(1)	(3) Same as 0–50
Fasteners	0–50	(1)	(2) Hydraulic loads	(1)
	50–120	(1)	(1)	(1)
	120–R.O.	(1)	(2) Same as 0–50	(1)

[a]To indicate a relative measure of importance of each stressor category, an approximate quantifier is shown as either low (1), medium (2), or high (3).

[b]R.O. = run-out flow.

Table 4. Stressor influence on pressure-containment casing

Part	Operating regime (% BEP flow)	Stressor influence and remarks[a]		
		Mechanical	Hydraulic	Tribological
Upper casing	0–50	(1)	(2) Near shutoff head, overpressure may cause casing distortion sufficient to allow excessive leakage at casing split line	(1)
	50–120	(1)	(1)	(1)
	120–R.O.	(1)	(1)	(1)
Lower casing	0–50	(1)	(2) Near shut-off head, overpressure may cause casing distortion sufficient to allow excessive leakage at casing split line	(1)
	50–120	(1)	(1)	(1)
	120–R.O.	(1)	(1)	(1)
Suction and discharge nozzles	All operating regimes	(1)	(1)	(1)
Fasteners	All operating regimes	(1)	(1)	(1)

[a]To indicate a relative measure of importance of each stressor category, an approximate quantifier is shown as either low (1), medium (2), or high (3).

[b]R.O. = run-out flow.

Table 5. Stressor influence on mechanical subsystems

Part	Operating regime (% BEP flow)	Stressor influence and remarks[a]		
		Mechanical	Hydraulic	Tribological
Thrust bearing	0–50	(1)	(1)	(3) Axial hydraulic load (static and dynamic)
	50–120	(1)	(1)	(1)
	120–R.O.	(1)	(1)	(3) Same as 0–50
Radial bearing	0–50	(1)	(1)	(3) Dynamic loads from hydraulic forces
	50–120	(1)	(1)	(1)
	120–R.O.	(1)	(1)	(3) Same as 0–50
Thrust balancer	0–50	(1)	(1)	(3) Radial and axial dynamic loads can cause rubbing
	50–120	(1)	(1)	(2) Static loads can increase gradually with use as wear-ring clearance opens up via normal wear
	120–R.O.	(1)	(1)	(3) Same as 0–50
Seals p — for packing gland type seals m — for mechanical type seals	0–50	(3) [p]Overtight packing gland causes excessive wear rate		(3) [p]Excessive rubbing possible due to dynamic hydraulic forces
	50–120	(3) [p]Same as 0–50		(2) [m]Potential for oxide abrasives to cause wear under all operating regimes
	120–R.O.	(3) [p]Same as 0–50 (2) [m]Excessive vibration		(1) [p]Same as 0–50 (3) [p]Same as 0–50
Coupling	0–50	(3) Axial rotor dynamic loads originating at the impellers		(3) Axial rubbing between mating teeth on gear couplings
	50–120	(1)		(1)
	120–R.O.	(3) Same as 0–50		(3) Same as 0–50
	All			(2) Breakdown or loss of lubricant in gear couplings (2) Running time on life-rated parts, fretting

[a]To indicate a relative measure of importance of each stressor category, an approximate quantifier is shown as either low (1), medium (2), or high (3).

[b]R.O. = run-out flow.

Table 6. Stressor influence on support

Part	Operating regime	Stressor influence and remarks		
		Mechanical	Hydraulic	Tribological
Base frame	All regimes	(1)	None	None
Fasteners	All regimes	(1)	None	None

In Tables 2–6, to indicate a relative measure of importance of each stressor category, an approximate quantifier is shown as either low (1), medium (2), or high (3). To some extent these quantifiers are somewhat subjective, but are based on the insight derived from extensive field experience on power plant pumps. Where warranted on the basis of field experience with multistage pumps, specific stressors of high relevance are referenced on Tables 2–6.

5. OPERATING EXPERIENCE*

The primary basis for information, insights, conclusions, and recommendations given in this report was the extensive data collection developed by Energy Research and Consultants Corporation from design, troubleshooting, and redesign experience. This collection was augmented through examination of nuclear power plant operating experience records as discussed below.

This section summarizes AUXFP aging information obtained from various nuclear power plant operating experience records. Several Licensee Event Report (LER)-based pump failure studies were examined for relevant pump operating and failure information. While these documents do not always contain specific pump aging-related failure data, the operating experience summaries and failure cause data, along with the overall analysis results, are helpful in understanding the aging degradation of pumps.

A number of operating experience data bases for nuclear power plants were examined for this report:

- LER file
- Nuclear Plant Reliability Data System (NPRDS)
- In-Plant Reliability Data System (IPRDS)

Specific information needed for AUXFP failure characterization includes: (1) failure modes, causes, and mechanisms; (2) frequencies of failures; (3) methods of failure detection — incipient, degraded, catastrophic; (4) maintenance actions; and (5) modifications resulting from failures.

Each of these items serves to build a failure "signature" that, when taken in total, can provide a comprehensive assessment of the component failure.

Unfortunately, no single data base provides all of the information desired for each failure. But each data base does possess some useful data elements that can be extracted for pump failure study. Table 7 lists the information available from various sources of operating experience. A summary of pump failure information available from each identified data source is contained in Appendix B.

5.1 Failure Modes and Failure Causes

The primary reported failure causes of AUXFPs were failure of the bearings, followed by shaft packing and seal failures. The predominant reported failure mechanism was improper or insufficient lubrication or cooling. Maintenance errors (that were not readily apparent) and wear were other major reported pump failure mechanisms. Wear of bearings,

*Work performed by G. A. Murphy, ORNL Nuclear Operations Analysis Center.

Table 7. Summary of AUXFP pump failure information available from operating experience and plant documents

Data/source[a]	Operating experience data bases			Plant-specific documents			
	LER[b]	NPRDS[b]	IPRDS[b]	SAR	SD	TS	ISI/IST
Pump type and description		X	X	X	X		
Manufacturer and model No.		X			X		
Operating environment		X		X	X		
Failure cause	0	X					
Failure mechanism	0	0					
Discrete failed part	0	X					
Maintanance action	0	0	X				
Modification to prevent recurrence	0	0	X				
Failure trend data						X	X
Incipient failure detection			X			X	X
Specific application					X	X	X

[a]Acronyms

IPRDS = In-Plant Reliability Data Study
ISI/IST = In-Service Inspection/In-Service Testing Program
LER = Licensee Event Report
NPRDS = Nuclear Plant Reliability Data System
SAR = Safety Analysis Report
SD = (Plant) System Description
TS = Technical Specification/Surveillance Test Program
X = Generally available
0 = Occasionally included in failure report

[b]Examined in this study.

packing, and seals is anticipated with extended use, but the factors mentioned above can accelerate the wear.

5.2 Frequency of Failure

No frequency of failure data for the AUXFP (as defined in this report) are available from data sources examined in this study. Numerical values published by NPRDS (and in IEEE-500, 1984) are derived from failure of pump function, which could include numerous components — valves, instrumentation, controls, etc. — all associated with the pump function. For example, an NPRDS data request for failures of AUXFPs yielded 70 records. Of the 70 records, only 14 (20%) reported a failure of the pump; the balance of the records described personnel and maintenance errors or failure of peripheral equipment that disabled the pump function. Similarly, the LER data base search for failure of AUXFPs yielded 1139 events. Only 53 events (4.6%) involved pump failures; the balance again described loss of pump function.

5.3 Methods of Detection

In the 14 NPRDs and 53 LER events, nearly one-half of the failures were detected during surveillance or other tests. About one-third of the failures occurred while the pump was in operation.

5.4 Maintenance Action

Listed maintenance action for the failure events, consisted mainly of replacing a worn or broken subcomponent. In less than 10% of the failures modifications were indicated as required to return the pump to service.

5.5 Modifications Resulting from Failures

Operating experience data bases do not always contain detailed descriptions of postfailure modifications. Some of the data included the following:

- impeller modifications were generally made to meet head-flow requirements, reduce stress in the key-way area, and increase clearance between the hub and the labyrinth rings;
- maintenance and repair procedures were continually revised to include proper alignment of pump subcomponents;
- bearing and packing cooling often required modification to prevent overheating; and
- special training for maintenance personnel complements improved repair procedures to gain overall pump reliability.

6. FAILURE MODES AND FAILURE CAUSES

In Sect. 5, failure modes and causes were introduced within the context of "operating experience," relying upon the available data-bases therein referenced. However, that information provides primarily sparse statistics for recorded failure experiences. Furthermore, the total number of operating hours on AUXFPs, industry-wide, is probably insufficient to establish from such data potential cause-and-effect relationships between failure modes and causes. Fortunately, there is a much larger body of applicable experience stemming from (1) feedwater pumps specifically and (2) turbomachinery and rotating machinery in general.

It is recognized that AUXFPs are subjected to intermittent operation which makes the experience differ from that of feedwater pumps, but there is not an available data base with better correspondence. There are many similarities in design and function between the two to justify use of feedwater pump experience. Thus, what has become fairly common practice with similar machinery in identifying failure modes and causes must be used to provide viable approaches to maximize operational readiness of AUXFPs.

6.1 Failure Modes

Failure modes must be defined carefully and uniquely because the word "failure," as herein used, has a much broader meaning than just total loss of function due to sudden breakage. The following subsections define failure modes.

6.1.1 Failure to operate

1. Required driver torque is applied to coupling, but rotor does not rotate.
2. Power interruption by automatic tripping devices, such as overspeed trip on turbine-driven units. This includes the possibility of a true trip or one resulting from malfunction of the trip device.

6.1.2 Failure to operate as required

1. Failure to provide required head-capacity pumping characteristic.
2. Critical parameter measurements (e.g., vibration, bearing temperature, delivered flow) outside acceptable ranges. Although the pump is still able to perform the design function and is, therefore, not failed, it is either repaired or replaced because it might fail in the immediate future. Because the pump has to be taken out of service to repair the potential problem, these events are considered failures.

6.1.3 External leakage

A leak of the pump body (i.e., seals or casing) could allow the contained medium (water) to escape from the boundary. This failure mode pertains to leakage rates that either indicate impending loss of function or significantly reduce the flow delivered by the pump.

6.2 Failure Causes

In this subsection, only the causes of the failure modes due to aging and service wear are identified in general terms; analyses of the causes are covered in Sect. 6.3 and substantially discussed in Appendix D. In the following subsections, potential failure causes for each failure mode of 6.1 (summarized in Table 8) are described and also summarized in Table 9 and deal primarily with proximate causes of failure in each case.

Table 8. Failure modes

Failure mode	Remarks
Failure to operate	Rotor does not rotate
Failure to operate as required	Pump fails to provide required head-capacity pumping characteristics Critical parameter measurements outside acceptable ranges
External leakage	Escape of contained medium from component boundary

6.2.1 Causes of failure to operate

Material lodging in rotor clearance spaces can precipitate failure to operate, as can broken coupling gear teeth. It is quite possible for dirt or loose metallic fragments (particularly in new systems) to become lodged in small rotor clearances. In most instances, such occurrences will "clear" themselves with some minimal amount of resulting wear. However, under extreme conditions, this can lead to galling and a rapid seizure of the rotor at some location with the stationary parts of the machine. Then the rotor will become "locked" against rotation. A far less likely result would be imposition of excessive torque due to drag that would limit rotation to slower than normal speeds.

Breaking off gear teeth may cause large side loadings on the shafts, which can produce a locked rotor condition. It may also cause the shaft to break and thus sever the transmission of torque. In either case, the pump rotor will cease to rotate.

Table 9. Summary of failure modes and causes

Failure mode[a]	Segments and parts involved	Failure causes	Failure mechanisms	Relative probability of occurrence[b]
1	Rotating element at close clearance with stationary parts	Binding between rotor and stationary parts	Dirt or metallic debris, galling and seizing	M Moderate
	Pump shaft and coupling	Loss of drive torque	Coupling or pump shaft breakage	M
2	Impellers and nonrotating internals, seals, wear surfaces, bearings, thrust balancer, shaft, coupling	Seizure or breakage of shaft, breakage of impellers or nonrotating internals, bearing or thrust balancer seizure	Hydraulic forces at high and low flows, material fatigue, high vibration	H Frequent
2	Impellers and nonrotating internals, interstage sealing clearances (wear surface)	Deterioration of pumping capacity resulting from rapid wear, structural failure of internals	Hydraulic forces, high vibration, cavitation	M
2	Rotating element, seals, bearings, thrust balancer, internals	High vibration, high bearing temperature, abnormal seal leak-off flow, abnormal pump performance	All of the above under failure mode 2	H Frequent
2	Speed control governor, overspeed trip device	Overspeed trip	Large speed overshoot on start-up due to excessively fast speed-up True overspeed trip	H or L, depending upon system configuration
3	Shaft seals, casing	Deterioration or breakage of seal components	High vibration, improper packing adjustment	H
		Leakage through casing static sealing joint	Casing distortion and "wire drawing"	L Infrequent
		Erosion, corrosion, or structural failure of casing		

[a]Failure mode legend: (1) Failure to operate
(2) Failure to operate as required
(3) External leakage

[b]L — Low
M — Medium
H — High

The one automatic tripping action that is built into all turbine-driven AUXFPs is an overspeed trip. In a number of plants and some quite recently, overspeed tripping at startup has been a problem. When startup action is initiated, the steam valve in the line to the drive turbine (typically a 6-in. line) fully opens, and maximum torque is applied to the pump, resulting in rapid acceleration to and above operating speed; this occurs in about 1 s. Before the speed control governor has time to pressurize its oil system and react correctively, the overspeed trip value of speed is exceeded and the automatic trip mechanism closes the steam valve, shutting down the machine (see Appendix C for additional discussion).

6.2.2 Causes of failure to operate as required

Causes include journal bearing or thrust bearing failure, galling and seizure of the thrust balancer, sudden shaft breakage, sudden disintegration of an impeller(s) or detachment of a stationary (diffuser) vane(s). None of these causes are uncommon in HHPS multistage centrifugal pumps. An additional cause is deterioration by cavitation of the first-stage impeller; others are unacceptable critical parameter readings and automatic tripping.

Operation of these pumps "far" from the BEP flow induces very strong unsteady flow conditions within the pump hydraulic passages. The strong unsteady flow phenomena result in very large dynamic forces (radial and axial) on pump internals, both stationary and rotating parts. The immediate by-product is high-amplitude vibration that causes rapid wear at critical clearances in the pump due to severe vibration-induced rubbing. This leads to rapid increase in stage-to-stage leakage, resulting in measurable reduction in delivered pump capacity. In addition, these large dynamic fluid forces can break loose large pieces of diffuser vanes, impeller side plates, and impeller vanes. Such rapid deterioration of pump internals will naturally result in considerable reduction in delivered capacity of the pump as well as catastrophic structural failure.

Deterioration of the first-stage impeller due to cavitation erosion also degrades performance. Not having sufficient Net Positive Suction Head (NPSH) available at the pump inlet over the entire operating range is sufficient to produce this type of first-stage impeller damage. The net effect will be a slow deterioration in the delivered capacity of the pump.

Monitored parameters such as vibration, bearing temperature, seal leakage, delivered flow, and rotor speed are typically measured and recorded continuously on main power cycle pumps (e.g., main feedwater, condensate, and reactor coolant pumps). At present, such monitoring capability is not typically installed on AUXFPs. One or more of these parameters can be reliable indicators of an impending pump failure as well as being correlated with aging and service wear factors and analyses. Obviously, an upward or downward trend away from normal seal leak-off flow is a fairly sure indication that a seal is wearing out or needs adjustment. An upward trend in vibration level or a sudden increase in vibration is a sure indication of impending pump malfunction (if proper action is not taken). Temperatures of journal bearings and especially oil-film thrust bearings are indicative of impending difficulties when they climb above established limits for reliable operation. Furthermore, excessive vibration or elevated bearing temperature has serious deteriorating effects on the pumps and thus requires immediate corrective action, regardless of the specific cause. A significant deterioration of delivered pump capacity would also be indicative of a potentially serious problem with the pump that should be immediately investigated and corrected.

The measured parameters by themselves may not divulge the full nature of a problem. However, they can at least give warning that the machine must be shut down as soon as possible for further inspection and possibly disassembly of the pump for inspection of internal parts for the purpose of determining the probable cause(s) of out-of-normal-range parameter readings.

Any one of these monitored parameters or a combination thereof could be incorporated into various automatic tripping schemes. This is common practice for large steam turbine-generator units, reactor coolant pumps, main feedwater pumps, large fans, etc., primarily to detect potentially catastrophic failures before they occur and thus prevent costly damage. However, the "philosophy" behind using such tripping schemes on AUXFPs should very definitely be given special examination. It would make no sense at all to trip an AUXFP during an emergency to save it from life-shortening wear due to excessive vibration or somewhat high bearing temperatures. The primary purpose of the AUXFP is to protect the reactor system, not to protect itself.

6.2.3 Causes of external leakage

Three causes are related to this failure mode. The first and most common is failure of the shaft seals to properly control leakage to within acceptable limits. This can occur either from an outright breakage of a sealing component or from a progressively deteriorating seal that may need replacing or simply a tightening adjustment as in the case of stuffing-box type shaft seals.

The second cause is malfunction of the flat static sealing joint formed at the split between the upper and lower casing halves. Though far less common than the first cause (i.e., shaft sealing malfunction), it is a possibility and does occur on structurally marginal split-casing multistage centrifugal pumps. When it occurs, it generally indicates that the overall casing design is marginal from a rigidity or deflection standpoint, and the joint simply opens up a little at maximum pressure conditions; that is, at or near shutoff head, the casing design is not quite stiff enough. The worrisome aspect of this phenomenon is that "wire drawing" damage to the flat-joint sealing surfaces is a possible by-product and thus a continually increasing leakage rate may develop, even at lower operating pressures, once the "wire drawing" erosion progresses.

The third and least likely cause is associated with erosion, corrosion, and structural failure of the pressure envelope. Based on extensive field experience with all types of multistage high-pressure centrifugal pumps, this type of failure should be categorized as one of extremely low probability.

6.3 Failure Cause Analysis

The purpose of this section is to provide more specific discussion on the various failure-mode causes, based on troubleshooting experience and insight on AUXFPs, main feedwater and booster pumps (both nuclear and fossil plants), and multistage centrifugal pumps for other service such as in petrochemical applications. The failure modes herein identified for AUXFPs are inherent as well in these other multistage pump applications.

HHPS multistage centrifugal pumps remain an engineering challenge. Unfortunately, this type of pump has not been viewed by manufacturers and users as the high-technology device that it really is. The amount of engineering research in the United States devoted to HHPS pumps over the last 25 years has been rather insignificant (see Appendix D).

The fact that AUXFPs spend most of their plant life in the standby mode is at the same time an advantage and a disadvantage. The advantage is that they are, on the average, less likely to fail from wear than their "big brothers" — the boiler feedwater or nuclear feedwater pumps that are operated continuously. On the other hand, whatever inherent design weaknesses exist in present AUXFP designs, they are less likely to fully surface on the short-term basis — a disadvantage that gives considerable importance to the main theme of this report.

Through recent field troubleshooting, specifically on AUXFPs and on quite similar pumps for nonclass applications, some specific factors have surfaced that impact significantly on the reliability of AUXFPs. These factors are discussed in the following subsections.

6.3.1 Large dynamic forces and vibrations created by pump hydraulics

Further review of stressor influence factors (Tables 2–6) and, correspondingly, the summary of failure modes and causes (Table 9) clearly shows that the large dynamic forces resulting from pump internal flows is the major failure-mode factor. Within this context, the most severe operating mode is at low flow (i.e., close to shutoff conditions). The hydraulic forces generated are large and the resulting vibration levels are high. Consequently, AUXFPs are installed with a bypass or recirculation line, which has long been standard practice on nuclear main feedwater pumps and boiler feedwater pumps.

A bypass flow line of 25% best-efficiency capacity ensures that no matter how low the delivered pumping capacity is at part-load operation, the pump internally never "sees" less than 25% flow. In main feedwater and boiler feedwater pumps, the bypass flow line is valved so that, at flow rates above bypass flow, the full capacity of the pump is available and the energy loss associated with bypass flow is incurred only under low-flow operation. This same approach of valving the bypass flow line has been applied in many AUXFP installations. However, an emerging trend has been to remove this bypass line valve from already installed AUXFPs or to delete it in newer plants; that is, to eliminate completely the possibility of having a bypass valve malfunction in the closed position or to be mistakenly left closed.

The required flow of main boiler and nuclear feedwater pump bypass flow lines is typically a relevant evaluation factor in pump vendors' sales proposals. This is because the larger the bypass flow required, the more pumping energy is wasted at part-load operation. In retrospect, pump manufacturers now indicate that this same evaluation philosophy was applied in the purchase of AUXFPs, forcing them to compete by providing small required (typically 10 to 15%) bypass flow figures. However, the energy consideration just noted is surely not relevant to safety-related standby pumps such as AUXFPs. Much more attention and engineering investigation is warranted, and each AUXFP design should be thoroughly tested to determine the true safe minimum flow rate. Because energy consumption is not a major consideration in AUXFP service, one simply can size the pump so that its full capacity is sufficient to always "waste" the bypass flow and, thus, protect the pump from the abusive operation and accelerated wear that it could otherwise encounter.

Obviously, the larger the percentage of BEP flow routed through the bypass line, the better it is for the pump. In well-engineered pumps, 25 to 35% bypass flow is sufficient to avoid the dangerous range of off-design operating flows (see Appendix D and Fig. D.1). However, for some pumps any flow below 50% of best-efficiency capacity may cause severe vibration. This raises a crucial and not yet widely recognized (and thus unresolved) potential safety-related problem. Apparently, determination of an appropriate AUXFP bypass flow rate has been done only on a plant-by-plant basis and then only in response to piping vibration problems, not in appreciation of AUXFP abuse considerations.

6.3.2 Shaft breakage

Several factors, identified through field troubleshooting, result in shaft breakage as an initiating event in a failure (see Appendix D for additional discussion). Because of the presence of large hydraulic forces, any design or quality control (QC) shortcoming affecting the shaft can readily lead to shaft breakage. A common such design flaw is an undersized shaft diameter, generally motivated to improve pump hydraulics by providing larger impeller inlet flow areas.

In some failures, it was determined that the shaft failed as a consequence of the retaining bolt (which retains the runner of the thrust balancer) not being square with the mating shaft shoulder when fully tightened. This results in high bending prestress of the shaft, leading to fatigue-initiated breakage of the shaft. This is a QC correctable problem.

Another source of shaft breakage observed in the field is improper heat treatment of the shaft. This is also a QC correctable problem.

A further source of shaft failure arises from overheating caused by overtightening of a shaft seal stuffing-box gland. This is a maintenance correctable problem. Stuffing-box shaft seals may require frequent tightening adjustment to control shaft seal leakage, even in the standby mode. Thus, one can envision an inadvertent overtightening of the packing gland in an effort to reduce the required frequency of adjustment. Although AUXFP manufacturers do not use forgings for the shaft, this more expensive option would certainly improve shaft durability and thus increase reliability.

6.3.3 Impeller and diffuser breakage

If the bypass flow line is undersized (see Sect. 6.3.1), then impellers and diffusers are among those pump internal components that are subjected to severe dynamic loading at low-flow operation. This can create fatigue failure resulting from cyclic stresses.

Even if the bypass flow line is adequately sized, a too small radial gap between impeller-vane outside diameter and diffuser-vane inside diameter frequently results in impeller and diffuser breakage. This breakage results from high-amplitude shock loads at vane-passing frequency (see Fig. D.5). The offending factor here is poor hydraulic design. This can be fixed in the field by experienced personnel and entails machining and recontouring the diffuser vanes to a larger radial clearance with the impeller. This fix also generally makes the pump run much quieter, which by itself generally is indicative of improved internal hydraulic configuration.

Cavitation damage to the first-stage (i.e., suction-stage) impeller can result in structural failure of the impeller. Insufficient NPSH is a common problem of feedwater systems in general and stems from a wide range of mostly optimistic opinions on what constitutes adequate NPSH.

A significantly reduced potential for impeller and diffuser breakage could be achieved if impellers were precision cast rather than using the lower quality sand castings as is presently done by all AUXFP vendors.

6.3.4 Thrust bearing and thrust balancer failures

Even for standardized pump configurations, there exist many possible variations that a vendor can tailor to meet a given application specification. For all the available designs and variations thereof, there is simply not adequate engineering test data on hydraulic axial thrust loads over the full flow range of each specific design. Thrust loads change considerably on a given pump as a function of flow, speed, and state of wear at wear-ring stage-to-stage leakage control clearances. Essentially, the axial thrust loads are not accurately known over the full ranges that feedwater pumps are commonly required to operate. Consequently, thrust balancers and thrust bearing failures in feedwater pumps are not uncommon because the inaccurate definition of axial thrust loads (both static and dynamic) often results in undersized thrust balancer and thrust bearing designs.

Excessive operating temperatures in thrust bearings are common symptoms indicative of this overall problem. Furthermore, on thrust balancers, improper heat treatment and material property control result in galling and seizing events, especially when combined with undersized geometries (see Appendix D for additional discussion).

6.3.5 Coupling breakage

Coupling breakage is not uncommon. The combination of excessively low-flow operation of the pump and an undersized thrust balancer can result in excessive axial dynamic loads on the coupling. Additionally, when the radial gap between impeller and diffuser vanes is too small, excessive coupling vibration at vane-passing frequency can readily destroy the coupling (gear teeth).

Breakage of seal injection piping due to excessive vane-passing frequency vibration also can occur when the impeller-to-diffuser radial gap is too small.

6.3.6 Seizure

Seizure is not uncommon in feedwater pumps and has been traced to a variety of causes including (1) excessive rotor vibration (both lateral and axial); (2) inadequate clearance at thrust balancer; (3) seizure-prone material hardness and geometry combination at impeller wear rings, thrust balancer, or any other close-running clearance; (4) overall hydraulic mismatch between impeller and diffuser; and (5) journal bearing failure resulting from excessive rotor vibration, poor babbitt bonding to bearing shell, dirt in the lubrication oil system, or combinations of these.

7. RECOMMENDED MAINTENANCE, SURVEILLANCE, AND MONITORING PRACTICES

AUXFPs spend most of their plant life in a standby mode. Consequently, present maintenance, surveillance, and monitoring practices are not elaborate. Pump manufacturers are mainly concerned with other applications involving continuous long-duration running where certain components clearly do wear out and where scheduled major overhaul and refurbishment is common practice. Regarding AUXFPs, the manufacturers make no definitive recommendations.

At present, most AUXFP installations contain no monitoring devices except for flow and head measurement, in contrast to continuously running machinery. Present AUXFP surveillance practice consists primarily of periodically* starting each pump for a short-duration test to verify its operational readiness.

7.1 Regular Maintenance

The only present regular maintenance activity pertains to shaft sealing components. In the normal standby mode, the shaft seals are still subjected to upstream system pressure, that is, from a condensate or storage tank. This pressure is typically 2 to 4 atm and thus maintains the shaft seals in a functional sealing mode, even though the shaft is not rotating.

Stuffing boxes as well as mechanical seals are in common use in AUXFPs. With stuffing-box shaft seals, the primary maintenance activity is to keep the packing gland properly adjusted. Based primarily on an observation of excessive leakage, a tightening adjustment is made. However, stuffing boxes must be allowed to maintain some continuous leakage. Overtightening of stuffing-box glands to eliminate leakage completely will likely result (and has on AUXFPs) in severe operating problems when the AUXFP is put into a pumping mode, that is, at normal rotational speed. Overtightened stuffing-box glands result in excessive temperature rise both in the packing and on the shaft, increasing considerably packing wear and the probability of shaft breakage. In many plants, AUXFP stuffing-box packing glands are now maintained somewhat looser than would be typical of non-AUXFP applications. Although this results in somewhat higher continuous leak-off flow, it is a prudent practice, since it gives a further safety margin against such heating problems and related potential failures during any pumping mode.

With mechanical shaft seals, the leak-off flow in the standby mode should be zero, if the seal is functioning properly. That is, at zero shaft rotation, a mechanical seal essentially closes the flow path for shaft-end leakage. If a mechanical seal is observed to be leaking in the standby mode, this is a strong indication that something is wrong with

*Plant Technical Specifications require operational readiness testing at least once every 3 months. However, in many plants, this testing is performed as frequently as once a month.

the seal, and thus it should be fixed or replaced as required to correct the problem.

In addition to these seal maintenance items, lube-oil sight glasses on the AUXFP should be regularly inspected to check oil level. Most other maintenance items would require some amount of disassembly to perform and thus would be objectionable, unless there were strong indications of pump malfunction, for example, as indicated by excessive vibration.

7.2 Surveillance

7.2.1 Current surveillance practice and limitations

Unfortunately, AUXFPs are rarely installed with the type of monitoring devices typical of equipment provided for continuous operation service. Installed rotor vibration probes and bearing inside temperature transducers (e.g., babbitt-contacting thermocouples) have generally not been required by AUXFP purchaser specifications. Only in a very few plants, where recurring AUXFP operating problems have otherwise been detected, has such monitoring equipment been retrofitted. Thus, in most plants, comprehensive surveillance of AUXFPs during pumping mode service is not readily done.

As noted in Sect. 4.3.2, some newer plants use a nonclass pump for startup, shutdown, and any other nonemergency service. This "saves" the N-class AUXFPs for strictly emergency service with one important exception. The Plant Technical Specification requires that, nominally every three months, each N-class pump is put into a pumping mode for a short period of time to demonstrate its continued operational readiness. During this regular test, the loads imposed on the AUXFP depend on the bypass flow line configuration. In most plants, the AUXFP bypass line is a single line sized to pass 5 to 15% of BEP flow. Thus, this type of testing could be the main contributor to wear and aging of various AUXFP components. In some newer plants (e.g., Palo Verde) an additional full-flow bypass line is provided to allow testing of the AUXFPs over the full operating flow range, even when the plant is operating in a normal generating mode. However, in most plants, this test provides neither the proper operating range of flow nor sufficient running time to comprehensively trend and assess an AUXFP's "vital life signs," even if a full complement of state-of-the-art monitoring devices were installed.

7.2.2 Interim recommendations for surveillance and monitoring practices

Based upon the described impediments to comprehensive surveillance and performance trending, some obvious interim recommendations can be manifestly considered.

First, present auxiliary feedwater systems need to be studied for modifications that would permit ease of testing the AUXFPs over the entire range of flow, independent of the main feedwater system. One possibility is to permanently install bypass flow lines to allow testing the

AUXFPs at operating conditions that fully simulate the various emergency pumping requirements that the AUXFPs have to meet. Built-in full-flow testing capability could be engineered to avoid adversely affecting the reliability of the auxiliary feedwater system in normal and standby modes. This has been demonstrated at the Palo Verde plant.

Second, a full complement of permanently installed monitoring devices should be considered on every AUXFP. Major candidate parameters should include at least (1) rotor orbital motion; (2) oil-film bearing temperatures; (3) head, flow, and speed values; (4) rotor axial position; and (5) metal fragment and/or sound emission for early detection of incipient failure of rolling contact bearings.

Third, a standardized sequence of full-service tests need to be developed and performed at appropriate regular intervals. The full complement of parameters monitored could be appropriately analyzed, trends established, and results compared over the accrued history of the pump. This data base could be used to indicate need for preventive maintenance, overhaul, and replacement of worn parts. Such monitoring and trending systems for rotating machines are now commercially available at a small fraction of the cost of an N-class AUXFP. Section 8.2 provides a more detailed discussion of parameter monitoring.

7.2.3 Interim recommendations for detailed inspection program

In addition to the comprehensive testing, surveillance, and monitoring approach recommended in the previous section, a detailed disassembly and inspection of AUXFP internals could be carried out whenever the reactor is in a scheduled shutdown for a number of weeks, such as at refueling time.

During this disassembly, a number of procedures and reliability enhancing actions could be taken: (1) replace rolling contact bearings; (2) replace renewable components of mechanical shaft seals; (3) replace any wear rings having clearance-doubling wear; (4) inspect impellers, diffuser vanes, and other hydraulic-passage internals for damage and replace any damaged or significantly worn parts based upon conservative criteria; (5) inspect journals, bearing surfaces, and thrust balancer components, replacing any distressed components; (6) inspect main sealing joint surfaces at casing joint to detect any sign of leakage; (7) check shaft for run-out (can readily occur through creeping) and correct as required by straightening or replacement of shaft; and (8) inspect and replace any fastener that shows signs of distress. Section 8.1 provides a more detailed discussion of the overall topic of detailed inspection.

8. AGING AND SERVICE WEAR MONITORING

As described in Sect. 7.2, nearly all AUXFPs presently in service are installed with little or none of the specific parameter monitoring devices that are commonly integrated into continuously running main power cycle equipment. Furthermore, most auxiliary feedwater systems are not presently configured to facilitate regular periodic testing of AUXFPs over the complete range of flow rates that would simulate the various emergency scenarios for which AUXFPs are installed. Consequently, with present typical installations, aging and service wear monitoring would have to be based primarily upon a detailed and thorough inspection of the pump in the disassembled state. Unless specific operating problems arose with a particular AUXFP, such a detailed inspection would be practical only during a scheduled plant shutdown, such as at refueling time.

Shaft seal leak-off flow is the one parameter that can be and typically is readily monitored, both in the standby ready mode as well as running modes. In the following sections, monitoring specifics that are advisable and can be readily performed with present typical configurations are described. In addition, monitoring specifics that could further assist in aging and wear analyses, utilizing state-of-the-art monitoring packages, are described for consideration as upgrading options.

8.1 Monitoring of Present Configurations

Here the discussion is aimed at existing configurations that typically have available neither a full complement of state-of-the-art continuous monitoring devices nor are set up to routinely run full-flow range AUXFP tests.

8.1.1 Rotor binding check

The following is a simple procedure that is probably not commonly practiced in existing plants although it is highly advisable. Just prior to starting the pump, the rotor should be hand rotated, grabbing it at the coupling. Generally, a pair of nonslip gloves must be worn to accomplish this. This procedure could also be performed at regular intervals even if the pump is not about to be started. For safety reasons, the pump driver must be "tagged out." Naturally, this procedure should be disregarded in the event of a true emergency startup of the pump.

If it is not possible to hand turn the rotor, then a suitable wrench or wrenchlike device should be used to attempt a wrench-assisted hand turning of the rotor. If this action fails to turn the rotor, then the cause of the binding can be identified and appropriate corrective action taken.

8.1.2 Shaft seal leak-off flow

8.1.2.1 Mechanical seals. Properly functioning mechanical seals should not exhibit leak-off flow large enough to be seen or measured. If leakage is clearly visible, then the seal is potentially in need of parts replacement and at least the reason for the leakage should be determined and appropriate corrective action taken.

8.1.2.2 Packing gland (stuffing box) seals. This type of seal should always leak a small amount to ensure that the packing gland has not been overtightened. Excessive leakage, which cannot be corrected through normal gland adjustment, could be indicative of worn or damaged components in the seal subassembly. The defective or worn part(s) should be replaced.

8.1.3 Disassembly and detailed inspection

The following subsections pertain to those aging and service wear monitoring actions that require disassembly of the pump and that could be performed during a scheduled plant outage, that is, at refueling time. To disassemble this type of split casing pump, the following steps are taken: (1) the coupling is disengaged, (2) the journal bearing caps are removed, (3) the shaft seals are removed, (4) the casing stud bolts are removed and the upper casing half lifted off of the bottom casing half, and (5) the rotor is lifted out of the lower casing half. At this point, each wear ring will be loosely surrounding its respective impeller and can be moved axially to hang on the adjacent shaft section so that the inside diameter of each wear ring can be measured.

Upon disassembly of the pump as outlined above, the following measurements and visual inspections can be made.

8.1.3.1 Journal bearings, journals, and thrust bearing. Visually inspect all bearing surfaces for evidence of (1) bearing metal (babbitt) fatigue cracks, (2) loss of babbitt-substrate bonding, (3) babbitt wiping or smearing, and (4) babbitt gouging that would occur from entrained foreign matter in the oil. If any of the above deteriorations are in evidence, then the bearing should be replaced, or repaired to essentially as-new condition. The journals should also be inspected for signs of distress or wear. Specifically, measurable or visual evidence of surface deterioration, such as deep scratches, heat checking, or scoring should require refurbishment to return journal surfaces to their as-new condition. If the journal surface distress is quite extreme, then the journals may have to be chrome-plated to a sufficient thickness so that resurfacing of the journals does not reduce the journal diameter to less than its original dimensional tolerance zone. If the journals and bearings are repaired during the same disassembly, the original bearing clearance could be preserved by remachining a rebabbitted bearing to a slightly reduced bore diameter, sufficient to offset any slight reduction in journal diameter resulting from redressing its surface.

The AUXFP thrust bearing is usually a rolling contact type, such as a double-row preloaded angular contact (duplex) ball bearing arrangement. This bearing can be inspected for signs of wear or damage although subsurface fatigue cracks are not visible. However, such bearings are relatively inexpensive, and thus automatic replacement is strongly advised.

8.1.3.2 Shaft seals and mating rotor sleeves. For mechanical seals, the mating sealing surfaces (rotating and nonrotating parts) should be closely inspected visually, with high magnification if necessary, to determine the presence of any measurable signs of wear. Any evidence of wear is sufficient cause to replace the worn seal component.

For stuffing-box seals, the rotor sleeve should be inspected for signs of wear and/or surface finish deterioration. It is not uncommon for the rotor sleeve to have deep grooves worn in from continual running with an excessively tightened packing gland. The shaft sleeve should be replaced in any case where this inspection uncovers measurable wear or surface deterioration. The packing material, though adjustable in service to compensate for wear, will eventually need replacing. It is inexpensive and thus should be automatically replaced at any such disassembly-for-wear inspection, using the same rationale previously applied to rolling contact bearings.

8.1.3.3 Coupling. The disassembled coupling should be inspected for any deterioration of the gear teeth. If the coupling shows measurable wear or tooth surface deterioration, then the coupling should be replaced. Here it is highly recommended that when coupling replacement is deemed appropriate, it be replaced by the dry diaphragm type of flexible coupling that requires no maintenance and no lubrication and has a very successful retrofit record on all types of feedwater pumps.

8.1.3.4 Fasteners. Each fastener that is disengaged during the disassembly process should be closely examined using approved procedures.

8.1.3.5 Wear ring clearances. Each wear ring inside diameter and mating impeller-eye outside diameter should be accurately measured using a micrometer. If the clearance has worn as large as twice the as-new clearance, the wear ring should be replaced to return the clearance to its as-new nominal value. For example, if the as-new clearance is nominally 0.015 in. diametral (typical), then a worn clearance of 0.030 in. or larger is justification for wear ring replacement.

When wear ring clearances are allowed to open excessively (as commonly occurs in poorly maintained pumps) a number of undesirable consequences will occur as the wear worsens:

- First, the delivered head-capacity performance of the pump will deteriorate because of the increased stage-to-stage leakage.
- Second, the axial thrust load on the thrust balancer and thrust bearing could increase, further taxing all thrust carrying components.
- Third, the rotor-dynamic performance of the pump will deteriorate significantly because proper (as-new) wear ring clearances assist in controlling residual rotor vibration levels arising from all sources of excitation (e.g., unbalance, hydraulic forces, etc.).

In cases of excessive clearances, the deterioration of pump hydraulic performance will significantly exceed that of just the leakage loss subtracted from the head-capacity characteristic, because the leakage itself becomes a large flow component and disrupts the main through-flow as it approaches and enters the suction side of the impeller. Thus, proper periodic monitoring of wear ring interstage sealing clearance is vitally important so that wear rings are replaced before clearances become excessive.

8.1.3.6 Thrust balancer. When a thrust balancing disk is employed, two areas should be closely inspected. First, the vertical face of the balancing disk (rotating part) and the mating stationary surface should be closely inspected (visually) to detect signs of heavy rubbing (e.g., scoring marks or wear). If any such distress of these surfaces is detected, the component(s) should be replaced or, if possible, repaired by remachining the surfaces. Furthermore, for thrust balancing disk configurations, the radial clearance of the cylindrical pressure breakdown serrated bushing should be measured using a micrometer on the mating diametral surfaces. The criteria here are exactly the same as those previously detailed for wear ring clearances (see Sect. 8.1.3.4). Namely, if the clearance has worn to twice the as-new clearance or larger, parts should be replaced to return this clearance to the as-new condition.

When a thrust balancing drum is employed, the inspection procedure and criteria for replacement are the same as for the cylindrical pressure breakdown clearance section of the balancing disk assembly. More discussion and technical insight on thrust balancers are given in Appendix D.

8.1.3.7 Suction (first stage) impeller. The first-stage impeller should be inspected to detect any evidence of erosion (i.e., pitting) indicative of cavitation-caused damage. If detected, the impeller should be replaced. Furthermore, if such cavitation damage is detected, the design parameters of the suction side of the auxiliary feedwater system should be reviewed and analyzed to determine why the pump NPSH is insufficient to suppress cavitation. In a properly designed system, adequate NPSH should be available to inhibit cavitation damage. The results of such an analysis could potentially suggest a redesign of the system to eliminate cavitation damage (see Appendix D).

8.1.3.8 Impellers and diffusers. A complete inspection of all impeller and diffuser vanes and impeller side plates should be made to detect any structural damage to these components. The unsteady flow dynamic forces that are produced, particularly under low-flow operation and especially on designs employing relatively small impeller-to-diffuser vane tip clearances, frequently break off pieces of impellers and diffuser vanes. Obviously, any such structurally damaged components must be replaced. Furthermore, modifications to pump hydraulic internals that will eliminate such structural damage in the future should be considered.

8.1.3.9 Shaft. The shaft should be closely inspected for any structural damage or cracks. If such are detected, the shaft must be replaced. Also, the shaft should be inspected to determine if any out-of-tolerance nonconcentricities have developed, such as may occur through creep bending of the shaft. In this case, the shaft may be salvaged through a proper straightening process in a qualified pump repair shop. A bent shaft can produce obvious undesirable results, including accelerated wear at wear ring clearances, seals, thrust balancer, bearings, and excessive rotor vibration.

All shaft sleeves, spacers, and fasteners should be closely inspected for wear or structural damage and replaced as needed.

8.1.3.10 Casing split surfaces. The flat casing split surfaces should be inspected to detect any sign of casing leakage, for example, as would be evidenced by "wire drawing" damage. Any such surface damage can be repaired, for example, through hand-lapping the damaged portions of the surface.

8.2 Continuous Parameter Monitoring

The aging and service wear monitoring methods here described are proposed as additional to (not in place of) those that can already be performed. The methods addressed in this section may not be applicable to most present AUXFP installations because of the absence of required monitoring devices and piping configurations that facilitate regular periodic testing of AUXFPs over the complete flow and head ranges to simulate the emergency conditions. In spite of these limitations, the monitoring methods described in the following sections are proposed to further assist in aging and wear analyses, utilizing state-of-the-art monitoring packages. These methods are thus recommended for consideration as upgrading options on existing plants as well as original equipment for future new plants.

8.2.1 Rotor vibration monitoring

Rotor vibration is a primary vital sign of rotating machinery just as body temperature is to animals and humans.

Firstly, excessive rotor vibration accelerates machinery deterioration. The major consequences of excessive vibration are increased potential of fatigue failures, shortened bearing life, breakage of shafts and shaft seals, and accelerated wear at internal leakage control clearances. Thus, early detection of excessive rotor vibration is an essential factor in the timely identification and correction of such problems.

Furthermore, real time analysis of rotor vibration signals can frequently allow one to detect and isolate the progression of various deterioration phenomena. For example, Fast Fourier Transforms (FFT) or spectrum analysis, which transforms time-base signals into the frequency domain, can reveal sources of deterioration through a study that includes establishing trends for dominant frequency components. A slow but steadily increasing vibration component with a frequency equal to the rotational speed multiplied by the number of coupling gear teeth can reveal that the coupling is approaching the end of its usable life. Strong subsynchronous (below rotational speed) vibration frequency components are generally indicative of either rotor-bearing instability or significant deterioration of pump hydraulic internals (primarily, wear ring clearances are badly worn). Strong synchronous vibration can be indicative of excessive unbalance, bent shaft, or both.

The generally accepted practice for rotating machinery is to have two noncontacting proximity probes mounted near each journal bearing at a 90° angular separation from each other. This allows one to observe the rotor (i.e., journal) orbit of vibration when the two time-base signals are spatially superimposed using an oscilloscope. The journal orbits can clearly reveal the detailed nature (on a continuous basis) of the rotor vibration. From this information, and a pre-existing data base for the AUXFPs, correlations and analyses for wear and aging factors can be made to assess the "state of deterioration" on a continuous basis.

8.2.2 Bearing temperatures and noise

8.2.2.1 Oil-film bearings. Bearing liner (typically babbitt) temperatures are commonly measured continuously on major power plant machines. As in the case of rotor vibration, excessive bearing temperatures by themselves accelerate deterioration, at least of the bearing. Furthermore, excessive bearing temperatures are generally a consequence of other problems such as lubricant starvation, general deterioration of the bearing, or excessive bearing loads as may occur in further consequence of deterioration of pump hydraulic internals, pump-driver alignment, and looseness of various centering fits or fasteners. A continuous time record of bearing temperature, combined with a pre-existing data base for the AUXFPs, can be used singly or in concert with other monitored information to develop a running evaluation of wear and aging deterioration of AUXFPs.

8.2.2.2 Rolling contact bearings. Properly measured acoustic emissions of rolling contact bearings can be an effective approach to detect early signs of bearing deterioration as well as incipient bearing failure. It is unlikely that rolling contact bearings would ever be employed in continuously running high-pressure multistage centrifugal pumps. Their frequent use as an axial thrust bearing in AUXFPs is indicative that these pumps are not intended for long continuous running. In other applications, such pumps are typically configured with double-acting pivoted-pad oil-film thrust bearing. Even for AUXFPs, the use of rolling contact bearings warrants engineering review to assess whether ball bearings are capable of performing reliably in this application.

8.2.3 Rotor axial position

Boiler feedwater pumps are typically supplied with a mechanical readout device (located near the thrust bearing) that indicates the axial position of the rotor relative to the stationary structure of the pump. Significant axial shifting of the rotor is generally indicative of wear of the thrust balancing disk. If a thrust balancing drum is employed, then significant axial shifting is generally indicative of thrust bearing wear. In either case, this type of wear is not consistent with reliable operation and, when detected, should immediately be investigated and fixed.

8.2.4 Pump head-capacity curve

A significant deterioration of the pump head-capacity curve is a certain indication that something in the pump internals has significantly deteriorated. Badly worn internal clearances or structural damage to pump internals has probably taken place. When combined with a pre-existing data base for the AUXFPs, periodic rerunning of the head-capacity curve could become an effective tool in assessing aging and service wear factors, singularly or in concert with other monitored information.

8.3 Summary of Aging and Service Wear Factors

A tabulation of AUXFP failure modes is given in Table 8. As more fully described in Sect. 6, the three failure modes are: (1) failure to operate, (2) failure to operate as required, and (3) external leakage.

A more definitive delineation of failure modes and corresponding failure causes is summarized in Table 9. Each failure mode is associated with segments and parts involved, causes, failure mechanisms, and relative probability of occurrence.

The relationship of aging and service wear factors and failure causes is summarized in Table 10. The lead entry of this table is pump segment with failure causes and failure modes keyed to it.

Table 10. Pump failure causes related to aging and service wear

Pump segment	Failure cause	Failure modes[a]		
		1	2	3
Rotating elements	Binding between rotor and stationary parts	x		
	Shaft breakage	x		
	Impeller wear, breakage		x	
	Thrust runner wear, breakage	x	x	
	Fastener loosening, breakage	x	x	
Nonrotating internals	Structural damage to stationary vanes (diffuser or volute)		x	
	Wear-surface wear, erosion, corrosion, seizing	x	x	
	Fastener loosening, breakage		x	
Pressure containment casing	Leak at casing split			x
	Suction nozzle leak, breakage			x
	Discharge nozzle leak, breakage			x
	Fastener loosening, breakage			x
Mechanical subsystems	Bearing wear, corrosion, breakage	x	x	
	Shaft seal deterioration, breakage			x
	Thrust balancer galling, seizing	x	x	
	Coupling wear, breakage	x	x	
	Fastener loosening, breakage		x	
Support	Base frame breakage		x	
	Fastener loosening, breakage		x	

[a]Failure mode designation:
1 — Failure to operate
2 — Failure to operate as required
3 — External leakage

Currently used methods for AUXFP failure detections are listed in Table 11. Essentially, this simple table relates failure modes and means of detection.

Table 11. Methods currently used to detect AUXFP failure modes

Failure mode	Means of identification
Failure to operate	Visual observation Pressure readings at flow-measuring orifice Pump driver current and voltage or steam flow measurement
Failure to operate as required	Pressure readings at flow-measuring orifice Measurements of key parameters, that is, vibration, temperature, rotational speed, and flow
External leakage	Visual

Table 12 summarizes the methods for differentiating between the different failure causes. Here, the failure mode is the main entry parameter with pump segments, failure causes, and differentiation methods keyed to it.

Table 13 lists the various measurable parameters. Here again, the main entry parameter is failure mode, with pump segments, failure causes, and measurable parameters keyed to it.

Table 14 is a major summary that condenses all key information on AUXFP part failure assessments, including parts, materials, significant stressors, failure causes, and measurable parameters.

Table 12. Methods for differentiating between failure causes

Failure mode	Pump segment	Failure causes	Methods for differentiation
Failure to operate	Rotating elements	Binding between rotor and stationary parts	Pump rotor cannot be manually rotated
		Shaft breakage	Visual examination[a]
		Thrust runner wear, breakage	Visual examination[a]
		Fastener loosening, breakage	Visual examination, inspection during maintenance
	Mechanical subsystems	Bearing wear, breakage	Visual examination,[a] clearance measurement, rotor axial position measurement
		Coupling wear, breakage	Visual examination[a]
		Thrust balancer galling, seizing	Pump rotor cannot be manually rotated, clearance measurement, rotor axial position measurement
	Nonrotating internals	Wear-surface seizing	Pump rotor cannot be manually rotated, visual examination (requires disassembly)
		Wear-surface wear erosion, corrosion	Visual examination, clearance measurement, rotor axial position measurement
		Fastener loosening, breakage	Visual examination, inspection during maintenance
Failure to operate as required	Rotating elements	Impeller wear, breakage	Visual examination (requires disassembly), large unbalance rotor vibration, delivered flow measurement
		Thrust runner wear	Transmitted torque measurement, rotational speed measurement, visual examination (requires disassembly)
		Fastener loosening, breakage	Visual examination, inspection during maintenance
	Nonrotating internals	Structural damage to stationary vanes (diffuser or volute)	Visual examination (requires disassembly), vibration monitoring, delivered flow measurement
		Wear-surface binding	Transmitted torque measurement, vibration monitoring
		Wear-surface wear, erosion, corrosion	Visual examination,[a] delivered flow decrease, clearance measurement, vibration monitoring
		Fastener loosening, breakage	Visual examination, inspection during maintenance

Table 12 (continued)

Failure mode	Pump segment	Failure causes	Methods for differentiation
Failure to operate as required (continued)	Mechanical subsystems	Bearing wear, corrosion, breakage	Bearing temperature measurement, vibration monitoring, transmitted torque measurement, visual examination,[a] rotor axial position measurement
		Coupling wear, breakage	Vibration monitoring, transmitted torque measurement, visual examination
		Thrust balancer galling, seizing	Transmitted torque measurement, rotor axial position measurement, rotational speed measurement, visual examination[a]
		Fastener loosening, breakage	Visual examination, inspection during maintenance
	Support	Base frame breakage Fastener loosening, breakage	Bearing temperature measurement, transmitted torque measurement, acoustic monitoring, vibration monitoring, visual examination, inspection during maintenance
External leakage	Pressure containment casing	Leakage at casing split	Visual examination
		Fastener loosening, breakage	Visual examination, inspection during maintenance
	Mechanical subsystems	Shaft seal deterioration, breakage	Visual examination, vibration monitoring, local shaft temperature measurement

[a]May require disassembly.

Table 13. Measurable parameters

Failure mode	Pump segment	Failure causes	Measurable parameters
Failure to operate	Rotating elements	Binding between rotor and stationary parts	Appearance (heat checking, welding)
		Shaft breakage	Appearance
		Thrust runner wear, breakage	Appearance, clearance
		Fastener loosening, breakage	Appearance, bolt torque
	Mechanical subsystems	Bearing wear, breakage	Appearance, clearance, rotor axial position
		Coupling wear, breakage	Appearance
		Thrust balancer galling, seizing	Appearance, clearance, rotor axial position
	Nonrotating internals	Wear-surface seizing	Appearance, clearance
		Wear-surface wear, erosion, corrosion	Appearance, clearance, rotor axial position
		Fastener loosening, breakage	Appearance, bolt torque
Failure to operate as required	Rotating elements	Impeller wear, breakage	Appearance, rotor vibration, delivered flow
		Thrust runner wear	Transmitted torque, rotational speed, appearance
		Fastener loosening, breakage	Appearance, bolt torque
	Nonrotating internals	Structural damage to stationary vanes (diffuser or volute)	Delivered flow, appearance, vibration
		Wear-surface binding	Transmitted torque, vibration
		Wear-surface wear, erosion, corrosion,	Appearance, delivered flow, clearance, vibration
		Fastener loosening breakage	Appearance, bolt torque
	Mechanical subsystems	Bearing wear, corrosion, breakage	Temperature, rotor vibration, transmitted torque, appearance, rotor axial position
		Coupling wear, breakage	Vibration, transmitted torque, appearance
		Thrust balancer galling, seizing	Transmitted torque, rotor axial position, rotational speed, appearance
		Fastener loosening breakage	Appearance, bolt torque
	Support	Base frame breakage Fastener loosening, breakage	Bearing temperature, transmitted torque, noise, vibration, appearance, bolt torque
External leakage	Pressure containment casing	Leakage at casing split	Leakage rate
		Fastener loosening, breakage	Appearance, torque
	Mechanical subsystems	Shaft seal deterioration, breakage	Seal leakage rate, rotor vibration, local shaft temperature

Table 14. Summary of important AUXFP part failure assessment

Part	Materials	Significant stressors/ failure causes	Measurable parameters
Shaft and fasteners	400-series S.S.	(a) Mechanical/breakage (b) Hydraulic/breakage and wear (c) Tribological/wear and seizing	(a), (b), and (c) vibration, bearing temperature, appearance, transmitted torque
Impellers	CrNi alloy steels, 17-4Ph	(a) Mechanical/breakage (b) Hydraulic/breakage and wear (c) Tribological/wear-surface wear	(a) and (b) rotor unbalance vibration, appearance, delivered flow; (c) rotor vibration, appearance
Thrust runners	400-series S.S.	(a) Mechanical/breakage, seizing (b) Hydraulic/breakage, seizing (c) Tribological/rubbing, lubricant dirt and breakdown	(a), (b), and (c) transmitted torque, rotational speed, rotor axial position
Stationary vanes (diffuser or volute)	400-series S.S.	(a) Hydraulic/breakage	(a) delivered flow, appearance, vibration
Wear rings	400-series S.S.	(a) Mechanical/seizing (b) Hydraulic/seizing (c) Tribological/wear-surface wear	(a), (b), and (c) vibration, transmitted torque, delivered flow, appearance, clearance
Thrust balancers	400-series S.S.	(a) Mechanical/breakage, seizing (b) Hydraulic/breakage, seizing (c) Tribological/wear	(a), (b), and (c) rotor axial position, transmitted torque, rotational speed, appearance
Thrust bearings	Rolling contact elements (Specialty steels)	(a) Mechanical/breakage (b) Hydraulic/breakage (c) Tribological/wear	(a), (b), and (c) rotor axial position, transmitted torque, rotational speed, appearance, clearance
Radial bearings	Bearing white metal (typically tin-base babbitt)	(a) Mechanical/breakage, seizing (b) Hydraulic/breakage, seizing (c) Tribological/seizing, wear	(a), (b), and (c) rotor vibration, bearing temperature, transmitted torque, appearance
Shaft seals	Stuffing-box or mechanical type	(a) Mechanical/breakage, wear (b) Hydraulic/breakage, wear (c) Tribological/wear	(a), (b), and (c) seal leakage rate, rotor vibration, local shaft, temperature
Coupling	Gear type (usually)	(a) Mechanical/breakage, wear (b) Hydraulic/breakage, wear (c) Tribological/breakage wear	(a), (b), and (c) rotor vibration, transmitted torque, appearance

9. SUMMARY AND RECOMMENDATIONS

9.1 Summary

AUXFPs are critically important safety-related equipment. Several design, operation, maintenance, and condition-monitoring factors of these pumps warrant investigation. For example, as detailed in Sect. 2.4, tighter specification of certain materials of construction and fabrication methods could potentially provide marked improvements in AUXFP durability and thus higher reliability.

As described in Sect. 6.3, present HHPS multistage centrifugal pumps suffer from years of dormant engineering research and development efforts in the United States. For example, bearing and thrust balancer designs may in some cases be based on marginally reliable definitions of the static and dynamic rotor loads produced by the complex fluid flow phenomena that occur within such pumps. Engineering test data of specific pump configurations over the full range of operating conditions and possible levels of wear are not always available to designers because such testing is not always performed on every specific configuration. This has often resulted in undersized and marginal load carrying components in similar pumps used in other (non-AUXFP) applications.

As described in Sect. 6.3.1, the question of a prudent bypass flow capacity (to minimize dynamic loads and vibration), which varies from design to design, needs to be further studied and clarified.

AUXFPs are now typically installed with virtually no monitoring devices for establishing data trends nor secondary bypass flow loops that would facilitate periodic regular testing over the full-flow ranges necessary to simulate the various emergency pumping scenarios. Also, there is presently no standardized requirement for scheduled disassembly and intensive pump internal inspection (as proposed in Sect. 7.2.3 and detailed in Sect. 8.1).

In response to these important considerations, the following observations are made.

9.1.1 Bypass flow criteria

Present practice should be reviewed with the intent of determining if present bypass flow rates are a significant contributor to wear and aging factors.

9.1.2 Secondary bypass flow test loop

Present practice should be reviewed with the intent of determining the degree of need for in-plant full-flow test loops to simulate the various emergency pumping scenarios. Such loops would permit the pump to be regularly "exercised" at typical emergency condition flow rates, rather than the potentially abusive bypass flow rates of most present installations. This would also permit performance monitoring and trend establishment at realistic operating conditions.

9.1.3 Monitoring and establishing trends

State-of-the-art monitoring capability should be studied in regard to its value in assessing wear and aging factors in AUXFPs. Monitoring factors to be studied should include at least the following: (1) orbital rotor vibration, (2) bearing temperature and noise, (3) rotor axial position, and (4) delivered head-capacity performance. Trends in this monitored information should be studied and correlations with wear and aging factors sought. Details of these topics are covered in Sect. 8.2.

9.1.4 Scheduled disassembly and detailed inspection

Periodic disassembly and detailed inspection of AUXFPs are necessities. The proposed inspection and component renewal factors are outlined in Sect. 7.2.3 and discussed in Sect. 8.1. The most likely time for these functions to be performed would be during each refueling period.

9.1.5 Pump specifications

It is recommended that, when new pumps are purchased, specifications be written to take advantage of the latest proven advances in pump design. Results from projects such as those being carried out by the EPRI[2] can be helpful in preparing such specifications.

9.2 Recommendations

It is recommended that the appropriateness and utility of the surveillance and monitoring methods described in this report be examined in the next phase of the investigation on AUXFPs. The measurable parameters identified also are to be evaluated in terms of effectiveness for use in detecting degradation and establishing degradation trends.

The relationship of the first-phase study to the NPAR Program strategy is illustrated by the cross-hatched part of Fig. 4.

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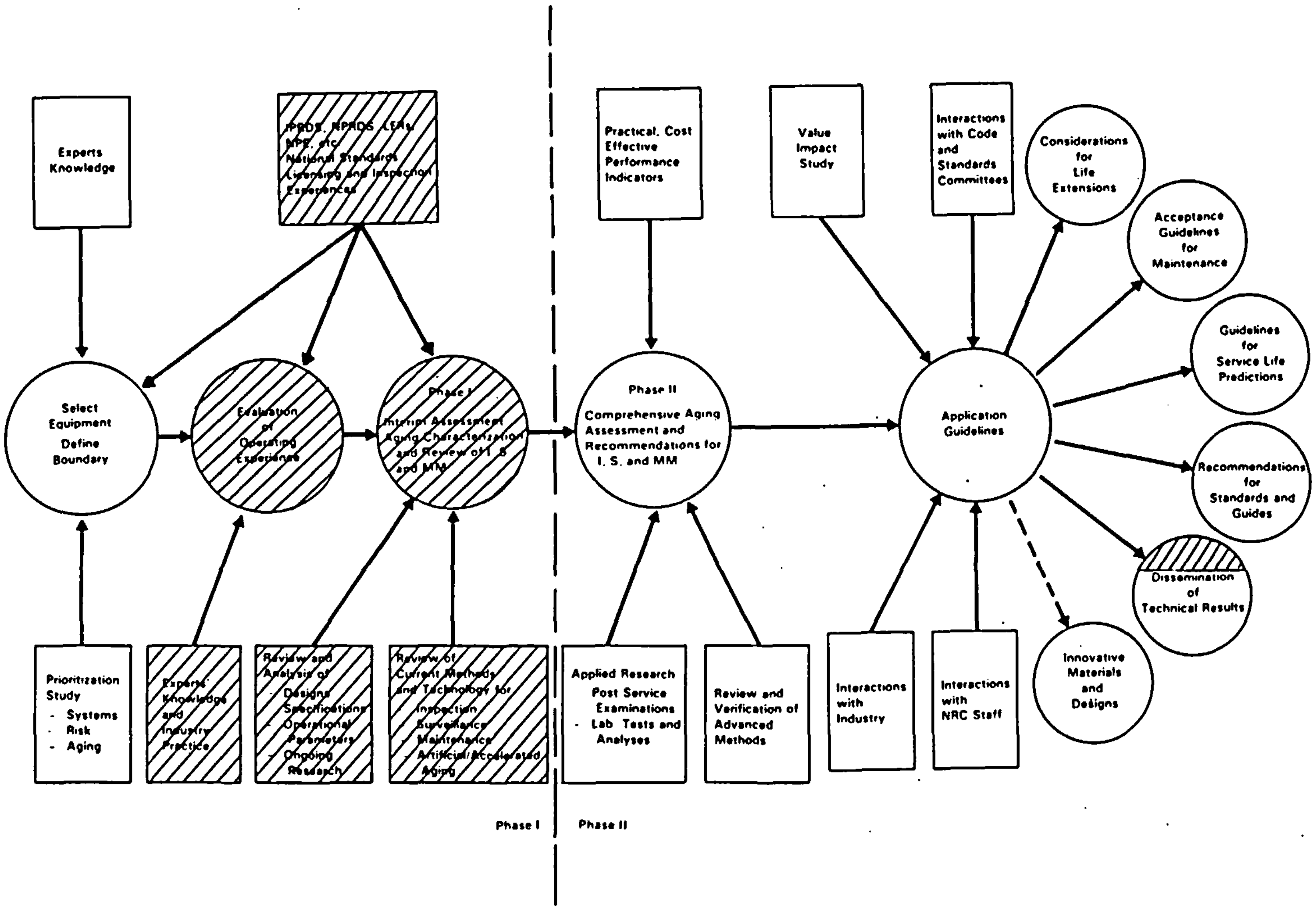

Fig. 4. NPAR program strategy.

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Appendix A*

SUMMARY OF *ASME BOILER AND PRESSURE VESSEL CODE*

SECT. XI REQUIREMENTS

The Sect. XI requirements for pumps are given in Subsection IWP, *In-service Testing of Pumps in Nuclear Power Plants*. This subsection discusses in-service testing of Class 1, 2, and 3 pumps required to achieve the cold shutdown condition for a reactor or to mitigate the consequences of an accident.

The in-service testing procedure given in Paragraph IWP-3100 is as follows:

> "An in-service test shall be conducted with the pump operating at nominal motor nameplate speed for constant speed drives and at a speed adjusted to the reference speed for variable speed drives. The resistance of the system shall be varied until either the measured differential pressure or the measured flow rate equals the corresponding reference value. The test quantities shown in Table IWP-3100-1 shall then be measured or observed and recorded as directed in this Subsection. Each measured test quantity shall then be compared with the reference value of the same quantity. Any deviations determined shall be compared with the limits given in Table IWP-3100-2 and the specified corrective action taken."

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*Work performed by G. A. Murphy, ORNL Nuclear Operations Analysis Center.

Table IWP-3100-1. In-service test quantities

Quantity	Measure	Observe
Speed N (if variable speed)	✓	
Inlet pressure P_i	✓[a]	
Differential pressure ΔP	✓	
Flow rate Q	✓	
Vibration amplitude V	✓	
Proper lubricant level or pressure		✓
Bearing temperature T_b	✓	

[a]Measure before pump startup and during test.
Source: *ASME Boiler and Pressure Vessel Code*, Sect. XI, IWP, p. 209 (1983).

Table IWP-3100-2. Allowable ranges of test quantities

Test quantity	Acceptable range	Alert range[a]		Required action range[a]	
		Low values	High values	Low values	High values
P_i[b]					
ΔP	0.93 to $1.02 \Delta P_r$	0.90 to $0.93 \Delta P_r$	1.02 to $1.03 \Delta P_r$	$<0.90 \Delta P_r$	$>1.03 \Delta P_r$
Q	0.94 to $1.02 Q_r$	0.90 to $0.94 Q_r$	1.02 to $1.03 Q_r$	$<0.90 Q_r$	$>1.03 Q_r$
V, when $0 < V_r < 0.5$ mil	0. to 1 mil	None	1 to 1.5 mil	None	>1.5 mil
V, when 0.5 mil $< V_r < 2.0$ mil	0 to $2V_r$ mil	None	$2V_r$ mil to $3V_r$ mil	None	$>3V_r$ mil
V, when 2.0 mil $< V_r < 5.0$ mil	0 to $(2 + V_r)$ mil	None	$(2 + V_r)$ mil to $(4 + V_r)$ mil	None	$>(4 + V_r)$ mil
V, when $V_r > 5.0$ mil	0 to $1.4V_r$ mil	None	$1.4V_r$ mil to $1.8V_r$ mil	None	$>1.8V_r$ mil
T_b[c]					

[a]See IWP-3230.

[b]P_i shall be within the limits specified by the Owner in the record of tests (IWP-6000).

[c]T_b shall be within the limits specified by the Owner in the record of tests (IWP-6000).

Source: Adapted from *ASME Boiler and Pressure Vessel Code*, Sect. XI, IWP, p. 210 (1983).

Paragraph IPW-3110 states the following:

> "Reference values are defined as one or more fixed sets of values of the quantities shown in Table IWP-3100-1 as measured or observed when the equipment is known to be operating acceptably. All subsequent test results shall be compared to these reference values or with new reference values established in accordance with IWP-3111 and IWP-3112. Reference values shall be determined from the results of the first in-service test run during power operation. Reference values shall be at points of operation readily duplicated during subsequent in-service testing."

Corrective action is given in Paragraph IWP-3230 as follows:

> "a. If deviations fall within the Alert Range of Table IWP-3100-2, the frequency of testing specified in IWP-3400 shall be doubled until the cause of the deviation is determined and the condition corrected.
>
> "b. If deviations fall within the Required Action Range of Table IWP-3100-2, the pump shall be declared inoperative and not returned to service until the cause of the deviation has been determined and the condition corrected.
>
> "c. Correction shall be either replacement or repair per IWP-3111, or shall be an analysis to demonstrate that the condition does not impair pump operability and that the pump will still fulfill its function. A new set of reference values shall be established after such analysis.
>
> "d. When tests show deviations greater than allowed (see Table IWP-3100-2), the instruments involved may be recalibrated and the test rerun."

Paragraph IWP-3400 specifies the frequency of in-service tests:

> "a. An in-service test shall be run on each pump nominally every three months during normal plant operation. It is recommended that this test frequency be maintained during shutdown period if this can reasonably be accomplished, although this is not mandatory. If it is not tested during plant shutdown, the pump shall be tested within one week after the plant is returned to normal operation.
>
> "b. Pumps that are operated more frequently than every three months need not be run or stopped for a special test, provided the plant log shows each such pump was operated at least once every three months at the reference conditions, and the quantities specified were measured, observed, recorded, and analyzed."

Paragraph IWP-3500 provides the duration of tests:

> "a. When measurement of bearing temperature is not required, each pump shall be run at least five minutes under conditions as stable as the system permits. At the end of this time at least one measurement or observation of each of the quantities specified shall be made and recorded.
>
> "b. When measurement of bearing temperature is required, each pump shall be run until the bearing temperatures (IWP-4310) stabilize, and then the quantities specified shall be measured or observed and recorded. A bearing temperature shall be considered stable when three successive readings taken at ten minute intervals do not vary by more than three percent."

Appendix B*

OPERATING EXPERIENCE DATA BASES AND REPORTS

Failure information obtained from LER, NPRDS, and IPRDS data bases is summarized below.

B.1 Nuclear Operations Analysis Center RECON Licensee Event Reports Data Base Survey

Abstracts of all LERs and reports issued prior to LERs by U.S. utilities are stored on the Nuclear Operations Analysis Center data base that can be accessed through the DOE RECON system. A search was made of this data base for all events indexed as auxiliary or emergency feedwater pumps. Each 100-word abstract was reviewed to determine (1) the mode of failure, (2) the mode of detection, (3) the maintenance activity, and (4) the cause of failure.

This review found 53 events out of a total of 1139 events involving AUXFPs during the time period 1973—1983. Results are summarized in Table B.1.

During the review of the 1139 events certain types of failures were noted that were excluded from this study: (1) failure to test; (2) obvious design errors including seismic analysis; (3) obvious operator errors; (4) failure of the pump driver (motor, turbine, diesel generator, etc.); (5) failure of a valve in the system; and (6) failure of instruments or controls. Only failures of the pump itself are included in Table B.1.

Bearing, packing, and seals are the primary failed parts. Failures of shafts, impellers, internals, and housings appear to be isolated cases. Forty-two percent of all failures were detected during testing of the equipment, while 29% were detected during pump operation. A few problems (6%) were identified during maintenance action.

Sixty-seven percent of the maintenance action was replacement or repair of damaged or worn subcomponents. In 6% of the events a modification was made to correct the deficiency. The major failure cause was insufficient or improper cooling or lubrication (23%). Improper maintenance accounted for 17%, while wear was the cause in 15% of the events. For 29% of the failures, the cause was either unknown or not stated.

B.2 Nuclear Plant Reliability Data System

The NPRDS data base was searched for AUXFP failures. Table B.2 is a summary of the results that contains the failure information derived from codes specified by the utilities. Out of a total of 70 events found, 14 were actual failures of the pump itself. In cases where more than one

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*Work performed by G. A. Murphy, ORNL Nuclear Operations Analysis Center.

Table B.1. Summary of AUXFP-type failures reported in LERs (1973–1983)

Item	Rate[a] (%)
Failed component	
Bearings	48
Packing and seal	30
Casing	4
Internal components	4
Impeller	2
Capacity	2
Shaft	2
Other	8
Methods of detection	
Testing	42
Operation	29
Maintenance	6
Not stated	23
Maintenance action	
Replacement	67
Repair	25
Modification	6
No repair required	2
Identified cause	
Lack of lubrication or cooling	23
Maintenance error	17
Wear/end of life	15
Design error	6
Crud	4
Operator error	2
Other	4
Not stated	19
Unknown	10

[a]Total of 53 events.

Table B.2. AUXFP-type failures reported in NPRDS data base (1974–1985)

Item	Rate[a] (%)
Failed component	
Packing/gasket	50
Bearings	38
Internal components	6
Shaft	6
Methods of detection	
Incidental observation	33
Surveillance testing	25
Routine observation	12
Audiovisual alarm	12
Operational abnormality	12
Special inspection	6
Maintenance action	
Repair/replace	94
Modify	6
Failure cause	
Wear	58
Lubrication	12
Binding	12
Aging	12
Abnormal stress	6

[a]Total of 14 failure reports.

cause code was given, only the first or the one supported by the event narrative was used.

The predominant failed subcomponents were pump packing and bearing failure, with only isolated failures of other subcomponents. Testing or observation detected most of the failures (88%), with 12% involving operational abnormalities.

Repair or replacement of subcomponents was the primary maintenance action (94%), with only 6% being modified. Wear was listed as the primary failure cause (58%). Lubrication, binding, and aging were 12% each.

B.3 In-Plant Reliability Data Study

The IPRDS contains equipment service histories obtained from maintenance records from four nuclear power plant sites covering six units. The various data are recorded in coded form rather than descriptive. Table B.3 lists IPRDS data for AUXFPs. A total of 12 age-related failures were found in 79 maintenance reports for these units. The listed failure mode is based on symptoms, including leaks (50%) followed by low output (25%). Vibration accounted for 8%, while 17% were unknown or not given. As reported in other data bases, repair or replacement of the subcomponent was the predominant maintenance action. Modifications were made in only 17% of the events. The failure cause (listed as failure mode in the other data bases) was seals and packing (50%). Bearing failures caused 17%, while personnel errors caused another 17%. Loose fasteners and couplings and design errors accounted for 8% each. The severity levels of the incidents were coded as incipient (75%) and degraded (25%).

Table B.3. In-plant reliability data study

Item	Rate[a] (%)
Failure mode	
Leak	50
Low output	25
Vibration	8
Other/unknown	17
Maintenance action	
Repair/replace	83
Modify	17
Failure cause	
Seals/packing	50
Bearings	17
Personnel error	17
Loose fastener/coupling	8
Design error	8
Severity	
Incipient	75
Degraded	25

[a]Total of 12 records.

Appendix C

AUTOMATIC TRIPPING AND FAILURE

An overspeed trip on turbine-driven AUXFPs gives the one automatic tripping action that is built into all turbine-driven AUXFPs. However, overspeed tripping at startup has been a problem. The cause (and thus available fixes) of this problem is well understood by operating personnel at some plants.

One recommendation by the major governor supplier is the addition of a separate oil-pressurizing pump for the governor, which comes on prior to the steam-valve-open action. Plant engineers are not favorable to this proposal because it involves an additional pump with additional potential reliability and maintenance problems. In at least one plant (Palo Verde), the corrective fix was to install an additional steam line (1 in.) to the drive turbine that is used for the initial part of startup and is opened before the main steam inlet valve is opened. This approach allows the speed control governor time to gain control of the situation and to achieve the operating speed without such a large overshoot as to overspeed trip the machine.

Appendix D

ENGINEERING INFORMATION RELEVANT TO AUXFP RELIABILITY

D.1 Introduction

Since the March 1979 TMI incident, considerable attention and scrutiny has been focused on auxiliary feedwater systems by the NRC and industry groups. This process is still in the phase of fact-finding, assessment, and determination of what industry-wide measures are required to ensure the high reliability appropriate to safety-related systems and equipment. AUXFPs are critically important safety-related components. Therefore, to provide further insight into the pertinent engineering areas, the following sections of this appendix present selected technical background information critical to high-head-per-stage centrifugal pumps.

D.2 Pump Hydraulic Instability

"Hydraulic instability" is now the term most commonly used in the centrifugal pump field to label the highly active unsteady flow phenomena that become progressively more pronounced the farther away from best-efficiency flow that a pump is operated. As clearly identified throughout the main body of this document, these unsteady flows are the most significant contributor to deterioration (i.e., aging and wear) of pump components because of the high-amplitude dynamic forces that they produce. In main feedwater systems, these unsteady flow phenomena also frequently produce hard-to-control conditions in the feedwater system, leading to transients, trips, and upset conditions.

Figure D.1 shows delineation of stable and unstable flow regimes for various classes (i.e., different specific speeds) of power plant pumps. It is based on a composite of field troubleshooting experience, shop tests, and laboratory tests; it represents what current hydraulic design technology can typically provide. Current EPRI-sponsored research[1] addresses this area of feed pump design technology.

Hydraulic instability has also been characterized by the label internal "flow recirculation," which occurs both at the inlet and discharge regions of a pump stage at off-design operating flows. These flow recirculation cells, as illustrated in Figs. D.2 and D.3, are highly unsteady, giving rise to large vibration excitation forces and hard-to-control flow pulsations in the entire pump loop. Figure D.4 illustrates the ramification of hydraulic instability with regard to the head-capacity characteristic of a pump. Figure D.4 shows how different pumps, all satisfying the same best-efficiency flow condition, can have a significantly different quality of hydraulic performance at off-design operating flows.

Figure D.5 shows how the vane-passing shock intensity increases with smaller impeller-diffuser vane tip clearance.

The feedwater pump is also an active element in the overall feedwater flow loop. Thus, proper analysis of pump system stability problems requires the complete feedwater loop to be treated as a connected system.

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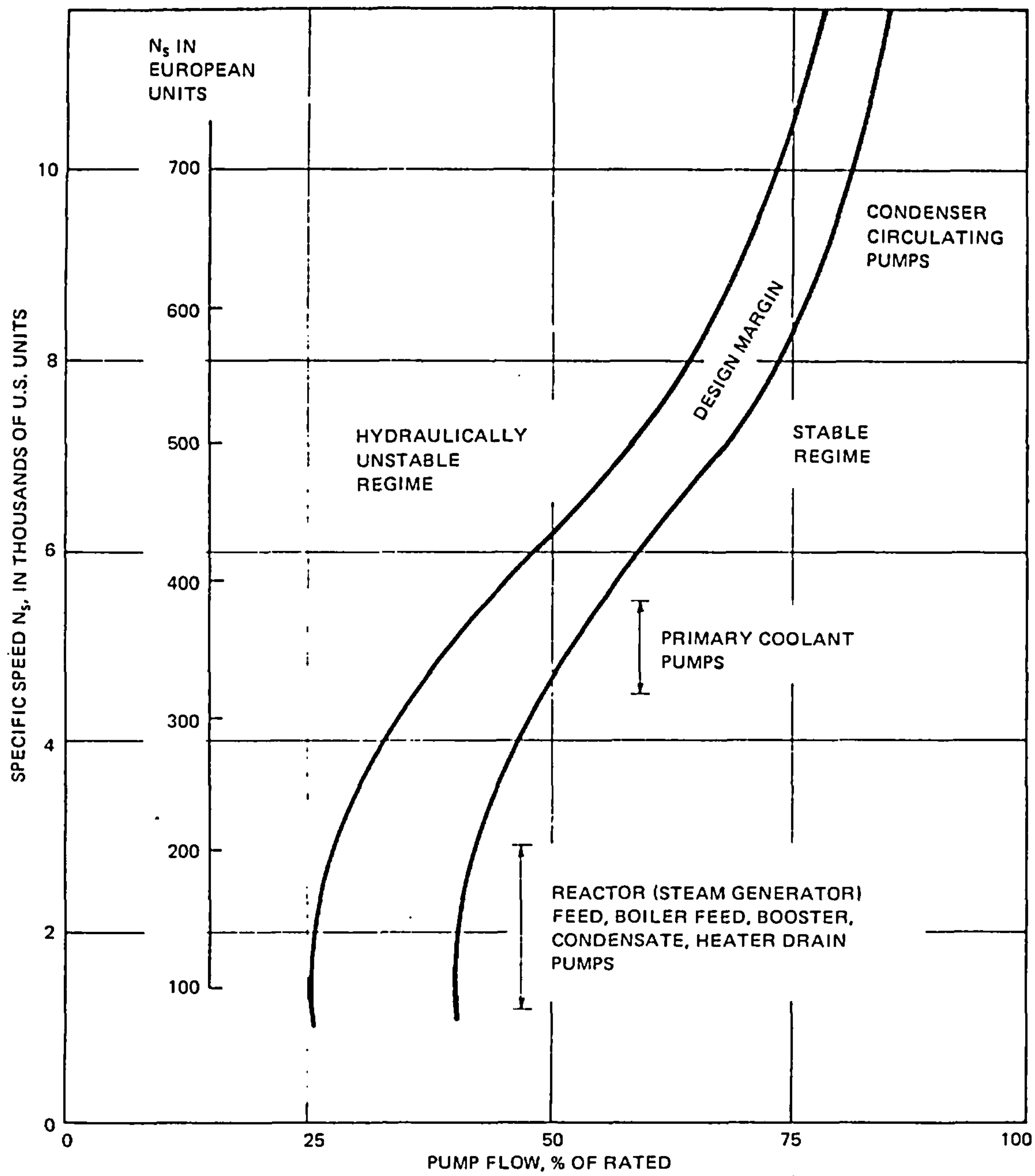

Fig. D.1. Anticipated useful operating ranges for pumps used in large nuclear and fossil power generating units. (Inner line of design margin area is preferred; if hydraulic instability occurs at higher flows various pump and system problems can be expected.) *Source*: E. Makay and O. Szamody, *Survey of Feed Pump Outages*, EPRI FP-754, Electric Power Research Institute, April 1978.

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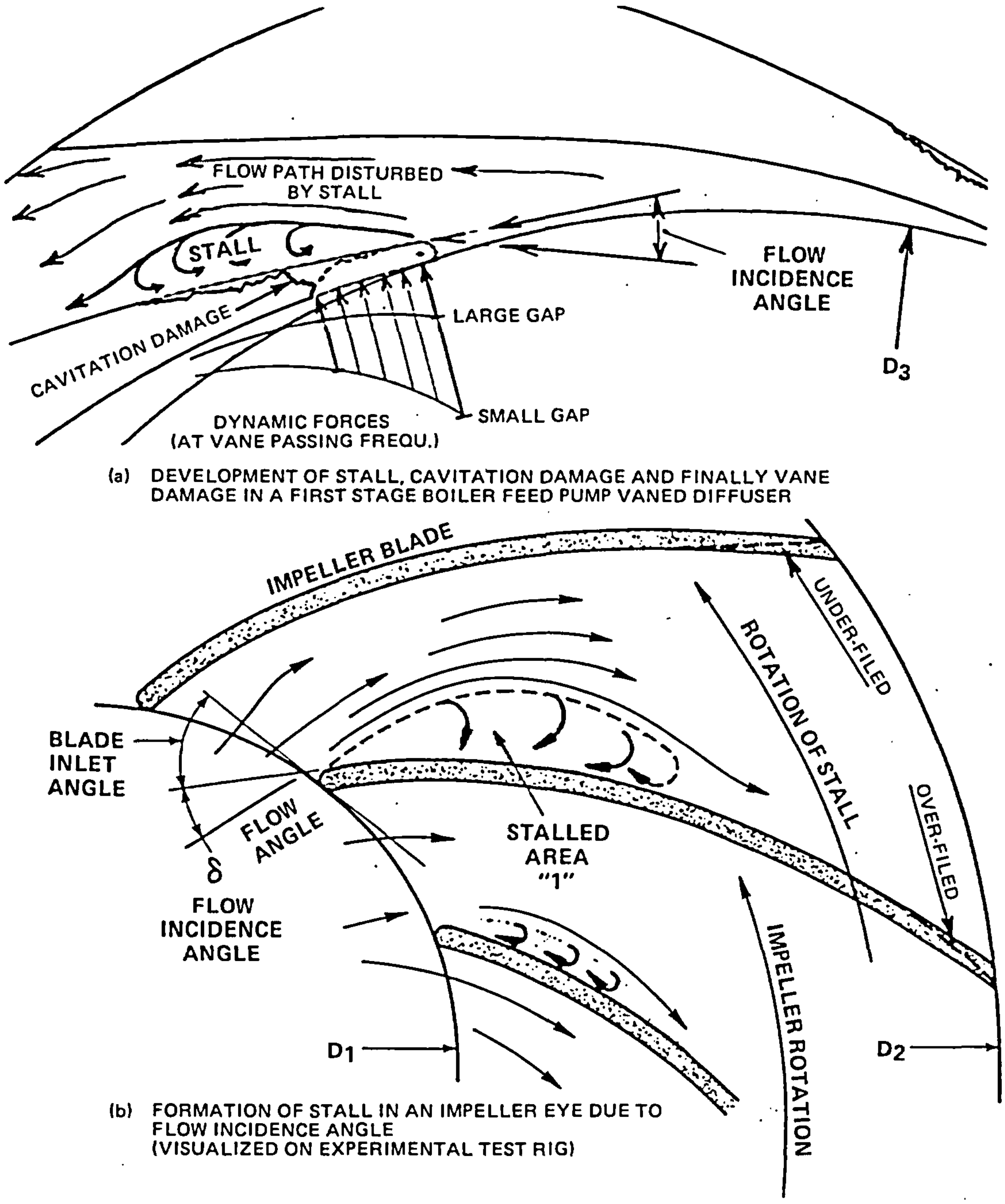

Fig. D.2. Formation of stall (*a*) in diffuser and (*b*) in eye of impeller. *Source*: E. Makay and O. Szamody, *Recommended Design Guidelines for Feedwater Pumps in Large Power Generating Units*, EPRI CS-1512, Electric Power Research Institute, September 1980.

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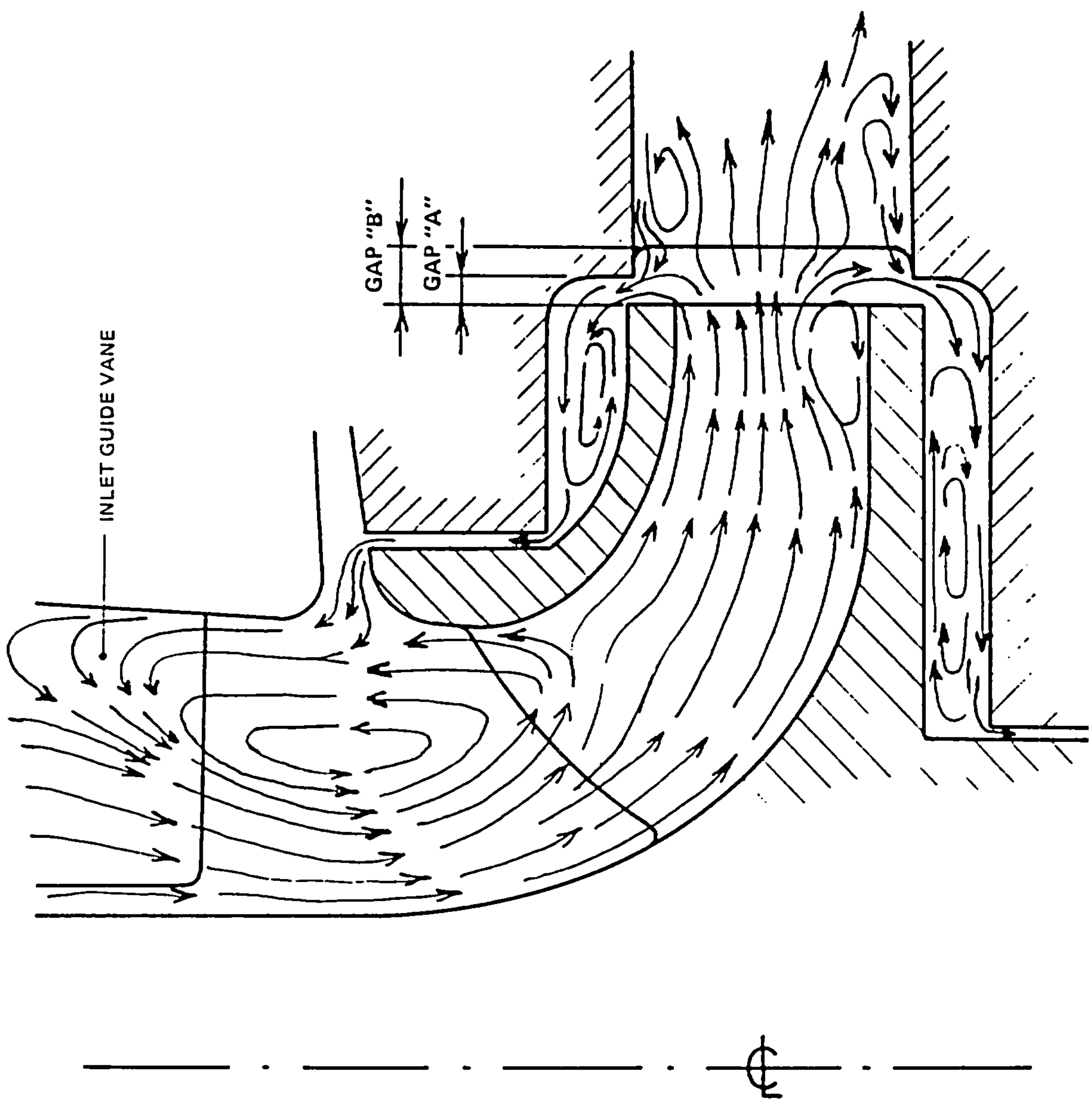

Fig. D.3. Secondary flow pattern in and around pump impeller stage at off-design flow operation. *Source*: E. Makay and O. Szamody, *Survey of Feed Pump Outages*, EPRI FP-754, Electric Power Research Institute, April 1978.

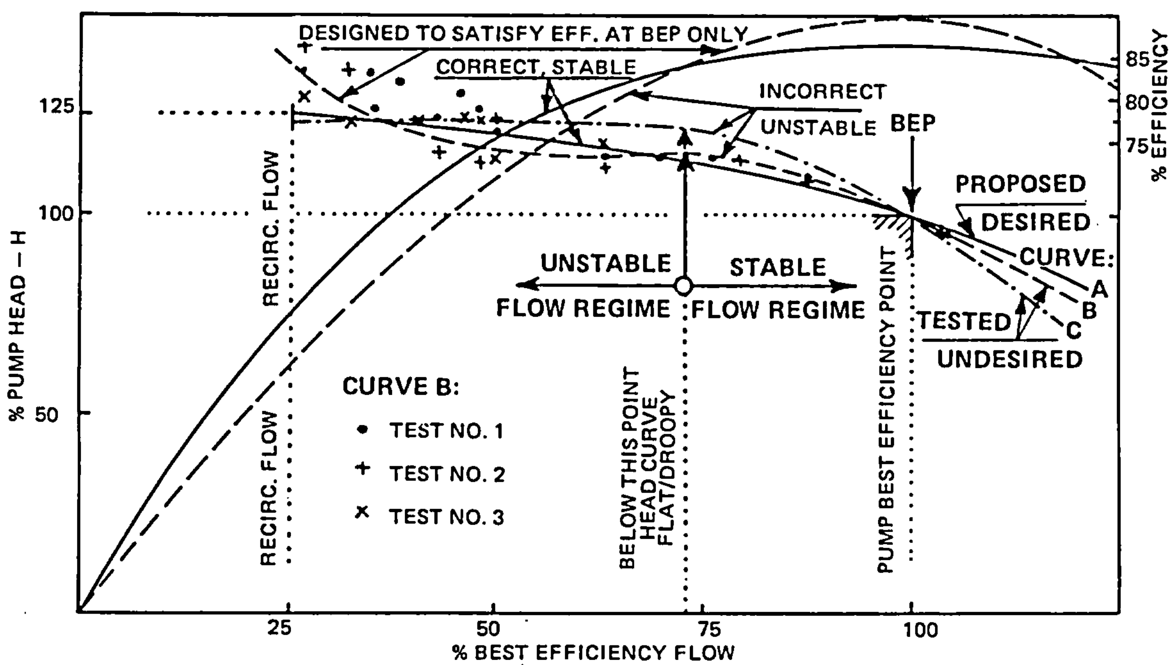

Fig. D.4. Head-capacity characteristics of multistage boiler feed pumps. Curve "A" is correct and desired for stable system operation. Curve "B" represents hydraulically unstable impeller-diffuser design. Parallel- as well as single-pump operation is difficult in the unstable flow regime. Curve "C" shows design with flat head curve at part load resulting in control system malfunctioning. Single-pump operation is possible in unstable regime. Designs "B" and "C" are not acceptable for utility applications. *Source*: E. Makay and O. Szamody, *Survey of Feed Pump Outages*, EPRI FP-754, Electric Power Research Institute, April 1978.

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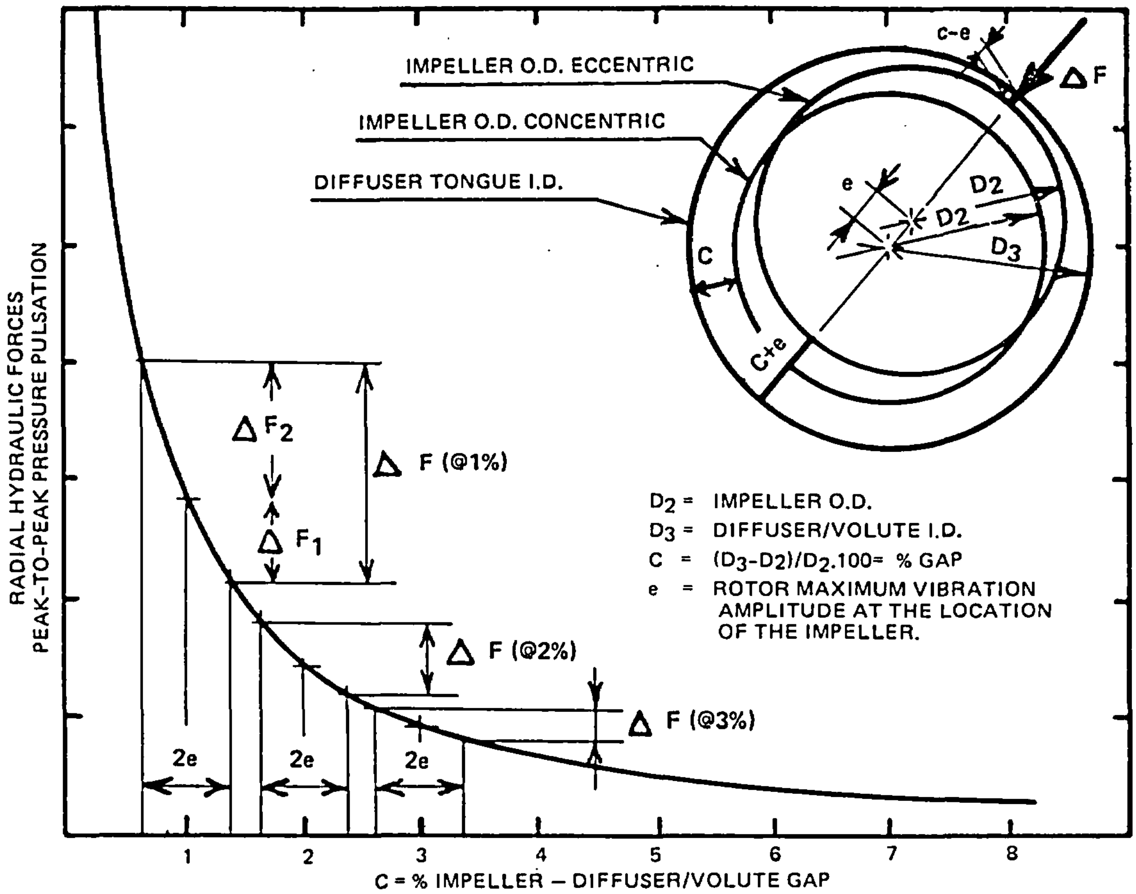

Fig. D.5. Influence of pump impeller to diffuser/volute radial gap on pressure pulsation at blade-passing frequency and rotor deflection caused radial forces. *Source*: R. Makay and O. Szamody, *Survey of Feed Pump Outages*, EPRI FP-754, Electric Power Research Institute, April 1978.

Laboratory tests[1] are now being conducted to determine the transfer matrix of pump stages for imbedding into a total system stability analysis.

D.3 Rotor Vibration

As described earlier, strong hydraulic forces at off-design flows are one major source of pump vibration. Other significant sources, include rotor unbalance, resonance, rotor-bearing instability (oil whip), internal rubbing, misalignment, coupling, and combinations of these. The use of spectrum analyzers has been an invaluable tool in diagnosing and correcting various pump vibration problems in the field. Through extensive field troubleshooting experience and related redesign work on high-pressure pumps, Fig. D.6 has been assembled to aid in isolating the fundamental cause(s) of excessive pump vibration. Figure D.6 uses pump flow as the independent parameter and is supplemented by Fig. D.7, which uses rotational speed as the independent parameter.

Rotor vibration levels, as usually measured at the bearing journals, can be evaluated as being acceptable or unacceptable, based on long experience of what various types of machinery can comfortably endure throughout the equipment's usable lifetime. Figure D.8 is typical of such guidelines used throughout the industry in assessing, from monitored vibration, whether the vibration levels are acceptably low to be consistent with high reliability and acceptable plant life.

D.4 Cavitation and NPSH

Boiler feed pump failures caused by cavitation erosion damage are among the highest-outage-producing pump problems, because cavitation frequently causes severe pump internal damage, requiring new internal components or lengthy factory repair. Cavitation is caused by the production of very low local pressures adjacent to flow boundaries, such that vapor-filled pockets form and then collapse violently as they are transported into higher pressure regions. Damage is most likely to occur in the inlet of the stage but may be carried through the impeller causing erosion of the impeller exit or the diffuser (or volute) inlet. Cavitation can be caused by:

1. inadequate NPSH of the feedwater system (i.e., not enough pressure at the pump suction);
2. flow recirculation at the impeller eye while operating in the off-design flow regimes;
3. incorrect hydraulic design of the first-stage impeller (incorrect blade inlet angle);
4. localized high velocities caused by sharp corners and other flow disturbances such as misplaced inlet guide vanes;

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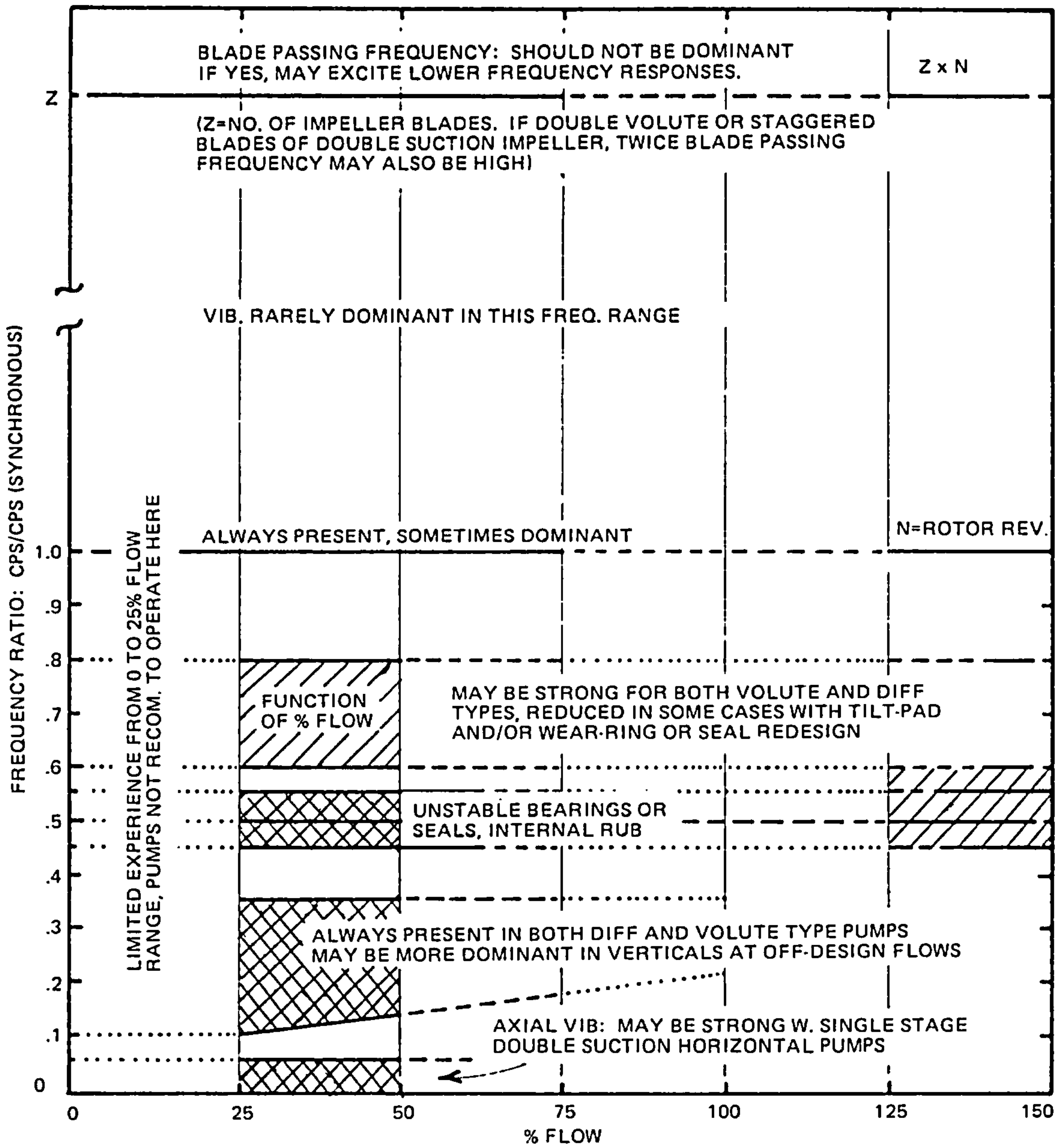

Fig. D.6. Frequencies of hydraulically induced dynamic forces acting on the rotor of a centrifugal pump. *Source*: E. Makay and O. Szamody, *Survey of Feed Pump Outages*, EPRI FP-754, Electric Power Research Institute, April 1978.

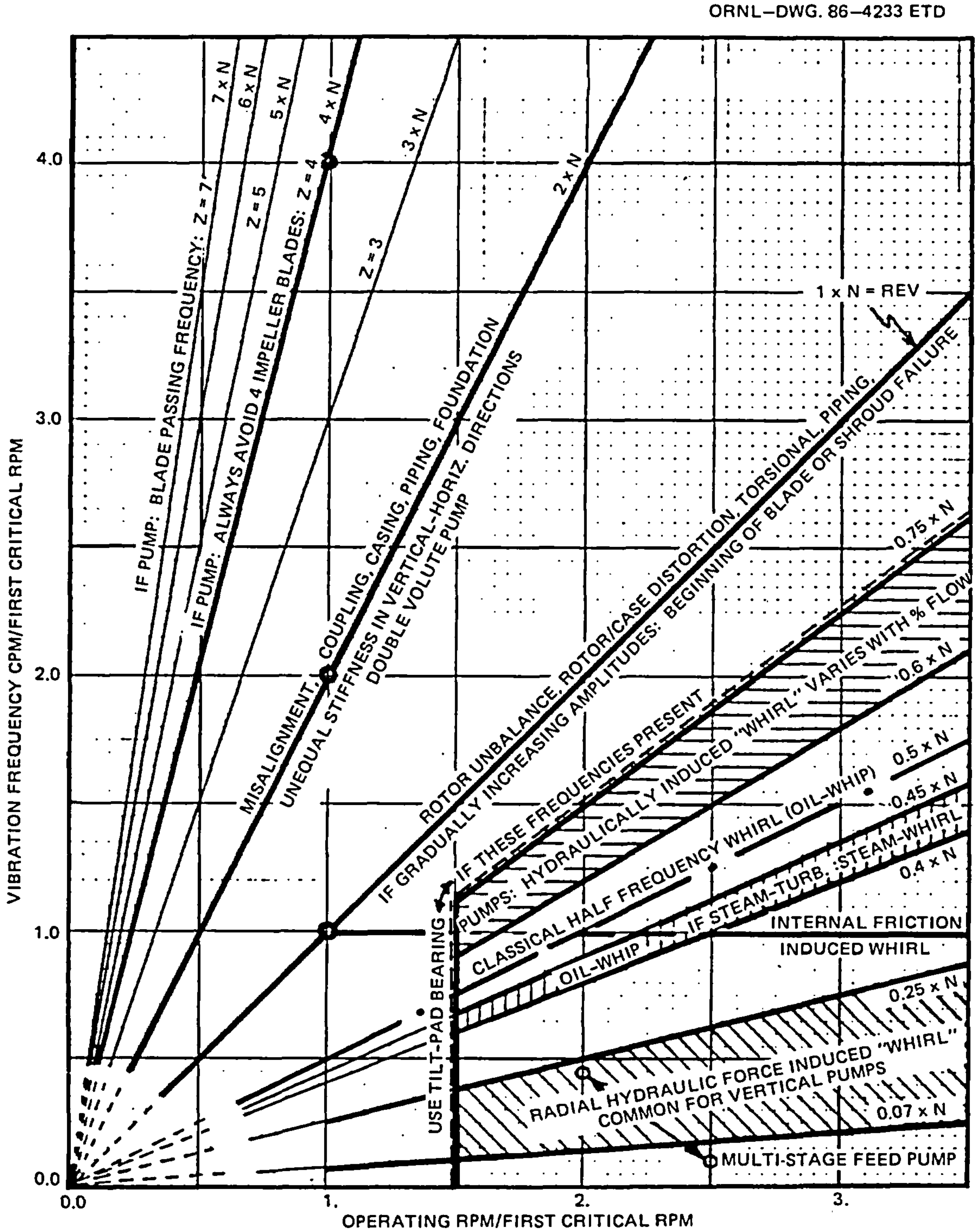

Fig. D.7. Vibration frequency vs speed. Causes and cures of common vibration problems of centrifugal pumps, steam and gas turbines, fans and blowers, compressors, and other centrifugal equipment. *Source*: E. Makay and O. Szamody, *Survey of Feed Pump Outages*, EPRI FP-754, Electric Power Research Institute, April 1978.

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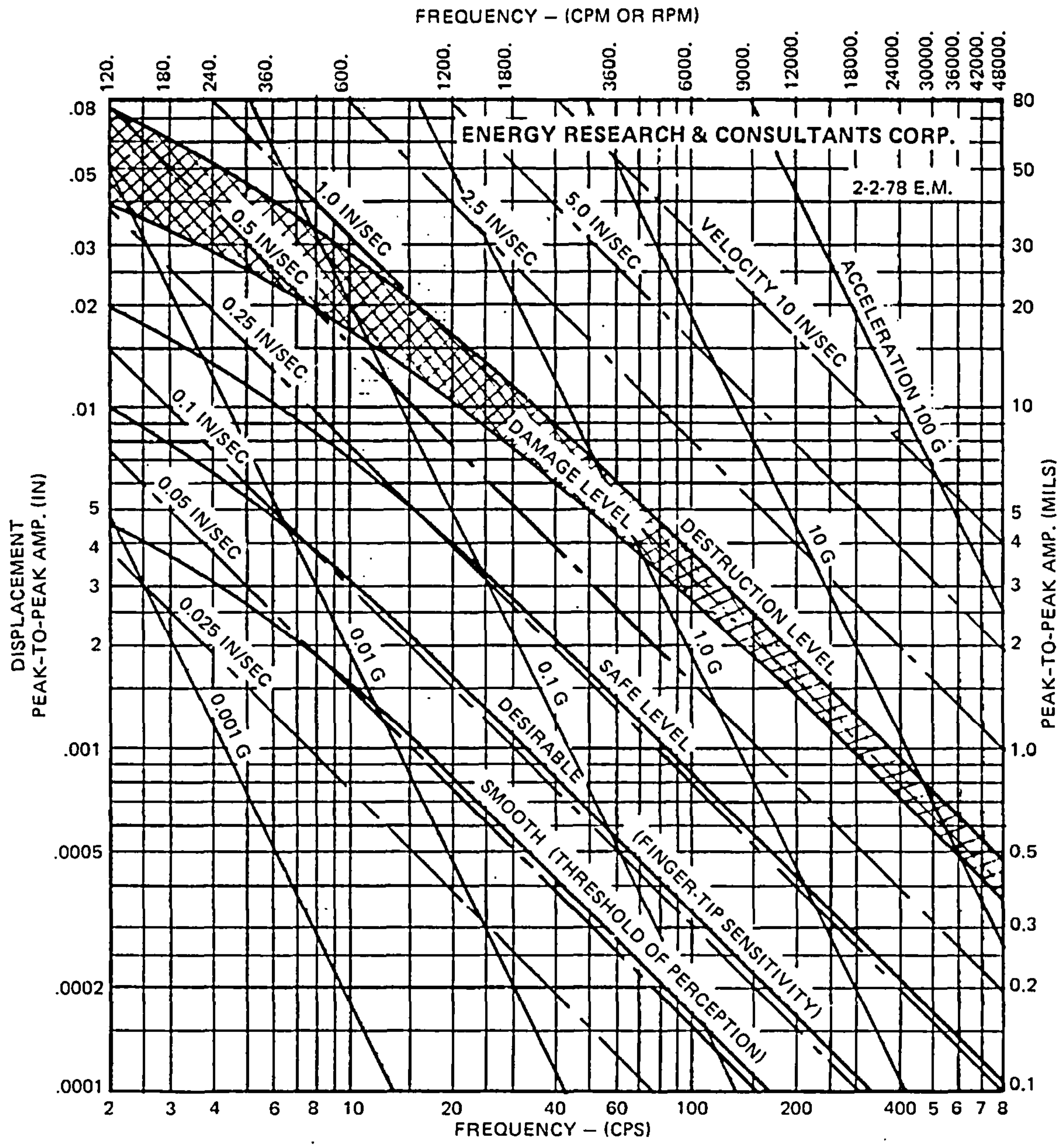

Fig. D.8. Allowable rotor vibration levels measured relative to bearing cap. (Values shown are filtered readings at that particular RPM or frequency; experimental standards). *Source*: E. Makay and O. Szamody, *Survey of Feed Pump Outages*, EPRI FP-754, Electric Power Research Institute, April 1978.

5. vortex formation due to obstacles in the flow path, sharp elbows in the suction piping, incorrect pump inlet geometry, and blunt inlet guide vanes; and
6. high-frequency machine vibration can displace water particles perpendicular to a solid surface creating vapor pockets, thus creating cavitation (least important type with centrifugal pumps).

Operation of pumps at off-design conditions for extended periods of time can cause cavitation damage independent of available NPSH, due to high-incidence-angle-caused stall and secondary flows like eye recirculation as shown in Fig. D.2 and D.3. In a multistage boiler feed pump, impeller cavitation damage usually occurs in the first stage, but it can occur at other locations where flow conditions satisfy the above requirements. It is important to distinguish between first-stage impeller cavitation and pump internal cavitation, because the latter is not related to pump NPSH. Feed pumps producing high-head/stage are more receptive to cavitation damage because of the higher energy-input densities to the fluid. Velocities and dynamic forces are high enough to accelerate cavitation and fatigue damage of pump internal components.

If cavitation damage occurs at design capacity due to insufficient NPSH, usually cavitation damage can be seen on either or both sides of the impeller blade inlet portion. The damage starts at the leading edge of the vane and may cover a large area. Another type of damage can be observed on the exposed side of the vane located in the corner where the blade joins the impeller hub. This type of damage indicates a mismatch between approach flow and impeller inlet angles that can be caused by extended operation of the pump in the low flow regime, even if NPSH is adequate to prevent cavitation. If severe impeller erosion appears somewhat downstream from the vane inlet edge at the periphery of the impeller eye, the damage may be caused by inlet flow recirculation (see Fig. D.3). The impeller is then operated in the off-design regime or the impeller eye is too large, causing development of flow recirculation at the impeller eye. If the damage starts from the vane inlet and is on the nonexposed side of the vane, then the pump is undersized for the application, that is, operated at substantially larger than best efficiency flow for extended periods of time.

D.5 Pump Internal Clearances

The critical clearances in a high-speed feed pump are those between rotating and stationary parts where high-pressure differentials exist (wear-rings of impellers, balancing device cylindrical surfaces, and seal surfaces) and the gap between impeller periphery and diffuser vanes or volute tongues. All these clearances affect pump efficiency as well as pump reliability. The closer the clearances, the higher the efficiency, but in most components, the lower the reliability (i.e., seizure and internal breakage are more probable).

The commonly used close clearance internal dimensions, which have evolved over years of experience, are nearly the same for all manufacturers and are suitable for reliable operation. If unreasonably high pump efficiencies are specified or demanded, the manufacturer is inclined to reduce these internal clearances below the commonly used values. Such a reduction of these clearances improves hydraulic efficiency because of the resulting reduction in interstage leakage, balancing device leak-off flow and seal flow. However, this efficiency improvement exists only during the factory acceptance test and for a short period of time in the field. The clearance surfaces wear-in to approximately the commonly used values, and the artificially produced higher efficiency vanishes. However, in the process the reliability of the pump is jeopardized by an increased potential for rotor seizure and rubbing-induced subsynchronous rotor vibration, either of which can result in destruction of the rotor and unexpected outage.

Reduction of the normal radial gap between impeller and diffuser (or volute) improves efficiency to some degree. As an impeller vane passes by a stationary blade (diffuser tip or volute tongue), a hydraulic shock occurs that can be observed in the liquid, on the structure, or on rotor vibration measured at the bearings or any part of the shaft. The influence of the radial gap on pressure pulsation at blade passing frequency and rotor-deflection-caused radial forces are shown in Fig. D.5. Numerical values are not given on the vertical scales, because they are also functions of other design parameters. The radial gap is given as a percentage of the impeller diameter. If the gap is too small (e.g., 1%) the phenomenon can be self-destructing, because the rotor exciting forces increase exponentially as shown in Fig. D.5. Rubbing at wear surfaces may also introduce subsynchronous vibration amplitudes that can rapidly destroy the rotor. If the impeller and stationary components are structurally marginal, the result can be disintegration of these elements. If these structures are strong, the result may be complete destruction of the whole rotating element. If such failure occurs, the radial gap is to be examined and if found too small, it is to be opened up to normal dimensions. Generally accepted dimensions are

Diffuser type, minimum gap: 3%

Volute type, minimum gap: 6%

D.6 Pump Component Design

A brief review of pump components subject to frequent failures or malfunctioning is given in this section. The experience was gained primarily on boiler feed pumps and therefore has equal applicability to AUXFPs, which are essentially of the same multistage high-head-per-stage configuration. In fact, AUXFPs are less robust than typical boiler feed pumps, because AUXFP designs appear to reflect the relatively small amount of design operating hours.

D.6.1 Axial balancing device

Axial forces in a boiler feed pump are, in the minority of cases, held in equilibrium by opposing equal number of impellers with a thrust bearing to assist the pump during startup and to take up the residual unbalance force caused by casting tolerances, minor dimensional differences, and wear. If the impellers all face the same way, either a balance drum or a balance disk is used to take the high axial thrust. When the impellers are opposed, although the forces are in equilibrium, some designs still utilize a balance drum for safety, because at part-load operation, large unpredictable hydrodynamic forces are produced that could damage the thrust bearing.

A balance drum is basically a rotating piston that has the characteristics of an ineffective water-lubricated radial bearing, so it influences the dynamic behavior of the rotor (Fig. D.9). The balance disk, on the other hand, is basically a water-lubricated hydrostatic thrust bearing, Figs. D.9 and D.10. The small gap "e" controls the pressure and consequently the thrust balancing pressure in cavity "A." If the gap "e" becomes too small or closes entirely, the faces will touch and destruction of the mating parts results. Introduction of a small taper between faces is an effective design improvement successfully recommended in numerous plants. It is the relative taper angle "alpha," and not the orientation of the faces, that is important. Figure D.10 shows the parallel and the tapered face designs. Also shown is a force balance diagram, clearly indicating the superior behavior of the tapered disk design. The disk force counterbalances the forces produced by the impellers, transmitted to the disk through the shaft. This force can easily have a magnitude over 100,000 lb and is responsible for internal damage in many pumps. Figure D.10 shows the force T_0, which is the force when the disk is closed. It is vital that the forces produced by the impellers can never be higher than T_0, otherwise failure results. Many

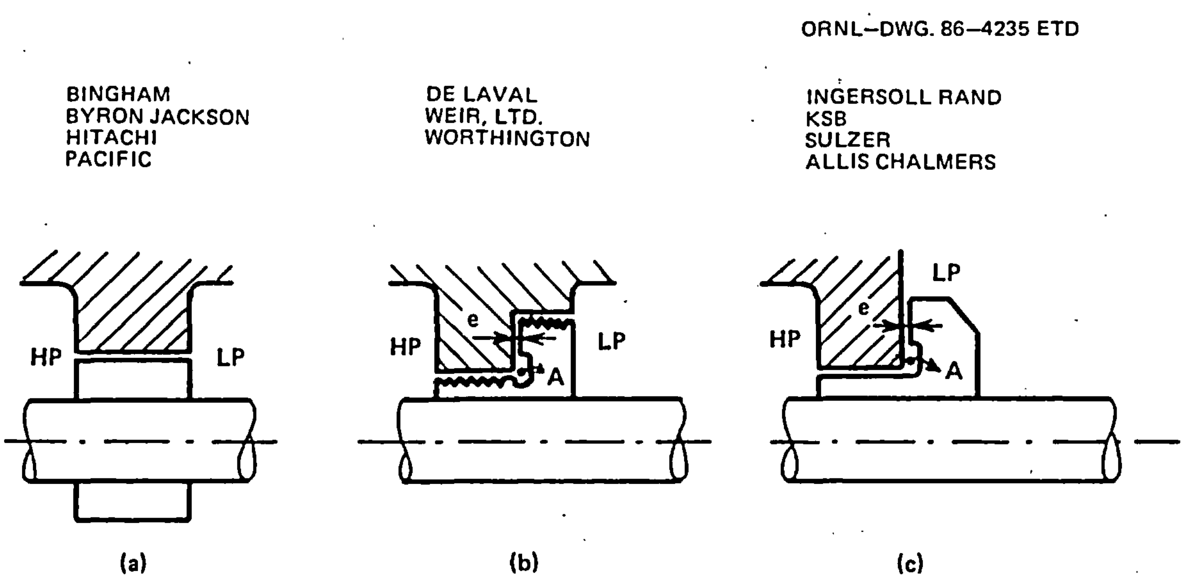

Fig. D.9. Customary axial balancing devices for high-pressure multistage boiler feed pumps. (*a*) Balance drum (*b*) and (*c*) balance disk. *Source*: E. Makay and O. Szamody, *Survey of Feed Pump Outages*, EPRI FP-754, Electric Power Research Institute, April 1978.

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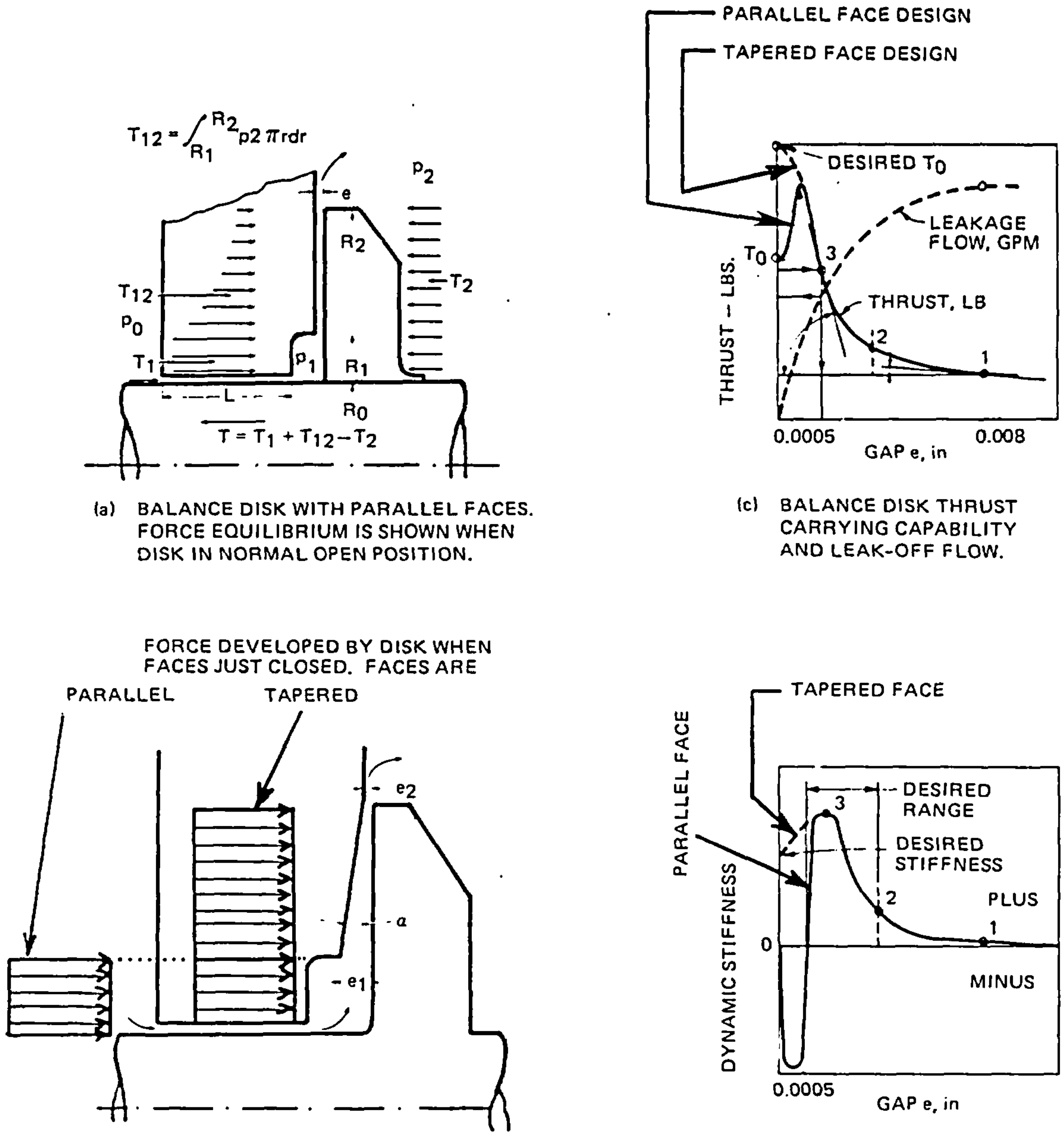

(a) BALANCE DISK WITH PARALLEL FACES. FORCE EQUILIBRIUM IS SHOWN WHEN DISK IN NORMAL OPEN POSITION.

(b) TAPERED FACE BALANCE DISK.

(c) BALANCE DISK THRUST CARRYING CAPABILITY AND LEAK-OFF FLOW.

(d) DYNAMIC STIFFNESS OF BALANCE DISK.

Fig. D.10. Parallel and tapered face balance disk designs. *Source*: E. Makay and O. Szamody, *Survey of Feed Pump Outages*," EPRI FP-754, Electric Power Research Institute, April 1978.

failures are due to the fact that the disk design load capacity is marginal when the pump is new. As the wear-ring surfaces wear with normal use, the hydraulic forces on the impellers grow, resulting eventually in a higher force than the marginal thrust-carrying capability of the disk. Figure D.10 shows the basic principal difference between parallel and tapered face designs, the tapered design having only advantages over the parallel. Note that when the balancing disk is approaching closed position, the tapered face design is able to take much higher thrust loads than the parallel face design, hence it is much more reliable, particularly under large transient loads that accompany severe hydraulic instability.

Complete disintegration of the rotating balance disk reported from one large generating station called attention to a typical failure mode. Customary material for that component is type 416 SS, where the material specification clearly states not to heat-treat above 42 Rockwell C hardness, because the possibility of surface cracking is high. When the failed component was tested for hardness, it was found to be over 50 Rockwell C. Other unused disks were tested and found to be over 42 Rockwell C hardness. Surface cracks were found severe enough to ensure future failures. This discrepancy was found in at least three manufacturers' products; thus, it was not a localized problem. It is a good practice to test for hardness every time a new disk is put in-service to intercept this type of failure cause. The best practice, however, is to heat-treat the stationary part to somewhat higher hardness, because that part is not subjected to centrifugal forces as is the rotating disk.

D.6.2 Shaft seals

Shaft seals have the highest failure rate for any component in boiler feed pumps. Figure D.11 shows the three seal designs used for boiler and nuclear feedwater pump services with single injection. Double injection is used for high-temperature applications to avoid flashing in the seal area. Injection water can be regulated by temperature or pressure control. Low-speed booster pumps may employ packing; however, high-speed applications (3600 rpm or above) are exempt from that seal application with the exception of AUXFPs. The majority of failures occur with seal designs other than labyrinth types. Labyrinth seals are the least demanding "work horses" of utility pumps.

Floating-ring type or mechanical seals have the highest failure rate among all pump component failures. When in good condition, the mechanical seals have lower leakage rate than the labyrinth or packing types but obviously are more prone to wear and failure. ~~After~~ some wear, the seal leakage increases rapidly, especially with the floating-ring type. This then requires excessive maintenance that results in lower pump, hence unit, availability or if not repaired, rapidly decreasing pump delivered capacity. Considering the high cost of unit down time to replace failed seals and the high repair cost with the mechanical or floating-ring seal types, it becomes obvious that the best seal type for large feed pumps is the labyrinth type. Several utilities report successful conversions to labyrinth seals after a long history of failures with the other seal types.

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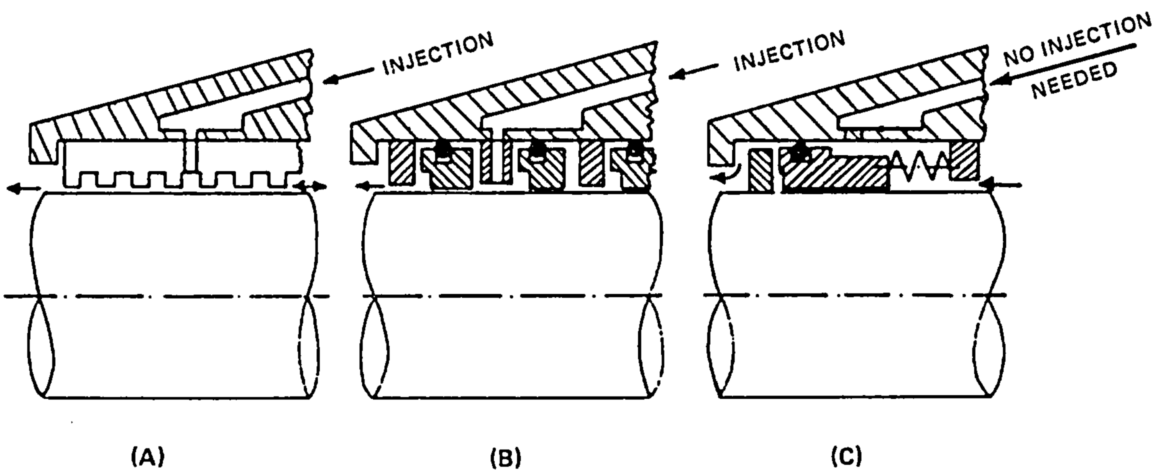

Fig. D.11. Shaft seal types used in boiler feed, nuclear feed, and feedwater booster pumps. (*a*) Labyrinth, (*b*) floating ring seals with single injection, and (*c*) mechanical seal. *Source*: E. Makay and O. Szamody, *Survey of Feed Pump Outages*, EPRI FP-754, Electric Power Research Institute, April 1978.

The demand of other seal types results from the higher apparent leakage rate of the labyrinth type. The leakage of a properly designed labyrinth seal is not higher than for a floating-ring type. Mechanical face seals, in spite of their high failure rate, are favored many times because they do not require external injection water, and when performing well, have the lowest leakage rate among all seal types. The tradeoff between apparent higher efficiency and pump reliability should be fully recognized.

D.6.3 Journal bearings and rotor

Practically all boiler feed pumps apply "flexible" shafts, which means that the pump operating speed is above the rotor first lateral critical speed. This is also true of most AUXFP designs. In addition to this, because of the relatively light rotor weight, the journal bearings are lightly loaded, which makes them prone to instabilities such as oil-whip (subsynchronous vibration component). Single-stage nuclear feed pumps in general operate below the first critical speed, but the rotor weight is even lighter than boiler feed pump rotors. The difficulty with light bearing loads is that the speed at which rotor dynamic instability starts is lower than normally expected. This speed is called the threshold speed, above which the bearing fluid film looses its ability to damp out rotor excitation forces at frequencies below approximately half the rotational frequency. Self-excited subsynchronous rotor whirl instability (oil-whip) may also occur. This loss of low-frequency bearing damping is particularly harmful in feed pumps (even at speeds below the threshold speed) because of the large low-frequency hydraulic forces that are produced by hydraulic instabilities, especially at part-load operation. Subsynchronous or low-frequency excitations also originate in the seals and wear-ring surfaces, induced both by fluid dynamical phenomena and by rubbing. Feed pump reliability would therefore be improved considerably by the development of advanced bearing and rotor configurations that introduce large amounts of low-frequency damping into the system.

A typical bearing system configuration for AUXFPs is shown in Fig. D.12.

D.6.4 Impeller breakage

Impeller breakage usually results in major pump damage, and causes are difficult to determine. Breakage is frequently the result of vibration, hydraulic instability, and cavitation damage. However, damage may also result from design deficiencies such as stress risers, inadequate strength, or faulty casting quality.

D.6.5 Shaft breakage

The existence of stress risers or material flaws can contribute to shaft breakage due to fatigue, but failure of other pump components will produce exceptionally high stress conditions. In most cases, the shaft failure is diagnosed as a secondary failure mode caused either by another component disintegration just prior to the shaft breakage or as a result of high-vibration amplitudes at impeller vane-passing frequency for an

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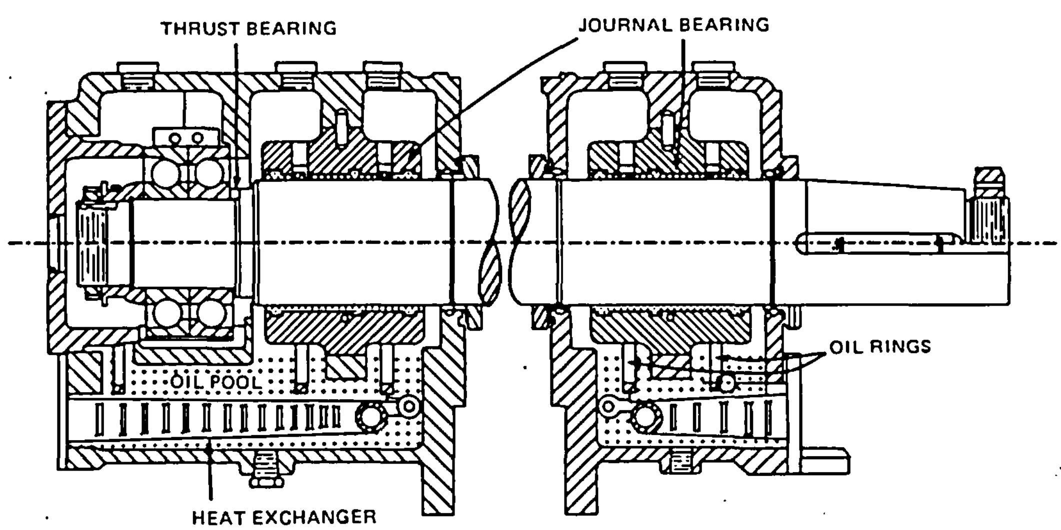

Fig. D.12. Typical AUXFP bearing system.

extended period of time. Frequent failures are a result of overloading the shaft by the high hydraulic forces acting on the axial balancing device. If the retaining mechanism of the disk (or drum) or the fitting of it is not proper on the shaft, very high cyclic forces result, accelerating fatigue failure (see Fig. D.13).

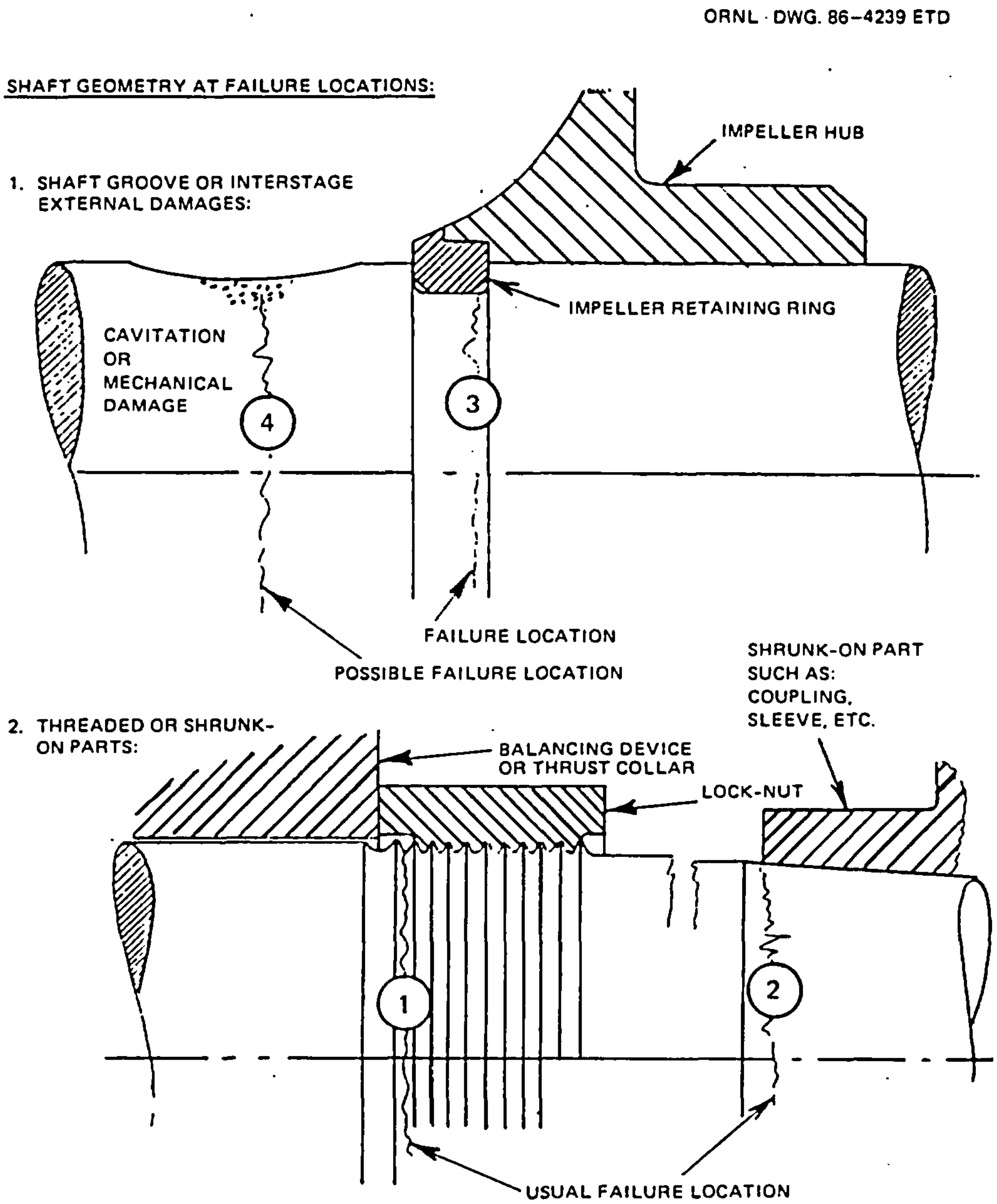

Fig. D.13. Typical multistage centrifugal pump shaft failure locations. The four most frequent locations are shown in order of failure frequency. *Source*: E. Makay and O. Szamody, *Recommended Design Guidelines for Feedwater Pumps in Large Power Generating Units*, EPRI CS-1512, Electric Power Research Institute, September 1980.

D.6.6 Wear rings

Excessive wear of the impeller wear rings is in most cases the result of excessive shaft flexibility and operation at conditions where large shaft vibrations are encountered; it can also follow journal bearing wear. Overly close wear-ring clearances in a new pump or new components will also lead to rapid wear of the rings. In any case, wear of these rings will make it impossible for pump efficiency to be maintained at the initial level. Therefore, new pumps should have sufficiently large clearances to ensure against rubbing or seizure. As with balancing disks, a proper choice of component materials is important for providing good accommodation of occasional rubs, thus avoiding consequential pump failures and premature replacement of wear rings.

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49-50. Technical Information Center, Department of Energy, Oak Ridge, TN 37831
51-425. Given distribution as shown in NRC category RV (10 - NTIS)

NRC FORM 335
(2-84)
NRCM 1102,
3201, 3202

U.S. NUCLEAR REGULATORY COMMISSION

BIBLIOGRAPHIC DATA SHEET

SEE INSTRUCTIONS ON THE REVERSE

1 REPORT NUMBER (Assigned by TIDC, add Vol No., if any)
NUREG/CR-4597,
Vol. 1, ORNL-6282/V1

2 TITLE AND SUBTITLE
Aging and Service Wear of Auxiliary Feedwater Pumps for PWR Nuclear Power Plants Volume 1. Operating Experience and Failure Identification

3 LEAVE BLANK

4 DATE REPORT COMPLETED

MONTH	YEAR
June	1986

5 AUTHOR(S)
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6 DATE REPORT ISSUED

MONTH	YEAR
July	1986

7 PERFORMING ORGANIZATION NAME AND MAILING ADDRESS (Include Zip Code)
Oak Ridge National Laboratory
P. O. Box Y
Oak Ridge, TN 37831

8. PROJECT/TASK/WORK UNIT NUMBER

9 FIN OR GRANT NUMBER
B0828

10 SPONSORING ORGANIZATION NAME AND MAILING ADDRESS (Include Zip Code)
Division of Engineering Technology
Office of Nuclear Regulatory Research
U.S. Nuclear Regulatory Commission
Washington, DC 20555

11a TYPE OF REPORT
Topical

b. PERIOD COVERED (Inclusive dates)

12 SUPPLEMENTARY NOTES

13 ABSTRACT (200 words or less)
Typical auxiliary feedwater pump (AUXFP) configurations are described in terms of configuration details, materials of construction, operating requirements, and modes of operation. AUXFP failure modes and causes due to aging and service wear are identified and explained, and measurable parameters (including functional indicators) for potential use in assessing operational readiness, establishing degradation trends, and detecting incipient failures are given.

A series of measures to correct present deficiencies in surveillance, monitoring, and in-service testing practices is discussed. The main body of the report is supplemented by a number of relevant appendixes; in particular, a major appendix is included on engineering and design information useful to assess AUXFP operational readiness.

This report was produced under the U.S. Nuclear Regulatory Commission's Nuclear Plant Aging Research Program.

14 DOCUMENT ANALYSIS - a. KEYWORDS/DESCRIPTORS
Pumps, centrifugal pumps, auxiliary feedwater pumps, aging, service wear, maintenance, degradation, trend, failure mode, failure cause, measurable parameter, incipient failure, inspection, surveillance, monitoring, functional indicators, operating experience

b. IDENTIFIERS/OPEN ENDED TERMS

15 AVAILABILITY STATEMENT
Unlimited

16 SECURITY CLASSIFICATION
(This page)
Unclassified
(This report)
Unclassified

17 NUMBER OF PAGES

18 PRICE