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NUREG/CR-4597
Volume 2
ORNL-6282/V2

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

Volume 2. Aging Assessments and Monitoring Method Evaluations

D. M. Kitch
J. S. Schlonski
P. J. Sowatskey
W. V. Cesarski

Prepared for the U.S. Nuclear Regulatory Commission
Office of Nuclear Regulatory Research
Under Interagency Agreement DOE 0551-0551-A1
NRC FIN No. B0828

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

Available from

Superintendent of Documents
U.S. Government Printing Office
Post Office Box 37082
Washington, D.C. 20013-7982

and

National Technical Information Service
Springfield, VA 22161

NUREG/CR-4597
Volume 2
ORNL-6282/V2
Dist. Category RV

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Manuscript Completed — May 31, 1988
Date Published — June 1988

Report Prepared by
Westinghouse Electric Corporation
Generation Technology Division
P.O. Box 355
Pittsburgh, Pennsylvania 15230
under
Purchase Order No. 41X-56549 V
for Oak Ridge National Laboratory

Work Performed for
U.S. Nuclear Regulatory Commission
under
DOE Interagency Agreement No. 0551-0551-A1
NRC FIN No. B0828

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37831
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U.S. DEPARTMENT OF ENERGY
under Contract No. DE-AC05-84OR21400

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ACKNOWLEDGMENTS

The authors wish to thank W. L. Greenstreet for his direction and a.vice in compiling this report. We also wish to thank the following equipment suppliers for their valuable discussions and input.

G. T. Morrissy	Pacific Pumps Division, Dresser Industries
D. Bowman	Pacific Pumps Division, Dresser Industries
J. McKenna	Ingersoll-Rand, Pump Division
L. Boswell	Byron Jackson Pump, Borg-Warner
D. Cummings	Bingham International
W. Lund	Crane Packing Company

EXECUTIVE SUMMARY

This report addresses aging assessments and monitoring method evaluations for auxiliary feedwater pumps (AUXFPs). It is a continuation of the findings reported in Vol. 1, *Operating Experience and Failure Identification.*

The report contains four major sections:

1. failure causes;
2. review of inspection, surveillance, and condition monitoring (ISCM) methods;
3. evaluation of ISCM methods; and
4. role of maintenance in alleviating aging and service wear.

The identification of failure causes is based on the results from Vol. 1 and Westinghouse postservice examinations and in situ assessments. Failure causes are attributable to various categories, such as

1. aging and service wear;
2. design and installation;
3. testing;
4. maintenance; and
5. other.

Pump failures attributed to aging and service wear are ranked according to frequency of occurrence, influence on operational readiness, and interaction consequences. An overall ranking was developed that allows the causes of failure to be listed in order of importance. Of the 20 failure causes listed in the report, the 15 most important are ranked. Following are the five that rank highest in importance: (1) bearing wear, corrosion, and breakage; (2) shaft seal deterioration and breakage; (3) binding between rotor and stationary parts; (4) impeller wear and breakage; and (5) thrust balancer wear, galling, and seizing.

Measurable parameters related to failure causes are identified. These parameters include rotational torque, appearance, pump speed, wear surface and critical fit clearance, pump head, delivered flow, vibration, balance return-line flow, noise, temperature, rotor axial position, fastener torque, leakage, lube oil purity, motor power, and surface indications. The ISCM methods that correlate to these measurable parameters are reviewed. Discussions of the relationships of identified ISCM methods to the detection of aging and service wear and to the establishment of degradation trends are included.

ISCM method recommendations are given. These were developed through evaluations based on Westinghouse experience and are intended to yield required capabilities for establishing operational readiness and determining degradation trends. Recommended ISCM methods include rotor binding inspection, visual inspection, dimensional inspection, audible noise inspection, bolt torque inspection, leakage rate inspection, lube oil analysis inspection, liquid penetrant inspection, rotational speed monitoring, developed head monitoring, delivered flow monitoring, vibration monitoring, balance return-line flow monitoring, incipient failure detection monitoring, bearing temperature monitoring, rotor axial position monitoring, and motor power monitoring. ISCM evaluations are summarized. They considered the effectiveness in determining aging and service wear, importance in terms of safety, and implementation ease and cost.

The role of maintenance in mitigating aging and service wear effects is discussed. Current utility practices and procedures are reviewed, as are vendor recommendations. Recommendations regarding inspection and surveillance practices for guiding maintenance actions are provided, and relationships between maintenance practices and ISCM methods are identified. Both regular maintenance and periodic inspection surveillance and maintenance practices are discussed. Also considered are recommended maintenance intervals. Predictive, preventive, and corrective maintenance practices are evaluated in terms of their effectiveness in addressing the identified failure causes.

Appendixes are included that contain definitions, failure data base information, AUXFP installation lists, discussion of low-flow testing, auxiliary feedwater (AFW) system description, AUXFP minimum-flow-rate criteria, and Westinghouse proposed guidelines for full-flow testing.

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ABSTRACT

This report was produced under the Detection of Defects and Degradation Monitoring Element of the Nuclear Plant Aging Research Program. It addresses aging assessments and monitoring method evaluations for auxiliary feedwater pumps (AUXFPs). The report contains four major sections:

1. failure causes;
2. review of inspection, surveillance, and condition monitoring (ISCM) methods;
3. evaluation of ISCM methods; and
4. role of maintenance in alleviating aging and service wear.

Failure causes attributable to aging and service wear are given and ranked in terms of importance. Cause identifications are made on the bases of experience, postservice examinations, and in situ assessments.

Measurable parameters related to failure causes are identified. ISCM methods are also identified, evaluations are made based on Westinghouse (W) experience, and recommendations are given. The methods are intended to yield required capabilities for establishing operational readiness, as well as for detecting and tracking degradation.

The role of maintenance in mitigating aging and service wear effects is discussed, and the relationship of maintenance to ISCM methods is identified. Predictive, preventive, and corrective maintenance practices are discussed and evaluated.

Appendixes have been included that contain failure data base information, AUXFP installation lists, discussion of low-flow testing, auxiliary feedwater system description, AUXFP minimum-flow-rate criteria, and W proposed guidelines for full-flow testing.

1. INTRODUCTION*

1.1 Background

This report is the second of three volumes on the study of aging and service wear of auxiliary feedwater pumps (AUXFPs) in pressurized-water reactor (PWR) nuclear power plants. This study program is sponsored by the Office of Nuclear Regulatory Research (RES) of the U.S. Nuclear Regulatory Commission (NRC).

The first volume was entitled *Operating Experience and Failure Identification* and addressed the time-related degradation of AUXFPs. This report builds on the results of the first by expanding on the identification, categorization, and ordering of aging and service wear related failures; providing a review and evaluation of inspection, surveillance, and condition monitoring (ISCM) methods; and discussing the role of maintenance in reducing or correcting aging and service wear effects. All definitions are listed in Appendix A.

1.2 Project Scope

Following is a summary of the three volumes.

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 1 has been published and issued as NUREG/CR-4597, Vol. 1 (ORNL-6282/V1) in July 1986.

Volume 2 — *Aging Assessments and Evaluation of Monitoring Methods (Phase 2)*

1. detailed identification of failure causes based on results from post-service examinations and in situ assessments; also classification and ordering of those causes;
2. review of currently used and developmental ISCM methods;
3. evaluation and recommendation of ISCM methods; and
4. examination of the role of maintenance in mitigating aging and service wear effects and recommendations.

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*Due to the nature of this subject matter and current practice in this field, SI units will not be used.

Baseline information used in the preparation of Vol. 2 was derived from Vol. 1 and other sources.

Volume 3 — *Analysis and Recommendations*

1. application of ISCM practices;
2. operational readiness acceptance criteria; and
3. criteria for maintenance, repair, or replacement.

The identification and analysis of failure causes in Vol. 2 are based on the premise that information in existing data bases regarding in situ testing and postservice examinations and tests is sufficient and that no additional examinations and tests supported by the Nuclear Plant Aging Research (NPAR) Program are needed. The results of the studies described in Vol. 2 attest to the validity of that premise.

2. FAILURE CAUSES

2.1 Results from Postservice Examinations and In Situ Assessments

Various data sources were used to assemble the results of this report. Westinghouse (W) sources include pump vendors (many of whom are involved in AUXFP manufacturing), architect engineers, nuclear and fossil utility companies, W site service representatives [located on many nuclear plant sites using W nuclear steam supply systems (NSSSs)], foreign architect engineers and pump vendors, chemical and petrochemical plants, and other NSSS suppliers. In particular, valuable information and discussions were provided through utility personnel and representatives from Ingersoll-Rand; Byron-Jackson; Pacific Pump Division, Bingham-Willamette; and the John Crane Packing Company. Also, W has performed (1) design evaluations on numerous pumping systems, (2) failure studies, and (3) analyses on numerous pump types used in the nuclear industry.

Other data sources retrieved through W facilities include reports about the High Head Safety Injection Pump (HHSIP). The HHSIP, although installed on the primary side of the nuclear plant, is functionally very similar to the AUXFP because it is an intermittently operating safety-related pump. HHSIPs are typically multistage, split-case, electric-motor-driven pumps that operate at 3600 rpm. They are similar in construction to AUXFPs except that HHSIPs are constructed of stainless steel and employ mechanical shaft seals instead of stuffing box packing. HHSIP data (Appendix B) are provided as a comparison to the AUXFP data. Other operating experience data base information is also provided in Appendix B. Appendix C contains installation lists for AUXFPs and provides general reference data.

2.2 Failure Causes

2.2.1 Failure cause identification

Failure cause nomenclature for this report was obtained from Vol. 1 to ensure that Vols. 1 and 2 are consistent. Specifically, only the failure causes considered to be proximate causes are addressed.

Table 2.1 is a master list of failure causes compiled from the studies and data base information of Vol. 1 and the postservice examinations and in situ assessments of Vol. 2. Also given in Table 2.1 are correlations between failure causes and failure modes. The latter are failure to operate, failure to operate as required, and external leakage. (See Vol. 1 for explanations of failure modes.)

Failure causes not identified by postservice examination and in situ assessments but identified in Vol. 1 include the following.

Table 2.1. Pump failure causes related to aging and service wear

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

[a]Failure mode designation:

1 - Failure to operate
2 - Failure to operate as required
3 - External leakage

Pump segment	Failure cause
Nonrotating internals	Structural damage to stationary vanes (diffuser or volute)
Pressure containment casing	Suction nozzle leak, breakage; discharge nozzle leak, breakage; fastener loosening, breakage
Support	Base frame breakage; fastener loosening, breakage

A failure cause not identified in Vol. 1 but identified from the postservice examination and in situ assessment data is casing rupture disk leak.

2.2.2 Failure causes attributable to various categories

Although the aging and service wear category is the focus of this report, other categories have also been reviewed:

1. aging and service wear,
2. design and installation,
3. testing,
4. maintenance, and
5. other.

Each of these is discussed in the subsections that follow.

2.2.2.1 Aging and service wear. Table 2.2 ranks pump failure causes attributed to aging and service wear according to (1) overall terms of importance, (2) frequency of occurrence, (3) interaction consequences, and (4) influence on operational readiness. The lowest number, unity, corresponds to the highest ranking. Rankings beyond 15 are not given because there is no meaningful basis for the assignment of larger numbers.

The ranking for overall terms of importance was obtained from a weighted average of the ranked categories, frequency of occurrence, interaction consequences, and influence on operational readiness. The weighting criteria used to establish the ranking for overall terms of importance was frequency of occurrence (low), interaction consequences (medium), and influence on operational readiness (high).

For most rotating machinery (e.g., pumps, compressors, and turbines), regardless of application, the most predominant failure causes appear to be (1) shaft seal deterioration and breakage and (2) bearing wear, corrosion, and breakage. The single most frequently reported failure cause attributed to normal wear was shaft seal (packing) leakage, but bearing wear, corrosion, and breakage ranked first in terms of overall importance. Each of the 15 causes ranked is described in more detail in the following paragraphs.

Bearing wear, corrosion, and breakage. Bearing-related problems rank first in overall terms of importance in the aging and service wear category. Bearing failures are the second most frequently reported failure and also have a significant impact on operational readiness.

Table 2.2. Pump failure causes attributed to aging and service wear rankings according to overall terms of importance, frequency of occurrence, interaction consequences, and influence on operational readiness[a]

Pump segment	Parts	Failure cause	O	FO	IC	OR
Rotating elements		Binding between rotor	3	3	4	2
		and stationary parts	6	9	3	3
	Shaft	Shaft breakage	4	4	5	7
	Impeller	Impeller wear, breakage				
	Thrust runner	Thrust runner wear, breakage	10	13	7	6
	Fasteners	Fastener loosening, breakage	8	7	10	10
Nonrotating internals	Diffusers or volutes	Structural damage to stationary vanes (diffuser or volute)	14	>15	14	13
	Wear surfaces	Wear-surface wear, erosion, corrosion, seizing	7	6	8	9
	Fasteners	Fastener loosening, breakage	12	10	12	12
Pressure containment casing	Casing	Leak at casing split	13	11	15	15
		Leak at casing rupture disk	>15	14	>15	>15
	Suction nozzle	Suction nozzle leak, breakage	>15	>15	>15	>15
	Discharge nozzle	Discharge nozzle leak, breakage	>15	>15	>15	>15
	Fasteners	Fastener loosening, breakage	>15	15	>15	>15
Mechanical subsystems	Bearings	Bearing wear, corrosion, breakage	1	2	2	1
	Shaft seals	Shaft seal deterioration, breakage	2	1	1	4
	Thrust balancer	Thrust balancer wear, galling, seizing	5	5	6	5
	Coupling	Coupling wear, breakage	11	12	9	8
	Fasteners	Fastener loosening, breakage	9	8	11	11
Support	Base frame	Base frame breakage	>15	>15	13	14
	Fasteners	Fastener loosening, breakage	15	>15	>15	>15

[a]These rankings are abbreviated as follows:

O - Overall terms of importance (weighted average of FO, IC, and OR)
FO - Frequency of occurrence
IC - Interaction consequences (one cause activating others)
OR - Influence on operational readiness

Even when properly applied and maintained, a rolling contact bearing will be subjected to normal fatigue wear and ultimately fail. Fatigue wear results from the cyclical shear stresses immediately below the bearing load carrying surfaces. However, premature bearing failure is normally associated with one or more of the following nonaging or service wear related factors:

1. defective bearing seats on shafts and in housings,
2. misalignment,
3. faulty mounting,
4. incorrect shaft and housing fit,
5. inadequate lubrication,
6. ineffective sealing (dirt and moisture),
7. vibration while the bearing is not rotating, and
8. passage of electric current through the bearing.

When the amount of lubrication is inadequate, the bearing will overheat, wear, and eventually fail. An excessive amount of lubricant can also produce an excessive temperature rise because of churning. Overheating creates a breakdown of viscosity in the lubricant, rendering it inadequate. Water has a similar degrading effect on oil viscosity. Also, water will begin to corrode the rollers, balls, or races of the bearing and ultimately lead to failure. This phenomena is characteristic of AUXFPs that normally remain idle. Because these pumps can be driven by steam turbines or are located near steam pipes, both of which produce high-humidity environments, the probability of the condensate accumulation in the lubricant can be high. Also, leaking packing or mechanical seals can result in lubricant water contamination.

Another factor that must be considered for idle equipment is false brinelling, which is vibration-fed through the foundation or piping system. This problem is caused by the continuous single-point impact of the rollers against the raceway. False brinelling can be mitigated by periodically rotating the shaft. Tables 2.3 and 2.4 show various failure mechanisms for antifriction and sleeve bearings, respectively.

Shaft seal deterioration and breakage. Shaft seals have the highest frequency of failure occurrence of any pump part. Their frequency of failure is disproportionately large compared with most other failure causes. Shaft seal failures rank highest in interaction consequences, in particular, the relatively frequently reported occurrences of water contamination in the lubrication system caused by leaking shaft seals.

The shaft seals used in AUXFPs are typically made of packing that is contained within a stuffing box located near each shaft end. Packing is the material used in an AUXFP to throttle or restrict feedwater leakage between the moving shaft or shaft sleeve and stationary part, normally called the "packing gland." Packing wear is a function of various factors, such as temperature, pressure, chemical action, shaft runout, shaft gyration, misalignment, pump vibration, and shaft surface condition. Figure 2.1 shows the typical cross section of a packed stuffing box. One characteristic feature of a packed stuffing box is that the packing must be permitted to leak to provide lubrication and cooling to prevent overheating and premature failure.

Table 2.3. Antifriction bearing degradation

Failure mechanisms	Resulting degradation																	
	Fracture/ separation				Deformation								Wear			Corrosion		
	Spalling/flaking	Cracks/heat cracks	Smearing + seizing	Cage fracture	Cage deformation	Indentations	Fragment denting	Brinelling/false brinelling	Ball path—widened	Ball path—skew	Ball path—uneven load zones	Fluting	Cage wear	Abrasive wear/wear	Overheating + burning + scuffing	Corrosion/etching	Fit corrosion/fretting	Rust staining
Assembly																		
Excessive, unevrn heat application	•																	
Hammer blows		•				•		•										
Improper tooling					•	•		•										
Loose/tight fits		•									•			•			•	
Distorted bearing housing	•							•		•								
Rotor unbalance											•							
Misalignment				•	•				•	•			•					
Operating conditions																		
Vibration Moving Stationary	•					•		•						•			•	
Current passage												•				•		
Life attainment/fatigue	•																	
Overload	•	•							•						•			
Design error		•	•											•				
Seal																		
Contamination						•	•						•	•		•		
Moisture ingress																•		•
Lubrication																		
Lack of lubricant			•	•									•	•	•			•
Excess of lubricant			•												•			
Unsuitable lubricant			•					•					•	•	•			•

Source: Reprinted with permission from H. P. Bloch and F. C. Gietner, *Practical Machinery Management for Process Plants, Vol. 2, Machinery Failure Analysis and Troubleshooting*, Gulf Publishing Company, 1983.

Table 2.4. Plain bearing (journal) degradation

Failure mechanisms	Resulting degradation												
	Fracture/ separation			Deformation			Wear				Erosion		
	Spalling	Cracking	Seizing	Deformation	Embedments	Uneven load patterns	Scoring + galling	Overheating + scuffing	Wear	Abrasive wear/scratching/grooving	Erosion	Cavitation	Corrosion
Assembly and manufacture													
Insufficient clearance			•					•					
Misaligned journal bearing				•		•							
Rough surface finish on journal								•	•				
Pores and cavities in bearing metal		•											
Insufficient metal bond	•												
Operating conditions and design													
General operating conditions											•	•	
Overload/fatigue	•	•										•	
Overload/vibration	•	•	•				•		•	•		•	
Current passage											•		•
Unsuitable bearing material			•						•				
Lubrication													
Contamination of lubricant			•		•		•		•	•			•
Insufficient or lack of lubricant		•	•	•			•	•	•	•			
Oil viscosity too low									•				
Oil viscosity too high									•				
Improper lubricant selection							•						
Lubricant deterioration									•				•

Source: Reprinted with permission from H. P. Bloch and F. C. Geitner, *Practical Machinery Management for Process Plants, Vol. 2, Machinery Failure Analysis and Troubleshooting*, Gulf Publishing Company, 1983.

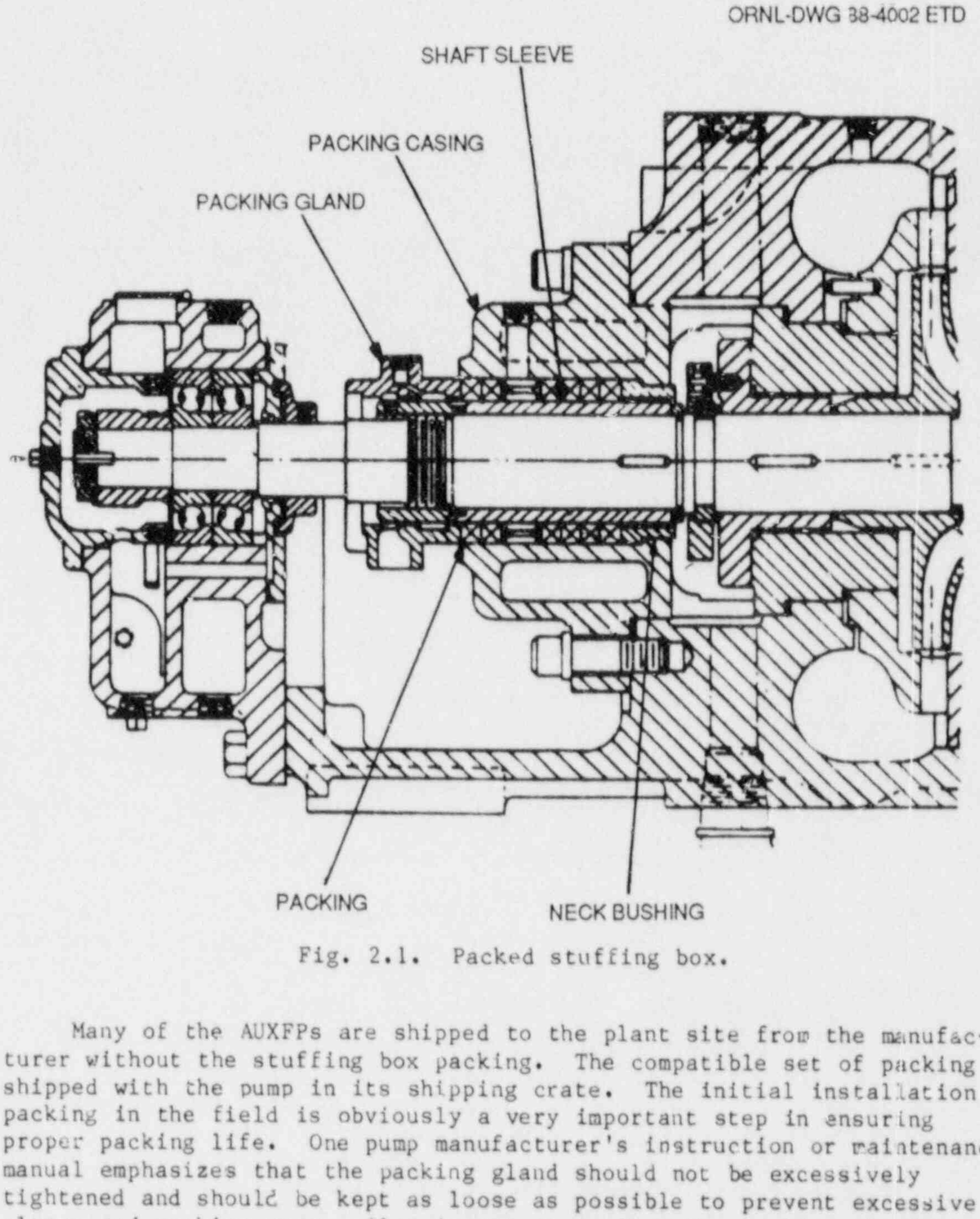

Fig. 2.1. Packed stuffing box.

Many of the AUXFPs are shipped to the plant site from the manufacturer without the stuffing box packing. The compatible set of packing is shipped with the pump in its shipping crate. The initial installation of packing in the field is obviously a very important step in ensuring proper packing life. One pump manufacturer's instruction or maintenance manual emphasizes that the packing gland should not be excessively tightened and should be kept as loose as possible to prevent excessive sleeve and packing wear. Also important is that the packing should be "broken in" after its installation to induce proper leakage. This task is accomplished by starting and quickly stopping the pump a few times until leakage is observed. If leakage fails to occur, the stuffing box is packed too tightly and should be disassembled and repacked.

Although not specifically identified in any of the data as being a failure cause, abrasives contained in the pumpage can also be a source of rapid packing wear and deterioration. When abrasives are a problem, a lantern gland must be installed into the stuffing box. Clean condensate can be introduced through the lantern gland to prevent abrasives from coming into contact with the packing or sleeves (Fig. 2.2). Flushing liquid should be introduced at 5 to 10 psi greater pressure than operating pressure.

The shaft sleeve surface finish and the packing gland internal surface finish have a direct relation to the packing life. One major packing supplier recommends that these finishes be between 16 and 20 μin. Rougher finishes will cause the packing to wear and polish these surfaces. However, during this process, both the packing and moving parts will deteriorate as well as "wear in." Proper finishes reduce the "wearing-in" time and increase the "wear-out" time. The packings must also seal against the inside diameter of the packing box (or box bore). Therefore, it is important that the finish be held to within 50 to 75 μin. If the shaft sleeve and packing gland surfaces are rough, they will not form a proper seal of the packing with the gland bore; consequently, in attempting to compress the packing sufficiently so that it will seal on the packing box bore, the packing will also be forced against the shaft and may be sealed off and thus overheat. Although the bore may be properly finished when the pump is new, corrosion often occurs at this location because there is so little movement of the fluid.

For normal multiring packing, the first few rings of packing absorb the bulk of the gland loading. These first few rings are also the rings that do the majority of the sealing. (This situation is analogous to a bolt-nut fastener combination: the first few engaged threads absorb a majority of the bolt load.) This loading condition is aggravated by improper individual ring seating at the time of initial installation and/or a rough surface in the bore of the stuffing box that restrains the rings and prevents them from sliding forward under gland adjustment (Fig. 2.3). The use of die-formed packing will help to alleviate this unbalanced loading condition.

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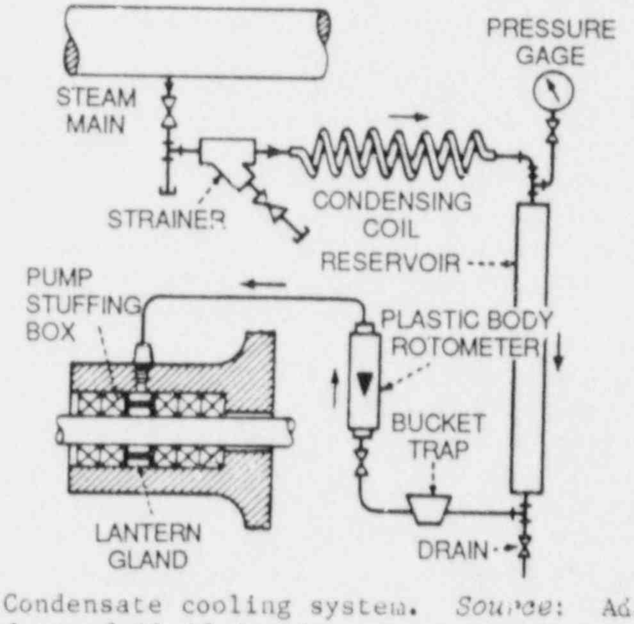

Fig. 2.2. Condensate cooling system. *Source*: Adapted with permission from *Engineered Fluid Sealing*, Crane Packing Co., 1976.

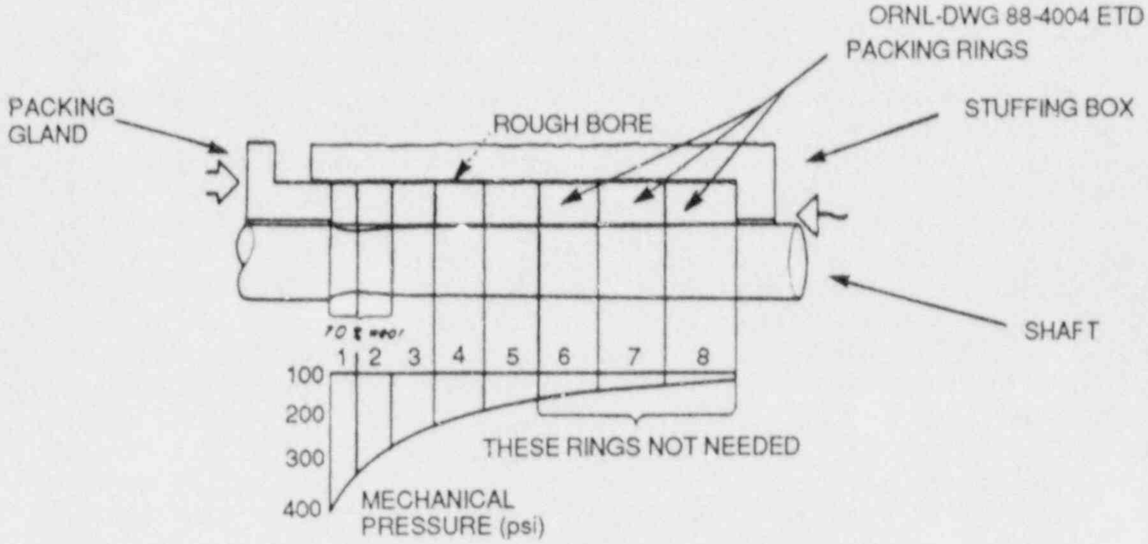

Fig. 2.3. Packing loading. *Source*: Adapted with permission from *Engineered Fluid Sealing*, Crane Packing Co., 1976.

A technique now being used on valves in the nuclear industry is the use of "live loading" with springs. However, this technique has not yet been widely applied in the pump industry. One form of spring-loaded packing is shown in Fig. 2.4. In this arrangement, the spring compensates for any slight wear or compression of the packing. The gland is tightened to the face of the stuffing box, and no further adjustment is

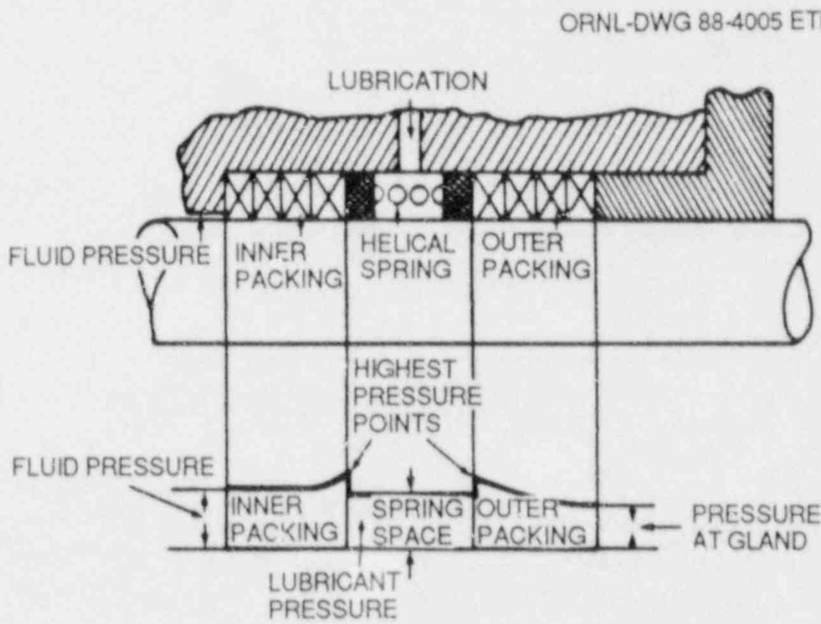

Fig. 2.4. Live-loaded pump packing. *Source*: Adapted with permission from *Engineered Fluid Sealing*, Crane Packing Co., 1976.

needed. (This arrangement is analogous to a cartridge mechanical seal: the assembly is preset and requires no final adjustment).

High vibration caused by worn bearings, unbalance, misalignment, or a bent shaft will also cause premature aging and wear of packing or mechanical seals. Table 2.5 identifies the more common causes of packing problems.

Mechanical seals are also installed on a small number of AUXFPs. A few of the AUXFPs were manufactured with mechanical seals in the original installation. Several utilities have chosen to replace the packing in AUXFPs with mechanical seals. Figure 2.5 shows a typical mechanical seal assembly. A mechanical seal is composed of a primary seal and secondary seals. The primary seal consists of a primary ring that normally rotates and floats against the face of a stationary seat. The primary ring is often carbon, whereas the seat is tungsten carbide. Other combinations include tungsten carbide against tungsten carbide and silicon carbide against silicon carbide. The secondary seal consists of a bellows and O-rings and is normally constructed of an elastomer material.

Mechanical seal age-related failures result from normal "nose" wear-out of the primary ring, overheating of the seal faces (which could result from dirt or poor coolant flow), and high shaft vibration. Leakage also results from aging, which causes a loss of elasticity and sealing capability of the secondary elastomer seals. Set-screws are often used to fasten parts to each other, such as the primary ring cartridge assembly to the shaft sleeve. Loose set-screws are a common problem with mechanical seals and can result in misalignment of seal parts and, ultimately, seal failure.

<u>Binding between rotor and stationary parts</u>. Rotating element binding or seizure often leads to total inoperability of the pump. Binding

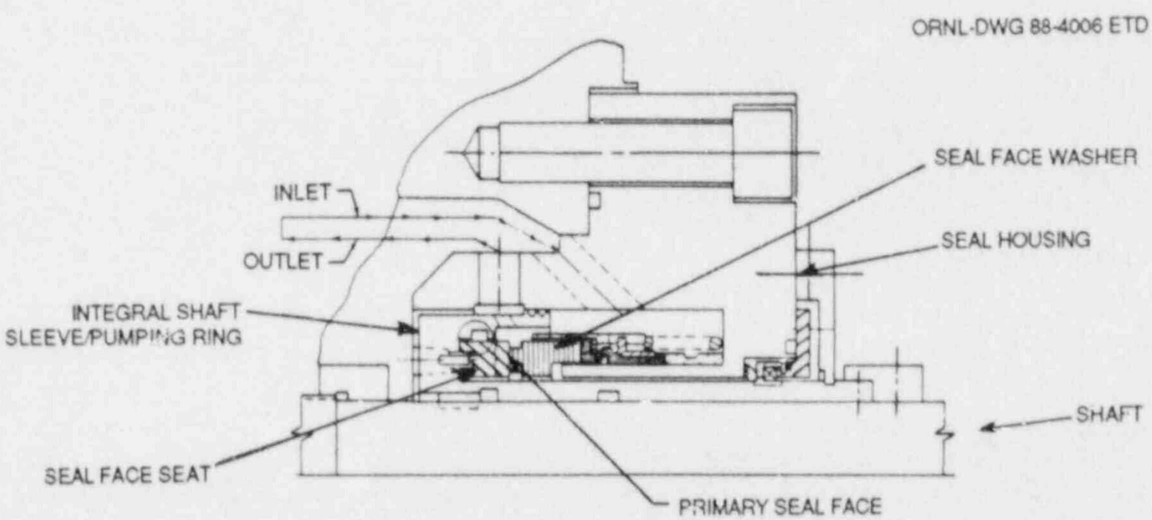

Fig. 2.5. Mechanical seal assembly. *Source*: Adapted with permission from David M. Kitch, "Pump Selection and Application in a Pressurized Water Reactor Electric Generating Plant," *Proceedings of the Second International Pump Symposium*, Texas A&M University, April 1985.

Table 2.5. Packing troubles, their cause and cure

Trouble	Cause	Cure
No liquid delivered	Lack of prime	Packing too loose or defective, allowing air to leak into suction Tighten or replace packing and prime pump
Not enough liquid delivered	Air leaking into stuffing box	Check for some leakage through stuffing box while operating; if no leakage after reasonable gland adjustment, new packing may be needed or Lantern ring may be clogged or displaced and may need centering in line with sealing liquid connection or Sealing liquid line may be clogged or Shaft or shaft sleeve below packing may be badly scored and allowing air to be sucked into pump
	Defective packing	Replace packing and check smoothness of shaft or shaft sleeve
Not enough pressure	Defective packing	Same as for preceding
Pump works for a while and quits	Air leaks into stuffing box	Same as for preceding
Pump takes too much power	Packing too tight	Release gland pressure Retighten reasonably; keep leakage flowing — if none, check packing, sleeve, or shaft
Pump leaks excessively at stuffing	Defective packing	Replace worn packing; replace packing damaged by lack of lubrication
	Wrong type of packing	Replace packing not properly installed or run-in; replace improper packing with correct grade for liquid being handled
	Scored shaft or shaft sleeves	Put in lathe and machine true and smooth or replace

Table 2.5 (continued)

Trouble	Cause	Cure
Stuffing box overheating	Packing too tight	Release gland pressure
	Packing not lubricated	Release gland pressure and replace all packing if any burnt or damaged
	Wrong grade of packing	Check with pump or packing manufacturer for correct grade
	Insufficient cooling water to jackets	Check if supply line valves opened or line clogged
	Stuffing box improperly packed	Repack
Packing wears too fast	Shaft or shaft sleeve worn or scored	Remachine or replace
	Insufficient or no lubrication	Repack and make sure packing loose enough to allow some leakage
	Improperly packed	Repack properly, making sure all old packing removed and box clean
	Wrong grade packing	Check with pump or packing manufacturer
	Pulsating pressure on external seal liquid line	Makes packing move and prevents it from taking a "set"; remove cause of pulsation

Source: Reprinted with permission from John Crane Packing Co., *Engineered Fluid Sealing*, 1976.

can result from anything that causes the loss of clearance in (1) impeller to nonrotating internal wear surface clearance, (2) balance drum to nonrotating bushing wear surface clearance, or (3) thrust runner to bearing surface clearance. Factors that cause increased changes in these internal clearances are (1) pump starts and stops, (2) overheating caused by pump operation with a closed discharge and miniflow valve or a closed suction valve, (3) worn bearings, (4) shaft misalignment, (5) dirt or debris, or (6) improper material hardness selection.

Impeller wear and breakage, wear-surface wear, erosion and corrosion seizing. Centrifugal pump impellers have wear surfaces that are normally located on both the front and rear hub. The impellers rotate within the wear surfaces (or rings) of the nonrotating internals. These surfaces form internal seals that restrict leakage from stage to stage. Figure 2.6 shows an impeller and wear-ring combination. As these internal wear-ring clearances increase, the stage-to-stage internal leakage increases and the pump efficiency decreases. Figure 2.7 shows the effect of this

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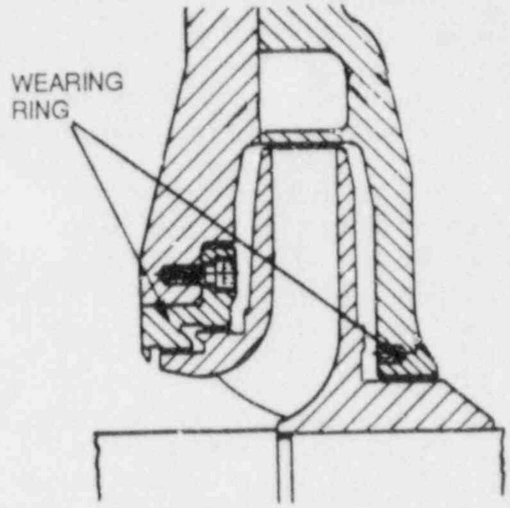

Fig. 2.6. Wear ring and impeller combination.

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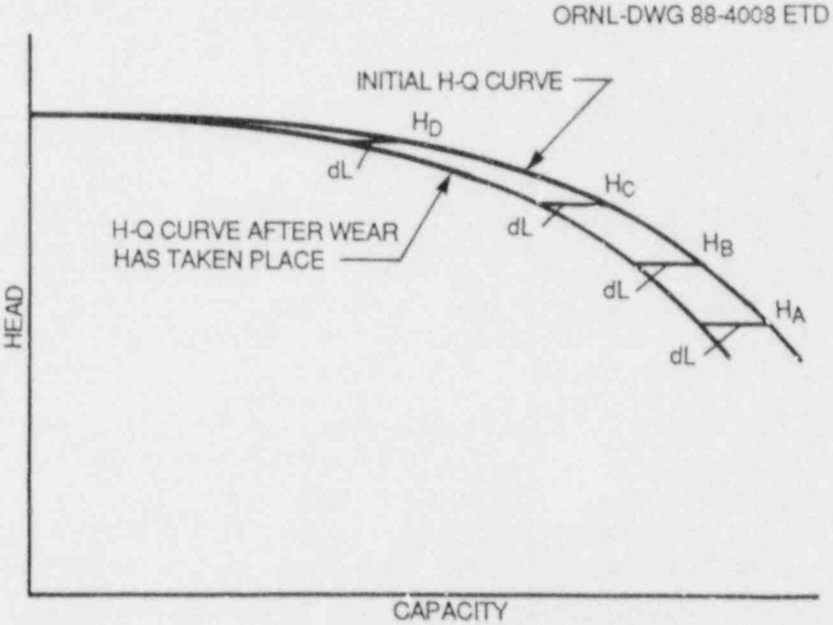

Fig. 2.7. Effect of wear on head capacity curve. *Source*: Adapted with permission from Igor J. Karassick, *Centrifugal Pump Clinic*, Marcel Dekker, Inc., 1981.

internal leakage increase on the pump head-capacity curve. An increase in pump power at a given flow rate may also result because the impeller is pumping not only the normal through flow, but also the added internal recirculation leakage. Another effect caused by internal clearance increases is vibration because these clearances also act as water bearings to provide rotor stiffness, and increasing clearances result in reduced rotor stiffness.

A further cause of worn or broken impellers is a loose impeller bore to shaft fit, which can result in fretting wear of the impeller bore or key, ultimately leading to slipping of the impeller on the shaft. Impeller failures can also be caused by hydraulically induced resonance, which can lead to catastrophic impeller structural failure. Impeller or nonrotating internal wear-ring failures can result from fatigued or broken welds or pins that affix the wear rings to their holders.

Low-flow testing can be a possible source of impeller cavitation damage. The resulting hydraulic recirculation present inside the pump at low flows can create cavitation damage. This problem is further discussed in Appendices D and F.

Thrust balancer wear, galling, and seizing. As discussed in Vol. 1, the thrust balancer is an axial balancing device used to balance thrust generated by hydraulic pressure forces within the pump. The thrust balancer rotates inside a bushing or against a nonrotating internal surface. Small clearances and large pressure differentials exist between these rotating and nonrotating internals. Their surfaces are thus subject to wear and erosion. Other factors that have been found to cause thrust balancer wear, galling, and seizing are identical to those discussed earlier in this report under "Binding between rotor and stationary parts."

Shaft breakage. Shaft failures reported on AUXFPs appear to be more a result of rotating element binding or seizure or other rotating element part failures than inadequate shaft fatigue or torsional design. The fact that AUXFPs do not operate long enough to accumulate significant shaft fatigue cycles, at least now, probably explains this finding. In any event, there were only a few reported shaft failure events. Obviously, shaft failure is a significant failure cause because it can render the pump totally inoperable.

Fastener loosening and breakage (rotating elements, nonrotating elements, mechanical subsystems, support, and pressure-containing casing). Fasteners are used on AUXFPs to locate axially items such as impellers, miscellaneous spacers, and sleeves on the shaft. Fasteners are also used to secure the bearing assemblies to the shaft. Set-screws are often used to lock these fastener nuts to the shaft and to affix various parts of the shaft seal to the shaft sleeves.

Influencing factors, including vibration, thermal expansion and contraction, and rotational torque often cause these fasteners to loosen or break. Improper torquing is a major source of fastener failure, which may occur during shop assembly or following field repair. Overtorquing of fasteners can lead to excessive prestress, which can result in thread stripping or, when combined with vibration, expansion, or contraction can result in fastener failure. Undertorquing can result in fastener loosening, thread stripping, or shear failure from loosely loaded parts. Loose foundation or support bolting can also lead to high pump vibration.

Coupling wear and breakage. Gear couplings are most commonly used on AUXFPs because they tend to be the least expensive type for this service. The most common failure cause experienced with couplings is coupling gear tooth wear. This failure results from improper or inadequate grease lubrication of the gear teeth and/or excessive misalignment of the driver and pump shafts. Other problems include loose hubs resulting from either improper shrink fit or fastener tightness. Catastrophic coupling failures can result from torsional resonance that occurs when the pump and driver rotor assembly torsional natural frequency is equal to the rotational speed of the pump.

Thrust runner wear and breakage. This discussion refers to the thrust runner as the rotating part, fastened to the shaft and captured by a double-acting oil-film thrust bearing. The failure of this component can lead to such problems as rotor seizure and shaft breakage. Thrust runner failures can result from inadequate thrust bearing lubrication, oil contamination, or excessive thrust loads caused by hydraulic unbalance or failure of the pump thrust balancer. However, most AUXFPs employ a double-acting ball bearing or antifriction bearing and do not incorporate a thrust runner.

Leak at casing split. AUXFPs are typically horizontal split-case pumps. The pressure containment casing consists of an upper and a lower casing that are held together by preloaded bolt fasteners. A gasket provides a seal at the casing split line. Gasket aging and degradation can cause leaks. Improper torquing or fastener failure can also result in a leak at the casing split. A small leak, if left unrepaired, can lead to much larger leaks because of casing erosion that occurs along the leak path.

Structural damage to stationary vanes (diffuser or volute). This failure cause often results from hydraulic pressure pulsations that create fatigue loading on internal parts, which are normally castings. Defective castings (i.e., cracked or porous castings) can accelerate structural failure. Fatigue loading cycles can accumulate quickly if the hydraulic pulsations occur at impeller vane-passing frequency. Hydraulic pressure pulsations are of stronger magnitude as the flow increases or decreases from pump best efficiency point (BEP) flow. Although not yet a common failure cause for AUXFPs, this problem is common to high energy per stage pumps, such as large boiler feed pumps. (See Sect. 2.2.2.3.)

2.2.2.2 Design and installation. PWR AUXFP designs evolved from small boiler feed pumps used in older low-pressure fossil electric generating stations. Presently, AUXFPs are classified as the American Nuclear Society (ANS) Safety Class 3 and built to the nuclear design rules of the American Society of Mechanical Engineers (ASME) Sect. III, Subsect. ND (Class 3) requirements. Although the ASME Code was introduced to specifically address the pressure boundary design, its introduction led to the improved quality of all aspects of AUXFP design and manufacturing. Most of the data obtained from operating nuclear plants and used in this report were based on AUXFPs designed and manufactured in the late 1960s to early 1970s. During that time, there was only a small operating experience base to draw from, and equipment suppliers were in strong competition with each other. Consequently, equipment, including AUXFPs, tended to be purchased on the basis of lowest price and not necessarily on the basis of quality or reliability.

Since that time, several design change recommendations have been made by AUXFP suppliers to improve the reliability of these pumps. In the problem area of bearing- and lubrication-related failure causes, one major factor appears to be water contamination of the oil, which ultimately results in bearing failure. Because of the close proximity of the bearing housings to the seal or packing glands, water either spraying or leaking from the glands enters the bearing housing and contaminates the oil, causing bearing failure. No deflector or seal was provided on these designs to divert or prevent water from leaking into the housings. Figures 2.8–2.11 all show recommended modifications that will prevent water from entering the bearing housings.

Many of the bearing-related problems have involved the thrust bearing. AUXFPs typically use a double-row rolling contact or antifriction bearing. As shown in Fig. 2.12, these bearings are usually the preloaded angular contact type that obtain their lubrication by immersion in surrounding reservoir oil. Conversely, double-acting tilt-pad thrust bearings (Kingsbury type) are used on many of the larger boiler feed pumps, and even on the lower-horsepower-rated nuclear centrifugal charging and HHSIPs used in PWR emergency core cooling systems (ECCSs). The tilt-pad bearing shown in Fig. 2.13 is capable of higher thrust loads than the comparably sized antifriction bearing. The tilt-pad bearing appears to be working reliably on the other cited applications; therefore, consideration should be given to the tilt-pad thrust bearing as an upgrade for AUXFPs or when new or replacement pumps are purchased.

Figure 2.14 illustrates another modification recommended by Ingersoll-Rand to improve the reliability of their supplied AUXFPs. Because the shaft must move to permit the balance disc to properly balance the hydraulic thrust, the addition of the spring reduces the thrust load imposed on the rolling contact bearing and allows the shaft to move axially to increase the thrust produced by the balance disk. A seemingly simple design modification to reduce the problem of oil contamination by particulates is to replace the existing standard oil drain plug with a magnetic plug. The magnetic plug attracts the concentration of free metallic particles floating in the lubrication oil.

One other important note of comparison is that the ECCS pumps (i.e., centrifugal charging and HHSI) use a forced-feed lubrication system for both the thrust and radial bearings. This system consists of an oil pump, reservoir, oil cooler, and filter. Unlike the ring oil feed radial journal bearings now used on many AUXFPs, the pressurized system provides a positive flow of oil to the bearings and filters dirt from the oil. Although this system has more auxiliaries, which are also subject to failure, it may be more reliable than the typical AUXFP system, as suggested by the HHSIP failure data.

Shaft seals (i.e., packing and mechanical seals) are the most troublesome pump parts and result in the largest number of recorded problems on all pump applications in the nuclear industry. One issue is the lack of any standard acceptable leakage criteria for seals or packing. It now appears that each utility establishes leakage criteria based on its own experience. Most of the AUXFPs now in operation use packing such as that shown in Fig. 2.1 or single-end-face mechanical seals such as those shown in Fig. 2.5. It is important to recognize that packing requires some finite leakage to ensure cooling and lubrication, whereas

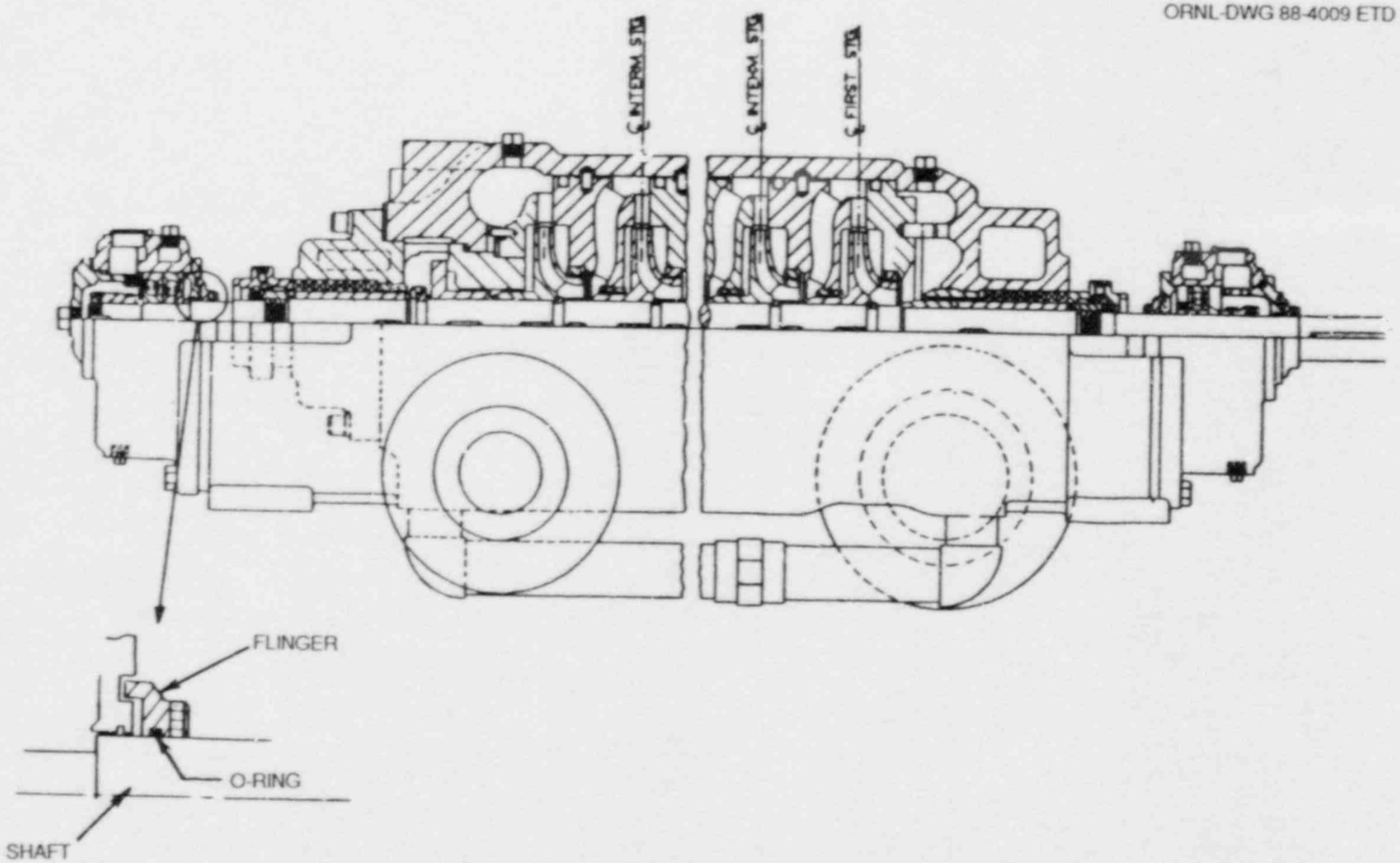

Fig. 2.8. O-ring under flinger. Adapted with permission from F. J. Mollerus, R. D. Allen, and J. D. Gilcrest, *Failures Related to Surveillance Testing of Standby Equipment, Vol. 1: Emergency Pumps*, EPRI NP-4264, Vol. 1, Electric Power Research Institute, October 1985.

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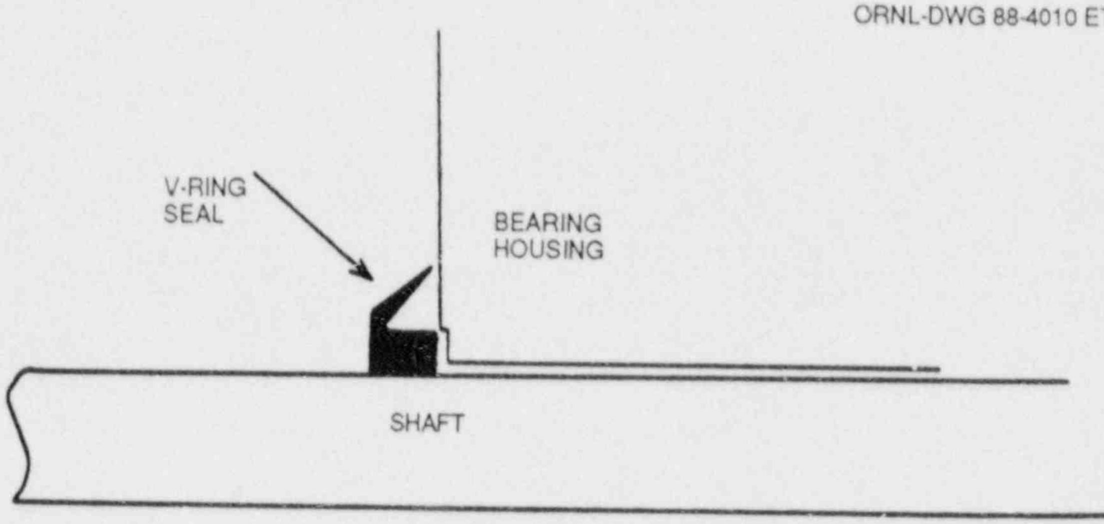

Fig. 2.9. Bearing housing lip seal. Adapted with permission from F. J. Mollerus, R. D. Allen, and J. D. Gilcrest, *Failures Related to Surveillance Testing of Standby Equipment, Vol. 1: Emergency Pumps*, EPRI NP-4264, Vol. 1, Electric Power Research Institute, October 1985.

that required by a comparable mechanical seal is much lower and normally undetectable. Packing is typically a much less expensive sealing technique than the use of mechanical seals. None of the pumps installed on the primary side (e.g., centrifugal charging and residual heat removal) is furnished with packing. Of course, unlike the AUXFPs, these primary-side pumps must seal against potentially higher temperature, pressure, and radioactive environments.

Recently, significant design and application improvements have been made to shaft seals. For example, new packing materials that contain colloidal graphite are now extensively used in nuclear valve applications. They provide much longer life than traditional packing materials, such as cottonite. Shaft sleeves used on packed pumps tend to exhibit more wear than those used on mechanical seal applications. Thus, it is important to select sleeves constructed of high-strength, corrosion-resistant steel (e.g., SA 276 type 410 or another equivalent 400 series) that has been properly heat treated to obtain adequate hardness.

Several utilities have chosen to replace the packing on AUXFPs with mechanical seals. Mechanical seals, if properly selected, installed, and maintained, will provide better absolute sealing than conventional packing. New improvements identified with mechanical seal configurations include better wearing mating materials and environmentally superior O-rings and bellows (secondary seals). Also, the implementation of a disaster or throttle bushing located at the extreme outboard end of the seal provides backup sealing in case of primary seal failure. This bushing is a spring-loaded packing that rides near the shaft and forms a very tight leakage clearance to seal against the catastrophic failure of the primary seal or its subcomponents. Other problems encountered with mechanical seals include the loosening of fasteners, such as the set-screws used to

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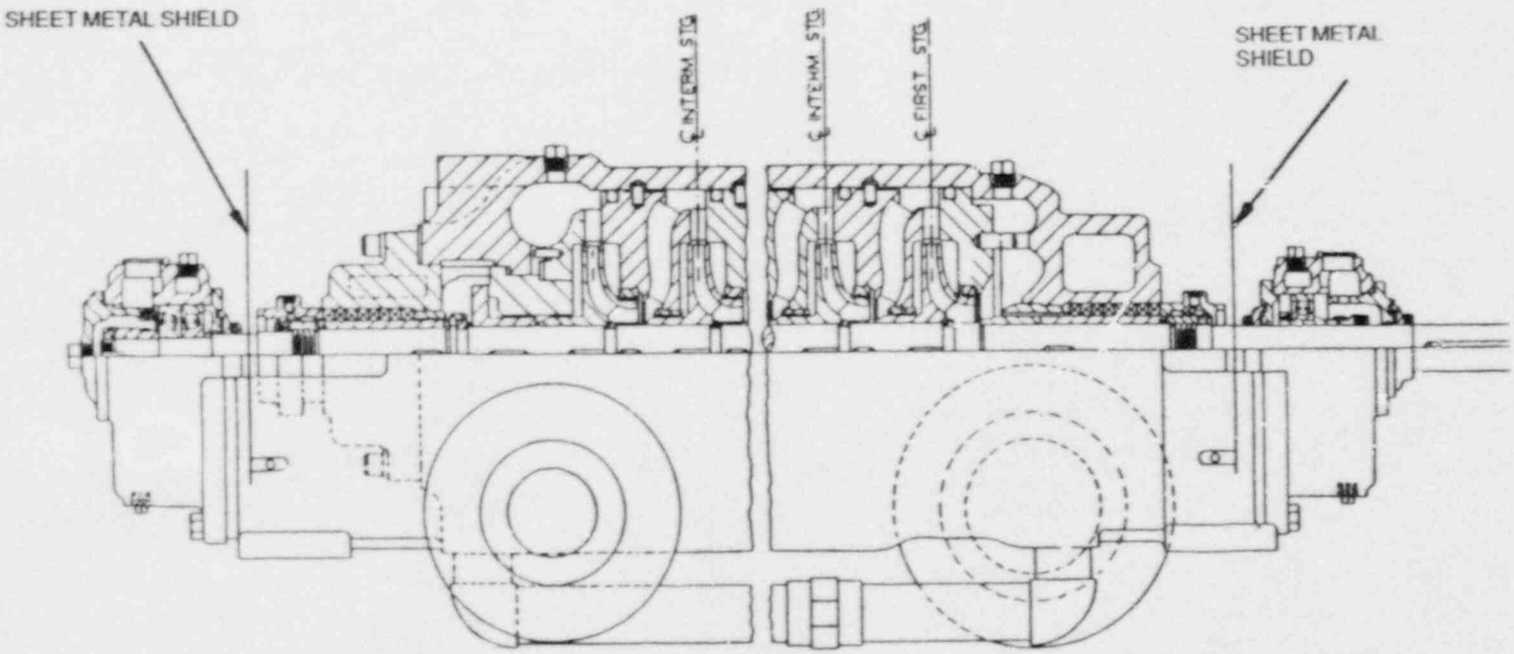

Fig. 2.10. Bearing housing shield. Adapted with permission from F. J. Mollerus, R. D. Allen, and J. D. Gilcrest, *Failures Related to Surveillance Testing of Standby Equipment, Vol. 1: Emergency Pumps*, EPRI NP-4264, Vol. 1, Electric Power Research Institute, October 1985.

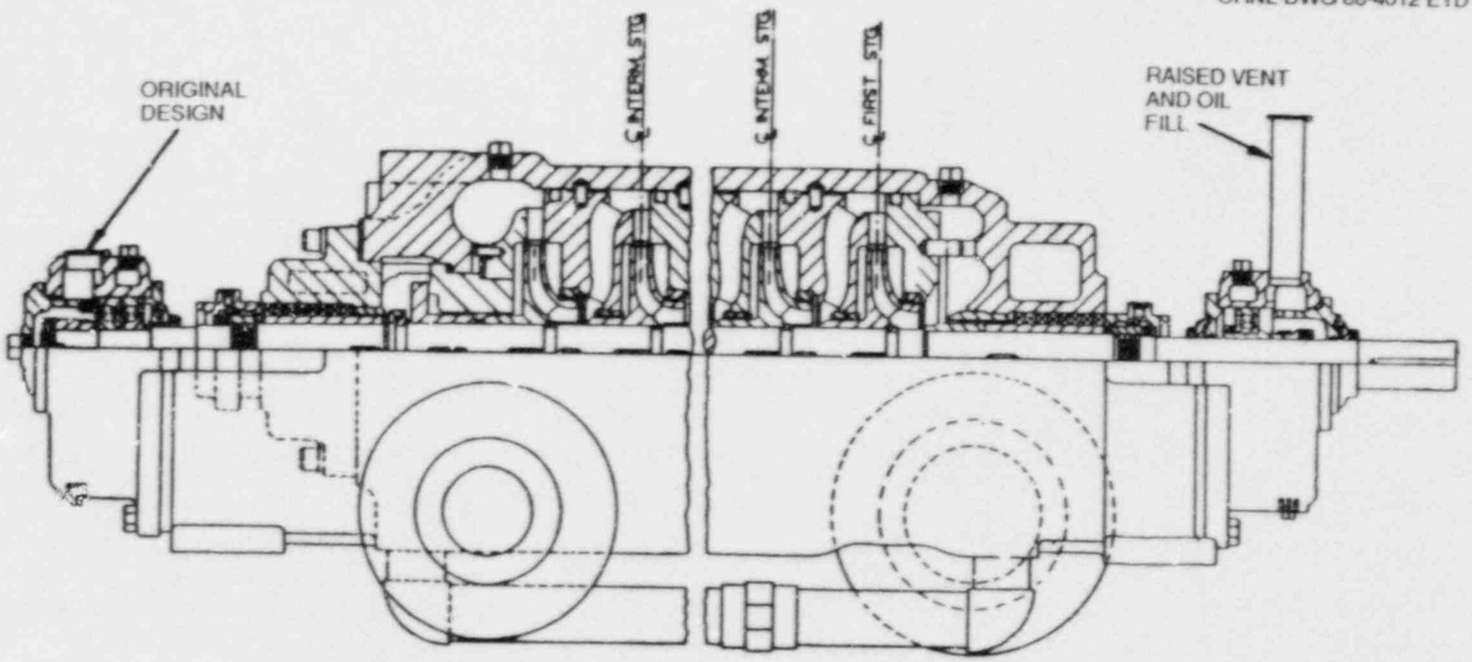

Fig. 2.11. Raised vent and oil fill. Adapted with permission from F. J. Mollerus, R. D. Allen, and J. D. Gilcrest, *Failures Related to Surveillance Testing of Standby Equipment, Vol. 1: Emergency Pumps*, EPRI NP-4264, Vol. 1, Electric Power Research Institute, October 1985.

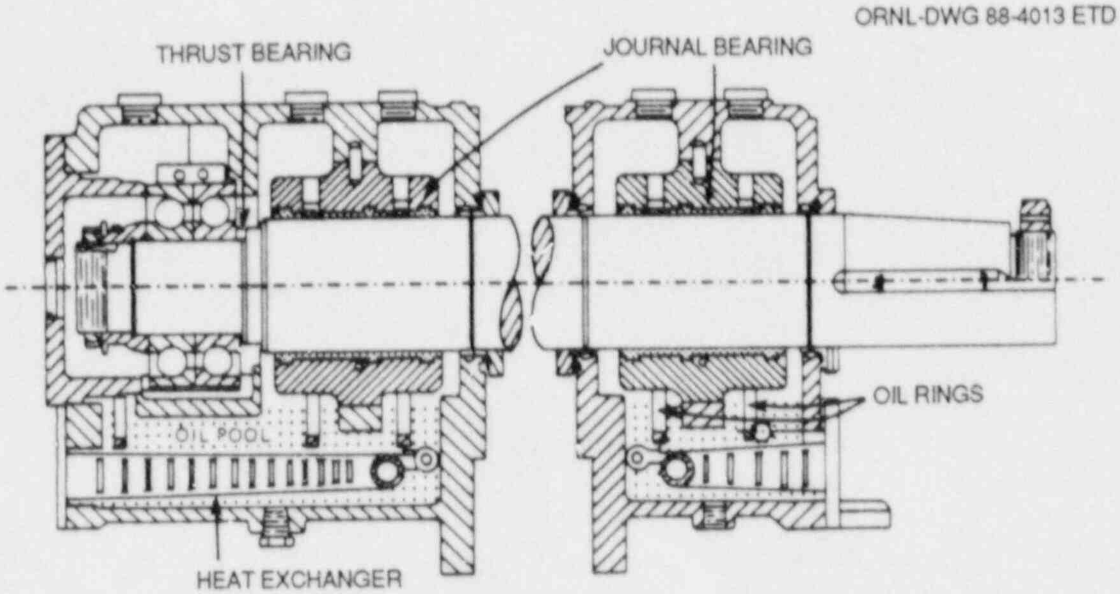

Fig. 2.12. AUXFP bearing system. Adapted with permission from M. L. Adams and E. Makay, *Aging and Service Wear of Auxiliary Feedwater Pumps for PWR Nuclear Power Plants*, Volume 1, Operating Experience and Failure Identification, NUREG/CR-4597, Volume 1 (ORNL-6282/V1), Oak Ridge Natl. Lab., July 1986.

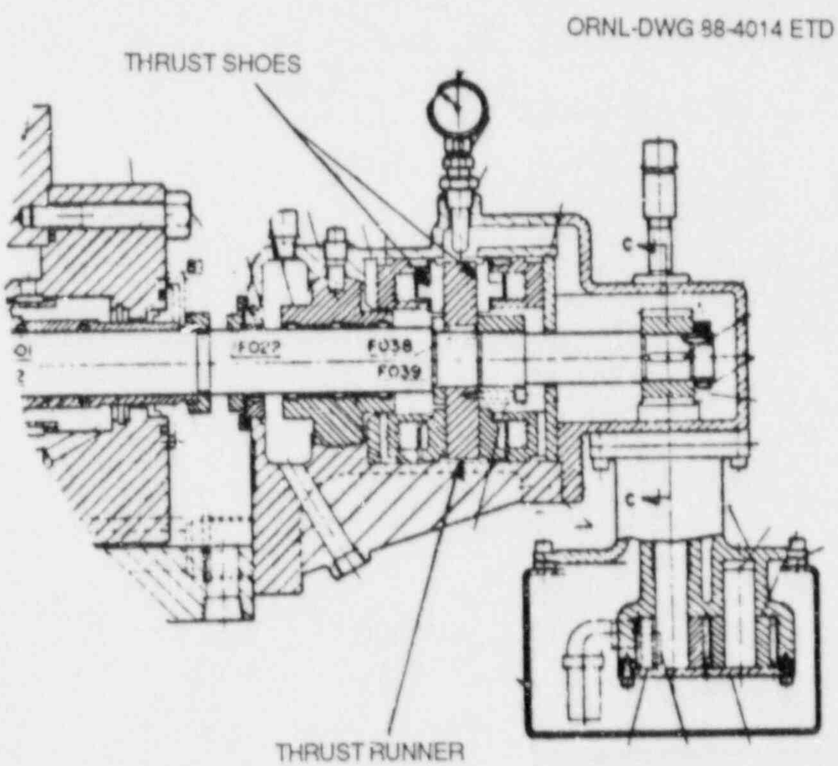

Fig. 2.13. Tilt pad thrust bearing system.

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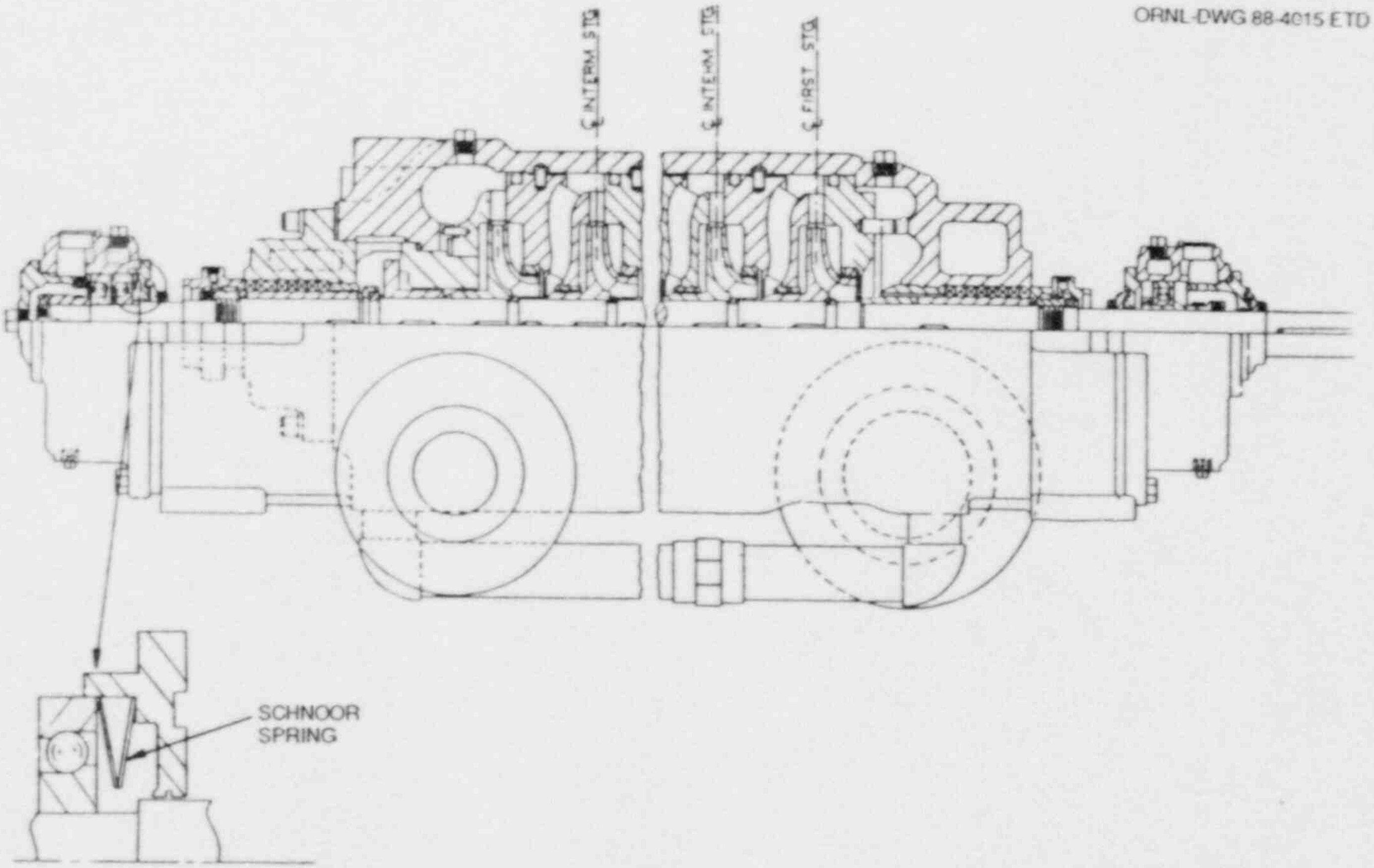

Fig. 2.14. Disk spring design. Adapted with permission from F. J. Mollerus, R. D. Allen, and J. D. Gilcrest, *Failures Related to Surveillance Testing of Standby Equipment, Vol. 1: Emergency Pumps*, EPRI NP-4264, Vol. 1, Electric Power Research Institute, October 1985.

secure the pumping ring to the shaft sleeve. One major nuclear pump supplier recognized these problems and has provided a significant improvement in seal assembly design. Figure 2.5 presents this modified design, which shows a one-piece integral shaft sleeve and pumping ring assembly.

Failure causes associated with impellers, shafts, wear rings, couplings, fasteners, thrust runners, thrust balancers, and stationary vanes (diffuser or volute) are not as common as failures associated with shaft seals and bearings. Binding or seizing between rotor and stationary parts can be a result of improper wear-ring material selection or a bent shaft caused by the shaft fabrication or straightening technique (whereby imposed residual stresses relieve themselves). Binding can also be attributed to high internal vibration resulting from inadequately specified minimum flow. It is important to review the wear-ring materials and select compatible materials that will not be susceptible to galling, yet provide adequate wear resistance.

Although binding can result in reduced pump capacity, this reduction can also result from wear that causes changes in internal clearances, thus increasing internal recirculation leakage. This loss can be mitigated through the proper selection of or upgrade to more compatible wear-ring and balance drum or bushing materials.

There was only one reported instance of cavitation damage on AUXFPs. This situation was discovered during a routine planned outage and repair on an AUXFP. The discovered damage was minor and was not reported to be a failure. The pump supplier thought that it might be caused by low-flow testing, which will be discussed later in this report. As specified earlier, AUXFPs have not accumulated many operating hours to date; consequently, unless there exists a significant or very poor hydraulic design resulting in cavitation, cavitation damage may not be visible by inspection for some time. This report of cavitation damage suggests that periodic disassembly and inspection may be warranted to assess adequately machine condition.

2.2.2.3 Testing. No identified failures can specifically be tied to testing. A recent Electric Power Research Institute (EPRI) report details the results of a two-fold evaluation: first, to identify significant failures or problems of standby emergency pumps, and second, to determine whether those failures were by-products of surveillance testing and, if so, to suggest ways to minimize them. AUXFP license event reports (LERs) from 1979 through late 1982 were reviewed to identity failure frequencies, failure modes, causes, and potential remedial action. Three pump manufacturers also supplemented these records with maintenance records that they had accumulated. The EPRI report concluded that none of the reviewed failures was directly caused by the surveillance testing although most of those failures occurred during testing. However, because of the destructive nature of low-flow testing, degradation associated with the practice of low-flow testing is to be expected.

Appendix D provides a perspective of low-flow (pump miniflow) testing of AUXFPs. Appendix G provides W proposed test guidelines for AUXFPs and discusses the potential for testing at higher flow rates.

2.2.2.4 Maintenance. Timely and correct maintenance is probably the single most important factor influencing premature AUXFP component failure. Table 2.6 ranks the importance and influence of maintenance

Table 2.6. The importance of maintenance in mitigating pump failure causes attributed to aging and service wear

Pump segment	Parts	Failure cause	Ranking
Rotating elements		Binding between rotor and stationary parts	1
	Shaft	Shaft breakage	7
	Impeller	Impeller wear, breakage	5
	Thrust runner	Thrust runner wear, breakage	12
	Fasteners	Fastener loosening, breakage	4
Nonrotating internals	Diffusers or volutes	Structural damage to stationary vanes (diffuser or volute)	
	Wear surfaces	Wear-surface wear, erosion, corrosion, seizing	8
	Fasteners	Fastener loosening, breakage	9
Pressure containment casing	Casing	Leak at casing split	13
		Leak at casing rupture disk	
	Suction nozzle	Suction nozzle leak, breakage	
	Discharge nozzle	Discharge nozzle leak, breakage	
	Fasteners	Fastener loosening, breakage	14
Mechanical subsystems	Bearings	Bearing wear, corrosion, breakage	2
	Shaft seals	Shaft seal deterioration, breakage	3
	Thrust balancer	Thrust balancer wear, galling, seizing	10
	Coupling	Coupling wear, breakage	11
	Fasteners	Fastener loosening, breakage	6
Support	Base frame	Base frame breakage	
	Fasteners	Fastener loosening, breakage	15

practices in mitigating the identified failure causes attributed to wear and aging. Again, the lowest number corresponds to the highest ranking.

The components that formed a part of the rotating element or mechanical subsystems were most influenced by maintenance. Failures attributed to aging and service wear are often actually caused by poor maintenance. Proper maintenance training cannot be overemphasized. Maintenance and operations personnel must receive regular training in the operation of the equipment, including its installation and repair. Technical bulletins, maintenance bulletins, vendor training programs, vendor changes, and current drawings and parts lists should be included in the overall training program. Components that are not properly repaired cannot be expected to fulfill their expected service life. For example, the consequences of excessive gland nut tightening cannot be overemphasized. The resulting inadequate leakage and lubrication not only burns the packing but damages the shaft sleeve and possibly the gland. Proper packing installation is another very important point in obtaining controlled leakage with a set of packing.

Again, it is recommended that die-formed packings be purchased to minimize the errors that may result during packing, cutting, forming, and installation. If mechanical seals are used, the cartridge seal, although more expensive, leads to fewer installation and maintenance problems.

Improper lube oil level was the single most important factor reported under maintenance involving bearings. Other factors contributing to bearing failures included water or dirt contamination of the oil, blocked oil drain return, and several instances of a leaking oil pipe or fitting.

Rotating element and mechanical subsystem problems included improper balance drum to bushings clearance and improperly installed wear rings.

2.2.2.5 Other. This category includes all other causes not previously discussed that specifically affect the pump. Failure causes attributed to the steam supply, turbine, or motor were excluded from this report.

The most commonly reported problems in this category include instrumentation and control logic circuitry malfunction, deficient procedures and documentation, and human performance errors. A recent Institute of Nuclear Power Operations (INPO) report stated that two nuclear units contributed over 50% of the pressure switch and pressure transmitter failures, the primary cause being human error, such as incorrectly calculated pressure switch set-points. Most actuation and control circuitry failures were also caused by human error, particularly the mispositioning of manually operated switches and the use of deficient procedures to verify operability following work on the circuitry. Other important human performance errors included improper valve alignment that caused a loss of water to the AUXFPs and actions that resulted in a loss of the automatic start-up signal to the pumps. Deficient procedures and documentation suggests that some of these events could be precluded by a systematic review and improvement of procedures for technical content and format.

Several nuclear plants are equipped with low-pressure switches that trip the AUXFPs on low water suction pressure. This instrumentation caused inadvertent equipment shutdowns, especially during plant start-ups when pressure surges caused temporary low pressure. A time delay was added to this switch to alleviate this problem. Another problem that can affect pump performance is steam binding caused by leakage from the main feedwater system. This problem was documented in NRC IE Bulletin No. 85-01, "Steam Binding of Auxiliary Feedwater Pumps."

The pump, driver, and auxiliary systems assembly is typically mounted on a common fabricated steel baseplate that is rigidly anchored to the building floor, normally through foundation bolting. The base is then filled with grout, which stiffens and strengthens the base installation. The suction nozzle is connected to a hydraulic circuit that originates at the condensate storage tank. The discharge nozzle connects to a piping circuit that ties into the main feedwater header. Pump operational problems can result from improper installation in the plant. Many such problems have been found in nuclear and fossil-generating plant and chemical plant applications. The structural baseplates must be properly leveled prior to final grouting. Improper leveling could lead to pump and driver vibration caused by the fact that pump and driver alignment cannot be initially achieved or maintained. Poor grout or improperly installed grout can also leave the base flexible, which could result in structural vibration or even resonance, possibly affecting the pump or driver. Excessive piping and end loads applied to the pump nozzles are probably the most prevalent sources of plant installation problems with pumps. If the piping is improperly aligned to the pump nozzles or the piping has too few supports, excessive loads can result, which could cause equipment misalignment and high stress at the pump supports and/or the nozzles. This condition is aggravated in a high-temperature service, such as boiler feed, in which piping thermal expansion adds additional nozzle loads. Although AUXFPs may be subjected to initial piping misalignment, the thermal changes through the pump are essentially nil, thus eliminating any local thermal effects.

3. CANDIDATE ISCM METHODS

3.1 Potentially Useful ISCM Methods

The information on pump failure causes and rankings in Table 2.2 was used to identify causes to be considered in an AUXFP ISCM program; the causes and associated measurable parameters to be considered are tabulated in Table 3.1. Table 3.2 summarizes the potentially useful ISCM methods for obtaining each measurable parameter. Discussions of ISCM methods and their relation to aging and service wear are contained in Sect. 3.2. A summary of correlations between measurable parameters and degradation is given in Sect. 3.3.

3.2 Potentially Useful ISCM Methods and Their Relation to Aging and Service Wear

3.2.1 Rotor binding inspection

This inspection is to be performed before pump start-up. Power to the pump driver should be disconnected to prevent injury caused by an inadvertent pump start. A suitable "spanner wrench" or other tool should be applied to the coupling and the pump rotor turned. The inspector should feel resistance or listen for rubbing that may result from insufficient clearance between the rotating and stationary parts, bearing failure, or shaft seal failure. If excessive rotating resistance results, then the cause of the failure should be investigated and corrected.

3.2.2 Visual inspection

This inspection pertains to the appearance of the parts. This detection method is composed of two types: regular inspection, which is the regular examination of the pump that can be performed without disassembly, and periodic inspection, surveillance, and maintenance, which requires disassembly of the pump.

Regular visual inspection should include observations for lube oil system leaks, shaft seal or casing leakage rates, coupling lubrication leaks, external fastener looseness, and lube oil level or quantity.

Periodic visual inspection requires some disassembly of the pump segments. Inspections should include examination for such things as binding, wear, mechanical looseness, erosion, corrosion, broken parts, and overheating. For example, binding between rotating and stationary parts will show galling indications and heat discoloration (depending on the severity of rubbing) to the wearing surfaces.

Normal wear of parts will also show metal removal and may result in mechanical looseness or change in clearance. Impellers should be examined for signs of cavitation at both the leading and trailing vane edges. Cavitation will result in erosion of the impeller surface. Impeller bores and key fits to the shaft should be inspected for signs of looseness. Bearing fits should be examined to ensure that the bearings are

Table 3.1. Failure causes and measurable parameters to be considered in an AUXFP ISCM Program

Pump segment	Parts	Failure cause	Measurable parameters
Rotating elements		Binding between rotor and stationary parts	Rotational torque, appearance, motor current, motor power, turbine power, speed, wear surface and critical fit clearance
	Shaft	Shaft breakage	Rotational torque, appearance, developed head, delivered flow, motor current, motor power, turbine power, speed, vibration, surface indications, dynamic pressure
	Impeller	Impeller wear, breakage	Rotational torque, appearance, developed head, delivered flow, motor current, motor power, turbine power, speed, vibration, balance return line flow, noise, wear surface and critical fit clearance, surface indications, dynamic pressure
	Thrust runner	Thrust runner wear, breakage	Rotational torque, appearance, motor current, motor power, turbine power, speed, vibration, temperature, axial position, surface indications, noise
	Fasteners	Fastener loosening, breakage	Rotational torque, bolt torque, motor current, motor power, turbine power, vibration speed, axial position, dynamic pressure
Nonrotating internals	Diffusers or volutes	Structural damage to stationary vanes (diffuser or volute)	Appearance, developed head, delivered flow, vibration, noise, dynamic pressure
	Wear surfaces	Wear-surface wear, erosion, corrosion seizing	Rotational torque, appearance, developed head, delivered flow, vibration, axial position, balance return line flow, wear surface and critical fit clearance, noise, dynamic pressure, motor power, turbine power
	Fasteners	Fastener loosening, breakage	Bolt torque measurement appearance, vibration, noise
Pressure containment casing	Casing	Leak at casing split	Appearance, leakage rate
Mechanical subsystems	Bearings	Bearing wear, corrosion, breakage	Rotational torque, appearance, motor current, motor power, turbine power, speed, vibration, temperature, lube oil purity, axial position, wear surface and critical fit clearance, noise, dynamic pressure, oil quantity

Table 3.1 (continued)

Pump segment	Parts	Failure cause	Measurable parameters
Mechanical subsystems (continued)	Shaft seals	Shaft seal deterioration, breakage	Rotational torque, appearance, temperature, vibration, leakage rate, wear surface and critical fit clearance, noise, dynamic pressure
	Thrust balancer	Thrust balancer wear, galling, seizing	Rotational torque, appearance, wear surface and critical fit clearance, motor current, motor power, turbine power, developed head, delivered flow, vibration, axial position, balance return line flow, noise
	Coupling	Coupling wear, breakage	Rotational torque, appearance, motor current, motor power, turbine power, speed, vibration, noise
	Fasteners	Fastener loosening, breakage	Bolt torque, appearance, vibration, noise
Support	Fasteners	Fastener loosening, breakage	Bolt torque, appearance, vibration, noise
	Base frame	Base frame breakage	Appearance, vibration, noise

Table 3.2. Measurable parameters and their relative ISCM methods

Measurable parameters	ISCM methods[a]
Rotational torque	*Rotor binding inspection
Appearance	*Visual inspection
Motor current	Motor current monitoring
Motor power	*Motor power monitoring
Turbine power	Turbine power monitoring
Speed	*Rotational speed monitoring
Wear surface and critical fit clearance	*Dimensional inspection
Developed head	*Pump pressure or developed head monitoring
Delivered flow	*Pump delivered flow monitoring
Vibration	*Rotor vibration monitoring *Incipient failure detection monitoring
Surface indications	*Liquid penetrant inspection
Balance return line flow	*Balance return line flow monitoring
Noise	*Audible noise inspection
Temperature	*Bearing temperature monitoring Stuffing box temperature monitoring
Axial position	*Rotor axial position monitoring
Bolt torque	*Bolt torque inspection
Leakage rate	*Leakage rate inspection
Lube oil purity, quantity	*Lube oil inspection
Dynamic pressure	Discharge dynamic pressure monitoring

[a]Items with asterisks are discussed in detail in Chap. 4.

not excessively worn or broken. Journal bearing babbitt may show wear or discoloration caused by dirty lube oil.

The shaft seals should be examined for wear or abnormal wear that may result in indications of overheating of parts such as packing, shaft sleeves, packing gland, stuffing box, mechanical seal washers, and seal seats.

Highly stressed parts, such as the shaft, antifriction bearings, and fasteners, should be examined for cracks or breaks.

3.2.3 Motor current monitoring

This measurement collects motor dynamic current signatures. By examining the electric current spectra analysis, one can determine whether there are factors actively producing change in the current signature. These include mechanical rubs within the pump, loose components, or worn parts. Instruments used to collect this data consist of clamp-on current probes that may be installed at the motor control center.

3.2.4 Motor power monitoring

This measurement indicates pump power and is applicable to motor-driven pumps. Rotor binding will result in an increase in motor inrush current to the point where complete pump rotor binding will result in the motor current being equal to the "locked rotor" current value. Normally, the motor circuit breaker will open shortly after this condition is reached; otherwise, motor overheating and damage will result. In some cases, an increase in pump internal clearances caused by aging can result in increased pump power consumption.

3.2.5 Turbine power monitoring

This measurement also indicates pump power and is applicable to turbine-driven pumps. Rotor binding will result in an increase in steam flow because the turbine cannot accelerate and the flow control valve will open to provide more steam. This situation is synonymous to motor locked rotor. The steam flow, pressure drop, and turbine speed are required to perform this measurement. In some cases, an increase in pump internal clearances caused by aging can result in increased pump power consumption.

3.2.6 Rotational speed monitoring

The speed measurement must also be used in verifying the cause of reduced pump delivered flow or developed head because these parameters are proportional to pump speed. This measurement can also indicate a problem with the power source.

3.2.7 Dimensional inspection

This inspection pertains to the measurement of critical pump part fits and wearing surfaces. These measurements include those on the impeller bore, shaft, bearings, impeller wearing rings, wear surfaces, thrust runner, thrust balancer, and shaft seal wearing parts. This inspection should include a comparison of the new part's dimensional requirements with the "as-inspected" part condition. The condition of aging and wear can often be correlated to the resulting change in, for example, fits and clearances. In all cases, these as-inspected dimensions should be compared with the pump vendor's specified dimensions, and the parts should be replaced as required (e.g., a rule of thumb stated by several pump vendors is to replace wear rings when clearances have doubled). Section 2.2.2.1 further discusses the effect on performance of changes in some of these critical dimensions and their relation to aging and service wear.

3.2.8 Pump pressure or developed head monitoring

The purpose of pump operation is to develop pressure or head and to deliver flow. As the pump ages, the ability to do both diminishes. This condition is caused primarily by the loss of critical wear clearances on the impeller, wear surfaces, and thrust balancer. The resulting increase in clearances reduces the pump's hydraulic efficiency because of the increased internal recirculation flow through these "opened" clearances. Section 2.2.2.1 and Fig. 2.7 more thoroughly address the resulting effect on hydraulic performance caused by these "opened" clearances.

3.2.9 Pump-delivered flow monitoring

As explained in Sect. 3.2.8, as a pump ages, its ability to deliver flow also diminishes because of the loss in hydraulic efficiency caused by the increased internal recirculation losses.

3.2.10 Rotor vibration monitoring

Rotor vibration monitoring is one of the most extensively practiced methods for detecting machinery condition. Excessive or increasing rotor vibration can be an indication of rotor component wear, unbalance, wear-surface wear, impeller wear, loose fasteners or fits, bearing deterioration, coupling wear, driver misalignment, and almost anything associated with the pump's rotating parts. The consequences of excessive vibration are an increased potential for fatigue failures for both rotating and stationary parts, reduced bearing life, accelerated shaft seal aging, shaft failure, rubbing at close clearance fits, and, ultimately, the loss of those close clearances, with a respective deterioration in hydraulic efficiency. Probably the single most important mechanism for wear and aging in a pump is excessive rotor vibration because this vibration can affect all of the pump parts.

Rotor vibration can be trended and the various harmonics examined. A gradual change in vibration indicates component wearing or aging, such as an increase in bearing or wearing ring clearances, rotor unbalance, parts looseness, and coupling misalignment. A sudden change in vibration can indicate catastrophic failure, such as a broken shaft, bearing or shaft seal failure, and coupling failure.

An examination and trend of the various multiples of the running speed component (e.g., 1X, 2X, 3X . . . blade passing frequency) can be used to predict the failure mechanisms progressing inside the pump.

3.2.11 Incipient failure detection monitoring

This monitoring method is especially useful in obtaining high-frequency noise or vibration measurements. (It is also referred to as acoustic, high-frequency, or stress-wave emission measurement.) Incipient failure detection (IFD) instrumentation is often used on rolling contact (antifriction) bearings. This bearing type is commonly used for both the radial and thrust bearings on AUXFPs. IFD is a relatively new monitoring method that can be successfully applied to measure and predict rolling contact bearing, coupling, gear wear, shaft seal deterioration, rotor rubbing, and other problems.

3.2.12 Liquid penetrant inspection

This inspection involves the application of a penetrant dye to part surfaces to determine the presence of surface indications (or cracks). It is to be used only on metal parts. The part is first carefully cleaned, and then the dye is applied. Once the dye dries, a developer is sprayed over the dye. The developer causes the dye to "bleed" from the locations that contain cracks. This bleeding is often visibly obvious; however, as with all nondestructive examination methods, the user must be experienced in the application of the method and reasonably experienced in the interpretation of results.

3.2.13 Balance return-line flow monitoring

This measurement pertains to the pump designs that include a thrust balance flow return line. This line connects the downstream chamber of the balance drum or disk to the pump suction chamber. Flow through the balance drum or disk clearances is returned to the pump suction. An increase in the flow rate signifies an increase in thrust balancing device clearance between the rotating device and its mating wearing surface. Increased thrust balancing device clearances are often indicative of similar wear at the impeller and wear surfaces.

3.2.14 Audible noise inspection

This inspection requires the pump to be operating. A properly running pump will produce a distinct audible noise signature, whereas an abnormally running machine will result in a change to the audible sound. Problems or failure causes associated with cavitation, rubbing, high shaft vibration, bearing wear, coupling wear, and so forth, can often be diagnosed through this method.

3.2.15 Bearing temperature monitoring

This method is primarily applicable to sleeve bearings, such as radial journal or tilting pad thrust bearings, although it has also been used to indicate the condition of rolling element (antifriction) bearings. An increase in bearing temperature normally indicates bearing wear or eventual failure. Increased bearing temperature can also be caused by inadequate lube oil flow or level, impure oil, general bearing deterioration, or overloading, possibly caused by the deterioration of other parts of the pump (e.g. the thrust balancer).

3.2.16 Stuffing box temperature monitoring

This method is used to detect shaft seal packing deterioration caused by packing wear, shaft sleeve wear, inadequate packing clearance, excessive packing gland tightness, or inadequate mechanical seal cooling. Shaft seal deterioration and leakage often result from worn packing and mating parts.

As these parts wear, rotating friction increases, and stuffing box temperature increases. Inadequate packing leakage causes inadequate packing cooling and lubrication, which in turn can cause severe wearing of the packing, shaft sleeve, and even stuffing box inside diameter. All of these conditions cause stuffing box temperatures to increase.

3.2.17 Rotor axial position monitoring

This measurement indicates rotor axial position relative to the stationary parts of the pump. Thrust balancer (drum and disk), thrust runner, or bearing wear will cause a change in the normal axial pump rotor position.

3.2.18 Bolt torque inspection

This inspection consists of measuring the critical fastener torque. A suitable calibration torque measuring device, such as a torque wrench, should be used to check bolt torque against recommended vendor values. Fastener wear or clamped part wear often causes a relaxation to specified torque values. Undertorquing can lead to fastener wear and ultimate fatigue failure, whereas overtorquing can lead to excessive preload to fastener and clamped part, causing overload failures.

3.2.19 Leakage rate inspection

This measurement consists of a visual inspection of shaft seal leakage and casing split line leakage. Shaft seal packing must allow a finite leakage to ensure proper cooling and lubrication of the rotating sleeve and mating stationary packing. As the packing wears, it will normally exhibit increased leakage. Mechanical shaft seals should not allow noticeable leakage. However, as the seal wears, noticeable leakage can also be expected.

No leakage at the casing split line should be observed. Leakage there may result from inadequate pressure boundary casing fastener torque, a deteriorated casing split line gasket, or erosion of the pump casing split line surface.

3.2.20 Lube oil analysis and quantity inspection

This inspection consists of periodic sample analyses of the lube oil to inspect for contamination caused by dirt, wear, or abnormal bearing and babbitt filings. An increase in the concentration of metallic contaminants indicates an increase in bearing wear. The presence of free water in the lube oil can cause roller bearing corrosion and ultimately failure. Improper oil viscosity can also cause bearing wear and failure because the viscosity index directly affects lubrication capability. Lube oil quantity is another important variable that must be checked.

3.2.21 Dynamic pressure monitoring

A measurement of the pump discharge pressure using a dynamic pressure transducer may show hydraulic instabilities occurring within the pump caused by various problems, such as low-flow internal recirculation, cavitation, and impeller stall. This measurement is commonly used in shop or laboratory tests of high energy per stage pumps, such as large boiler feed pumps, steam generator feedwater pumps, and condenser circulating water pumps to determine the point of low-flow instability. A centrifugal pump should produce a smooth, pulsation-free flow signature. Flow-induced instabilities will cause a change in the flow signature, generally manifested as low frequency pulsations (<10 Hz).

Cavitation or impeller stall often causes a strong vibration that occurs at a frequency equal to blade passing frequency (i.e., rotating speed multiplied by number of impeller vanes). Not only can this frequency be detected with a pressure transducer, but it can also be detected with vibration monitoring equipment, such as a vibration transducer mounted to monitor shaft vibration.

3.3 Summary of Measurable Parameter vs Degradation Correlations

This summary, which is based on the previous discussions, is given in Table 3.3. The parameters are to be used for both detecting and tracking degradation caused by aging and service wear.

Table 3.3. Parameter vs degradation correlation

Parameter	Degradation
Rotational torque	Rotor-stationary part binding
Motor current Motor power Turbine power	Rotor rubbing or binding and loss of efficiency due to wear
Rotational speed	Overall machine health[a]
Wear surface and critical fit clearance	Rotating and stationary part wear and galling
Pressure or developed head	Overall machine health
Delivered flow	Overall machine health
Vibration	Wear of wear surfaces, impeller, bearings, gears, and coupling; loose fasteners; shaft seal deterioration; and rotor rubbing
Surface indications	Cracks, surface irregularities, etc., related to structural integrity and machine operation
Balance return line flow	Internal clearance increase
Audible noise	Rotor-stationary part rubbing or binding and rotating part wear
Temperature	Bearing wear Stuffing box packing deterioration
Rotor axial position	Thrust balancer, thrust runner, or bearing wear
Bolt torque	Fastener loosening
Leakage rate	Shaft seal and casing split seal deterioration
Lube oil purity	Bearing wear
Discharge dynamic pressure	Passage clearance changes due to wear

[a]Required for reference in determining pump head and flow.

4. EVALUATION OF ISCM METHODS

4.1 Recommended ISCM Methods

This section identifies recommended ISCM methods selected from those discussed in Chap. 3 and discusses the recommended methods. Major factors influencing the selections were based on their ability to (1) establish operational readiness, (2) detect aging and service wear degradation, and (3) determine aging and service wear degradation trends. The selections were also guided by W field experience with pumps similar to AUXFPs, reactor coolant pumps, heat exchangers, steam turbines, steam generators, reactor internals, and piping systems, as well as laboratory tests performed on pumps. An evaluation of the selected ISCM methods is summarized in Table 4.1.

The ISCM methods selected are labeled with an asterisk in Table 3.2. The ISCM methods not selected are motor current, turbine power, stuffing box temperature, and dynamic pressure monitoring. The reasons are as follows.

Motor current monitoring. Work has been done on the signature analysis of motors in valve operator and other applications, and the results show broad potential for diagnostic use. However, experience with signature analysis of this type is limited, and the data base required is not available for near-term use.

Turbine power monitoring. Turbine power monitoring requires simultaneous knowledge of steam flow, steam pressure drop, and turbine speed. It is unlikely that the accuracy and repeatability of these parameters, when taken together, would provide a reasonable indication. Considering the expense of such measurements and the lack of confidence in the results, turbine power monitoring based on the three variables mentioned was not selected. However, other means for determining turbine power should be investigated because this is an important parameter to correlate with pump flow and head.

Stuffing box temperature monitoring. An experience base for using this method has not been identified. Consequently, there is not enough information to select this as a viable method at this time.

Dynamic pressure monitoring. Pump hydraulic passage clearance changes resulting from aging and service wear will often produce changes in discharge pressure signatures. But it is unlikely that AUXFPs will produce sufficient change in this signature to warrant monitoring of this parameter because of their low to moderate energy per stage.

4.2 Further Discussion of Recommended ISCM Methods

In the following subsections, additional elaboration on the use and capabilities of the selected methods is given to provide added insight into details of their application and usage and the results to be obtained.

Table 4.1. ISCM evaluation summary

Method	Measured parameter	Effectiveness (in determining aging and wear)	Importance in terms of safety	Cost[a] of implementation	Remarks[b,c]
Rotor binding inspection	Rotational torque	Determines pump rotating element's ability to function; indicates operational readiness	High	Low	Break-away torque measurement should be recorded/trended and compared to baseline value
Visual inspection	Appearance	Very elementary but highly effective means to determine pump operational readiness	High	Low	Most elementary form of inspection that can be performed; should be performed during regular (routine) maintenance and periodic inspection, surveillance, and maintenance
Motor power monitoring	Power	Indicates hydraulic and mechanical degradation	Medium	Implementation cost — low; instrumentation cost — low	Should be trended Instruments Watt transducer: Accuracy — medium Repeatability — high
Rotational speed	Pump speed	Necessary to correlate delivered flow and head	Low	Implementation cost — medium; instrumentation cost — low	Should be trended Instruments Proximity probe Infrared stroboscope Accuracy — medium Repeatability — high
Dimensional inspection	Wear surface and critical fit clearance	Indicates mechanical wear	Medium to high	Low to medium	Should be performed during regular (routine) maintenance and periodic inspection, surveillance, and maintenance; should be trended
Developed head monitoring	Pump head (pressure)	Indicates hydraulic degradation; low flow — may indicate only gross degradation due to curve "flatness"; wear and aging indication improves with increasing flow; indicates operational readiness	High	Implementation cost — high (if instrumentation is installed in pressure boundary); instrumentation cost — low	ASME XI test acceptance standard; tech. spec. standard; should be trended Instruments Gages Transducers Transmitters Accuracy — Medium Repeatability — high

Table 4.1 (continued)

Method	Measured parameter	Effectiveness (in determining aging and wear)	Importance in terms of safety	Cost[a] of implementation	Remarks[b,c]
Delivered flow monitoring	Delivered flow	Indicates hydraulic degradation; low flow — poor accuracy and repeatability; indicates operational readiness	High	Implementation cost — high (if installed in pressure boundary); instrumentation cost — high	ASME XI test acceptance standard; should be trended Instruments Orifice or Venturi: Accuracy — medium Repeatability — medium Externally mounted instruments (e.g., ultrasonic meter): Accuracy — medium Repeatability — low
Vibration monitoring	Vibration (displacement, velocity, acceleration)	Indicates mechanical degradation and mechanical operational readiness	High	Implementation cost — medium to high; instrumentation cost — medium	ASME XI test acceptance standard; should be trended Instruments Proximity probes: Accuracy — medium Repeatability — high Velocity or accelerometers: Accuracy — low Repeatability — medium
Balance return-line monitoring	Balance return-line flow	Indicates hydraulic degradation by virtue of increased internal leakage measurement	Low	Implementation cost — high (if installed in pressure boundary); instrumentation cost — high	Should be trended Instruments Orifice or Venturi: Accuracy — medium Repeatability — medium Externally mounted instruments (e.g., ultrasonic meter): Accuracy — medium Repeatability — low
Audible noise inspection	Noise	Change in audible level may indicate impending pump problem; requires experienced individual to perform the assessment	Medium to high	Low	Should be performed during periodic surveillance testing
Incipient failure detection (IFD)	High-frequency vibration	Indicates mechanical degradation; especially effective in determining impending antifriction bearing and shaft seal problems; indicates operational readiness	Medium to high	Implementation cost — medium; instrumentation cost — medium	Should be trended; relatively new technique Instruments Piezoelectric sensor: Accuracy — low Repeatability — low

Table 4.1 (continued)

Method	Measured parameter	Effectiveness (in determining aging and wear)	Importance in terms of safety	Cost[a] of implementation	Remarks[b,c]
Bearing temperature monitoring	Temperature	Indicates bearing condition; can also provide an indication of lube oil purity and quantity; indicates operational readiness	High	Implementation cost — high, if installed in or in proximity to bearing; instrumentation cost — low	ASME XI test acceptance standard; should be trended Instruments Thermocouples or resistance thermal detectors (RTD): Accuracy — high Repeatability — high
Rotor axial position monitoring	Rotor axial position	Indicates mechanical degradation of the thrust bearing, balance drum, or balance disk; indicates operational readiness	Medium to high	Implementation cost — medium to high; instrumentation cost — medium	Should be trended Instruments Proximitor: Accuracy — medium Repeatability — high
Bolt torque inspection	Torque	Indicates fastener tightness, worn condition of fastener or assembled parts	Medium to high	Low	Should be trended; should be performed during routine inspection (where practical) and during periodic inspection, surveillance, and maintenance
Leakage rate inspection	Leakage rate (shaft seal, case, nozzles)	Most practical indication of shaft seal condition	Medium to high	Low	Should be trended; should be performed during routine inspection
Lube oil analysis inspection	Lube oil purity	Indicates condition of lube oil; can prevent impending bearing wear, corrosion, or seizure	High	Low — if performed "in-house"	Should be trended
Liquid penetrant inspection	Surface indications	Indicates cracking caused by overstress, corrosion, erosion	High	Low	Should be performed during periodic inspection, surveillance, and maintenance

[a]Cost criteria:
- High $10—$50K
- Medium $5—$10K
- Low $0—$5K

[b]Accuracy criteria:
- High 0.5—1%
- Medium 1—10%
- Low >10%

[c]Repeatability criteria:
- High 1—5%
- Medium 5—10%
- Low >10%

4.2.1 Rotor binding inspection

This inspection is recommended as an important prerequisite to each controlled pump start. The power to the pump driver should be disconnected to prevent inadvertent start-up during this inspection. A suitable spanner wrench or other tool should be applied to the coupling and the pump rotor turned. The inspector should determine whether any resistance to rotation is excessive and should listen for rubbing noise that may result from insufficient clearance between rotating and stationary parts, bearing failure, or shaft seal failure. For consistent inspection results, it is recommended that the torque required to maintain rotation (free rotation torque) be measured and compared with the manufacturer's specifications or a previous baseline measurement. Acceptance criteria should be made available to determine whether the free rotation torque is excessive. If excessive rotating resistance exists, the cause should be investigated and corrected to avoid the possibility of catastrophic failure during startup or operation. A stethoscope or similar device can be used to determine whether the rubbing is within the pump or driver. The following inspection steps are suggested to minimize the investigation effort:

1. determine whether binding is in the pump or driver;
2. check for proper lubrication;
3. determine whether binding is caused by misalignment of the pump and driver; and
4. upon determining that the pump is at fault, disassemble and inspect the pump in steps according to the manufacturer's procedure or other approved procedure to determine the following:

   — quality of lubricant;
   — seal condition;
   — bearing condition, including thrust bearing;
   — corrosion, scoring, pitting, and overheating effects on rotor and stationary surfaces; and
   — dimensional changes of critical pump parts and wear surfaces.

4.2.2 Visual inspection

Regular inspections are those that are normally performed on a routine basis and do not require disassembly of the pump. Regular visual inspections should check the following:

1. lube oil level or quantity,
2. shaft seal leakage rate,
3. casing split leakage rate,
4. lube oil system leaks,
5. coupling lubrication leaks, and
6. external component or fastener looseness.

Periodic inspection, surveillance, and maintenance that should be performed at certain intervals as described in Chap. 5 require disassembly of the various pump parts. This inspection should include a visual inspection and be combined with a dimensional inspection to include at least the following:

1. radial bearings (journal or antifriction);
2. thrust bearings (tilting pad or antifriction);
3. shaft seals, packing, packing gland, stuffing box, and shaft sleeves;
4. coupling;
5. fasteners;
6. wear-ring surfaces and clearances;
7. thrust balancing device (drum or disk);
8. impellers;
9. diffusers and return channels;
10. shaft; and
11. casing split surfaces.

Chapter 5 discusses periodic inspection, surveillance, and maintenance in more detail.

4.2.3 Motor power monitoring

Motor power monitoring can correlate the changes in developed head and delivered flow. Comparing the changes in measured head and flow with a resulting change in power will add confidence to the head and flow measurements. The power measurement would be most useful in the part of the pump performance region that shows a rising brake horsepower curve as a function of flow rate. At high-flow rates the power curve often "flattens," thus making it difficult to measure power changes unless the pump efficiency were to change significantly.

Increases in internal clearances caused by wear and aging can result in an increase in power at a given flow rate because the impellers are pumping both the delivered flow and the internal recirculation flow.

4.2.4 Rotational speed monitoring

The ability of a centrifugal pump to produce both developed head and delivered flow for a given impeller diameter is a function of pump rotational speed. ASME Sect. XI IWP requires that pump speed be measured on pumps powered by variable speed drivers, such as steam turbines or diesel engines.

Speed measurement is required so that the proper head and flow correlations can be established to compare pump hydraulic performance against the base parameters. A baseline speed must be chosen and the pump developed head, flow, and horsepower corrected to this base condition. To properly judge whether the pump head and flow are degrading because of aging and service wear, the actual speed should be measured and the performance corrected through the "affinity laws" to baseline speed conditions. It is also important to measure the speed of motor-driven pumps because voltage and frequency variations to these drivers will affect the motor speed. The speed measurement may also be used to confirm a problem with the driver power source.

A permanently installed speed measuring instrument, such as an eddy current proximity probe, should be installed on the pump to read a signal from the pump shaft. Such an instrument will provide repeatable measurements and minimize errors obtained with a portable device, such as a stroboscope or tachometer. The proximity probe can also double as a key phasor indicator for the rotor vibration measurement to facilitate rotor vibration phase angle measurements.

4.2.5 Dimensional inspection

This inspection should be performed concurrently with the periodic inspection, surveillance, and maintenance visual inspection. As explained in Sect. 3.2, these measurements should include the impeller, wear rings, shaft, bearing journals, keyways, thrust runner, thrust balancer, shaft seal wearing parts, and coupling.

Pump vendors normally will not provide detailed part drawings to a customer; however, they will provide the nominal new part dimensions and tolerances as well as recommended dimensions to be used as guidance for part replacement. These dimensions should be recorded and compared with the new part dimensions. Proper records should be kept of all dimensions to establish a history of part wear.

4.2.6 Pump pressure or developed head monitoring

As a pump ages and wears, its ability to produce delivered flow and developed head reduces because of the reduction in hydraulic efficiency caused by worn hydraulic parts. The effect of wearing parts on developed head is described in the following subsection on delivered flow. It is necessary to measure both the pump developed head and delivered flow to determine specifically how the characteristic curve is degrading.

Existing plant instrumentation used to measure developed head normally consists of local suction and discharge pressure gages. Often, the main control board contains a discharge pressure indicator. The existing pressure measuring instrumentation may not be sufficiently accurate to measure small changes in developed head. Therefore, it is recommended that accurate pressure gages, or preferably a differential pressure transducer connected to a digital readout, be used to measure and trend small changes in developed head.

4.2.7 Pump-delivered flow monitoring

The primary purpose of an AUXFP is to deliver flow to the feedwater system to provide secondary system emergency heat removal. Therefore, it is important to accurately measure pump delivered flow.

As a pump ages and wears its hydraulic efficiency is reduced, and for a given system resistance and pump speed, the pump-delivered flow will be reduced. The reduction in efficiency results from the change in wear-ring clearances between the impeller and stationary wear surfaces and between the hydraulic balancing device (drum or disk) and its mating wear surface. When these clearances are excessively opened, the hydraulic performance of the pump suffers because of the increased internal stage-to-stage and balance return-line leakage. A centrifugal pump is "volume sensitive," meaning that it will develop head as a function of flow. When the increased stage-to-stage recirculation flow is added to the normal through flow, the flow through the impeller is increased, and because of the rising head characteristic, the pump impeller provides less head. Each pump impeller is actually pumping the delivered flow and the recirculation flow so that the net effect is to develop less head than that developed with only the delivered flow. Figure 2.7 shows the effect of this recirculation flow on the pump head capacity curve. A secondary effect of this recirculation leakage is the disruption in the main flow component as it approaches the impeller eye. In Vol. 1, Fig. D.3 demonstrates the secondary flow effects caused by the recirculation leakage at the impeller eye. Although this figure is designed to show the effect on inlet flow at off-design conditions, it also demonstrates a similar condition resulting from the recirculation leakage.

Presently, testing on AUXFPs is performed at the pump minimum-flow rate. As explained in Appendix D, Low-Flow Testing, the effect of aging and service wear is more difficult to detect in the low-flow region. Technically, the change in delivered flow caused by wear and aging is greater in the low-flow region than in the high-flow region because at lower discharge flow rates, the internal recirculation loss will increase with the increased head by the square root of the head difference. The greatest flow loss effects will result when testing at low-flow, high-head conditions, whereas the head degradation will be more pronounced at high-flow, low-head conditions.

Although testing at miniflow may confirm pump mechanical operational readiness, it may not demonstrate the hydraulic operational readiness of the pump, which is expected to deliver flow in an emergency at conditions closer to the pump BEP.

Most AFW systems now contain a flow-measuring meter in the system header. This meter may be an orifice or venturi. A delivered flow readout is also contained on the plant main control board. However, because we are often looking for small changes (<5% flow), it is unlikely that this readout is sufficiently accurate to predict small changes in pump performance. If the existing flowmeter system is judged not to be sufficiently accurate to measure small changes, the meter upstream and downstream taps should be connected to a differential pressure transmitter and ultimately to a digital readout so that accurate and repeatable flow readings can be monitored, trended, and evaluated. A dual-range instrument system should also be considered. One instrument would be selected to measure low flows accurately, while the second instrument would measure in the high flows. This system is probably the best to maximize accuracy and repeatability at all flow conditions.

4.2.8 Rotor vibration monitoring

Comparison studies have shown the value of a vibration surveillance program in reducing both pump failures and maintenance costs. This demonstrated value has resulted in the alteration of vibration monitoring policies. Whereas previous practice used intermittent or as-needed vibration monitoring as a diagnostic, troubleshooting tool, plants are increasingly using vibration monitoring in combination with other proven methods to implement predictive or preventative maintenance programs. With AUXFPs, operational readiness is critical to plant systems' functionability and safety; therefore, the value of vibration monitoring increases. Periodic vibration monitoring of AUXFPs is recommended to enhance and complement existing operational readiness testing. Vibration monitoring during operational readiness testing can provide additional means for the early detection of abnormal vibration caused by bearing wear, shaft seal deterioration, impeller and wear-surface wear, rotor unbalance, driver misalignment, and shaft failure.

As discussed in Chap. 2, as a pump wears, the close running internal clearances and the bearing clearances increase and change the rotor dynamics. As wear progresses, the rotor vibration amplitude normally increases and vibration responses caused by multiples of the rotation speed may change. By monitoring and trending the change in vibration amplitude and frequency, it becomes possible to predict the extent of wear and aging of the machine.

Recommended sensors include permanently installed noncontacting rotor shaft proximity probes to measure shaft displacement. The proximity probes should be mounted 90° apart on the circumference of the bearing housing. Bearing housing vibration should also be measured in the same directions as the probes to determine shaft absolute vibration amplitude. Bearing housing vibration can be measured using either the dual-probe concept or bearing housing accelerometers. The dual-probe concept uses a velocity transducer to measure housing vibration in addition to shaft vibration. After integrating velocity signals, displacement signals from a proximity probe and velocity transducer are summed to provide shaft absolute motion. Data forms, such as Lissajous (orbital) patterns, frequency vs amplitude plots, and time vs amplitude plots can be generated from proximity probe data.

Lissajous (orbital) patterns are produced by inputting the proximity probe signals to the X and Y axes of an oscilloscope or signal analyzer and observing or recording the pattern produced by amplitude changes in the two signals with respect to time. The Lissajous pattern describes the motion of the rotor shaft relative to the bearing housing. Figures 4.1–4.4 show Lissajous patterns along with corresponding time histories and frequency spectra for increasingly complex data signals. Time histories and frequency spectra are included to illustrate the need for evaluating multiple data forms because the Lissajous pattern can be complex and inconclusive, depending on signal content and data signal filtering. Figure 4.1 shows an ideal Lissajous pattern, corresponding time histories, and frequency spectra. The data shown in Fig. 4.1 would result from a single frequency response, such as rotation speed. In actual data, other frequency responses, such as machine rocking, shaft modes, and piping vibration modes may affect the Lissajous pattern. These responses should be identified to recognize and trend only the responses that are significant to pump degradation. Figure 4.2 shows a typical Lissajous pattern resulting from misalignment, bearing wear, or possible unbalance. The X-amplitude is significantly larger than the Y-amplitude. The data in Figs. 4.1 and 4.2 may result from band-pass filtering of the 1X-rpm rotation speed, which would exclude lower and higher frequency responses, leaving only the rotation speed component. Figure 4.3 shows patterns that could result from a combination of vibration indications, including misalignment, unbalance, rubbing, sliding,

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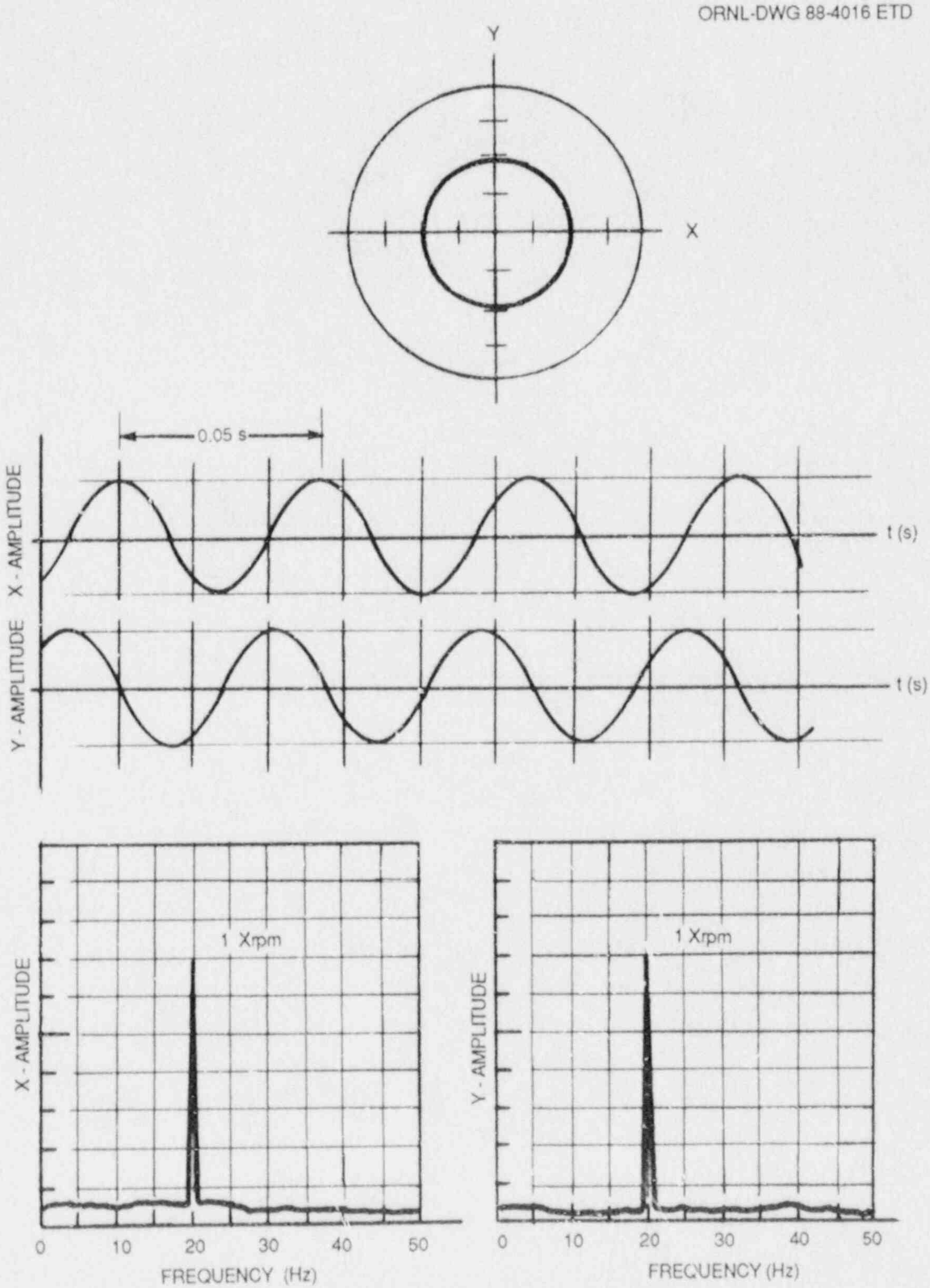

Fig. 4.1. Ideal 90° out-of-phase time histories and frequency spectra for shaft proximity probes mounted 90° apart on the circumference of an AUXFP shaft.

ORNL-DWG 88-4017 ETD

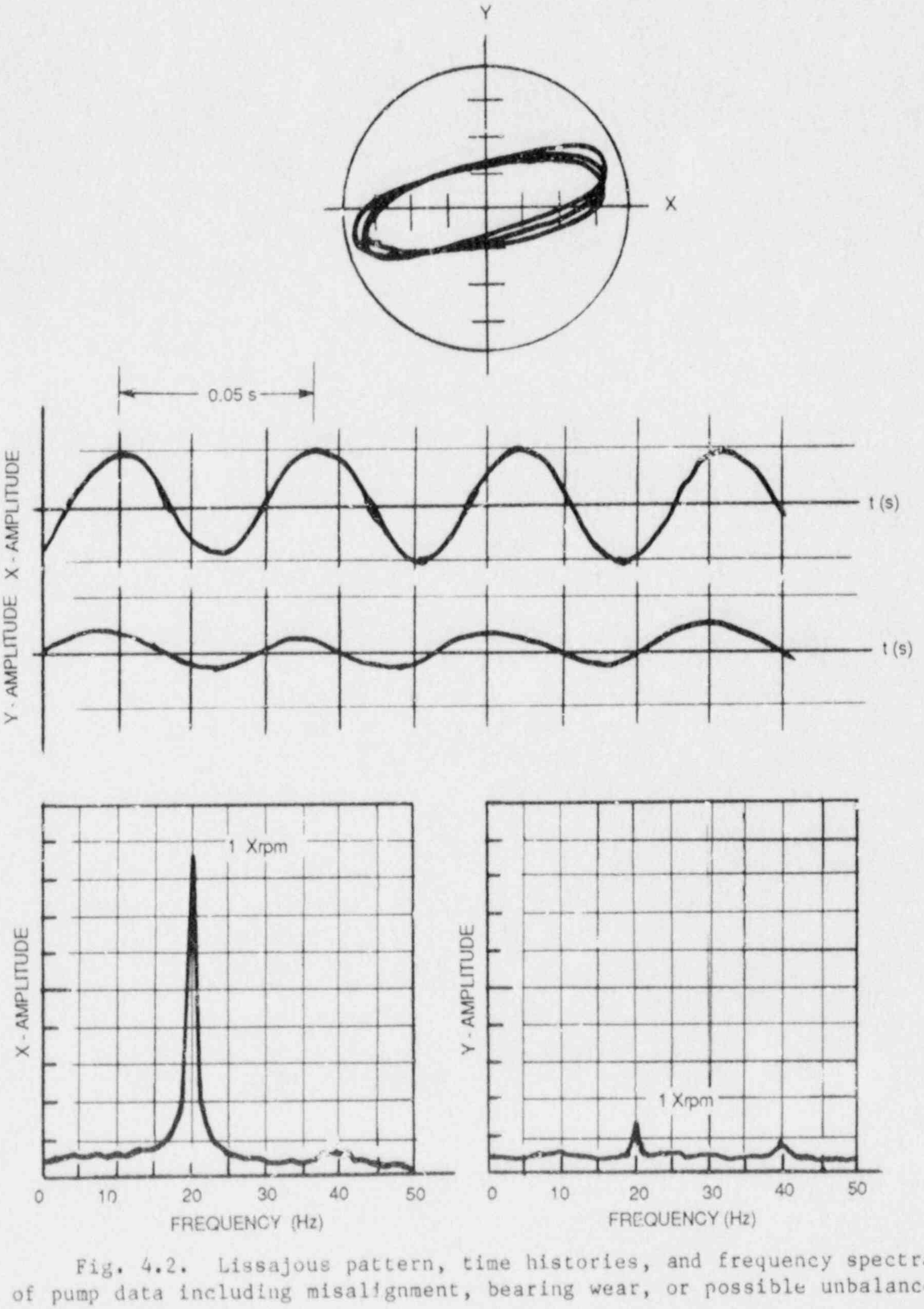

Fig. 4.2. Lissajous pattern, time histories, and frequency spectra of pump data including misalignment, bearing wear, or possible unbalance.

ORNL-DWG 88-4018 ETD

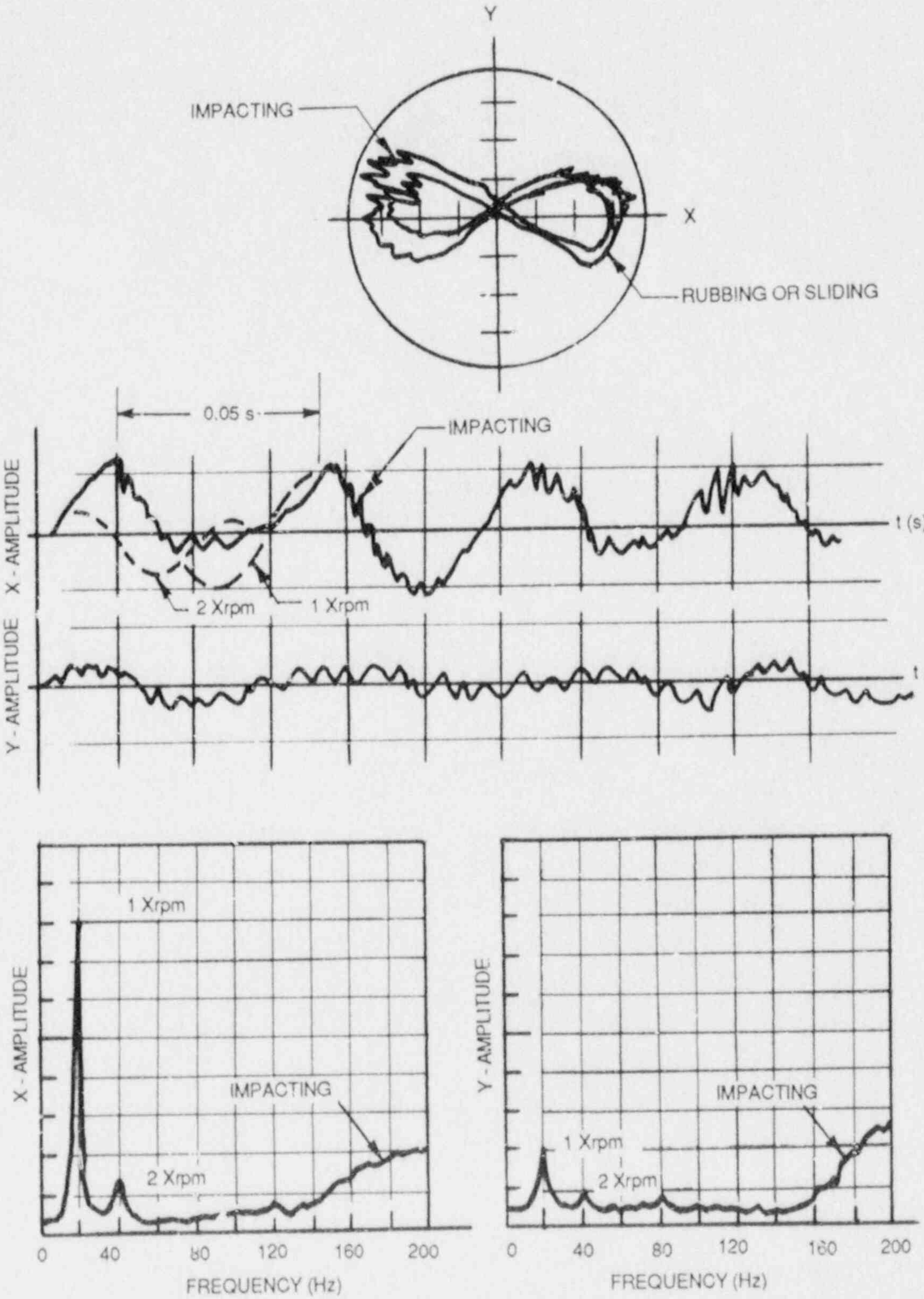

Fig. 4.3. Lissajous pattern, time histories, and frequency spectra of pump data showing a combination of misalignment or unbalance, rubbing, sliding, and impacting.

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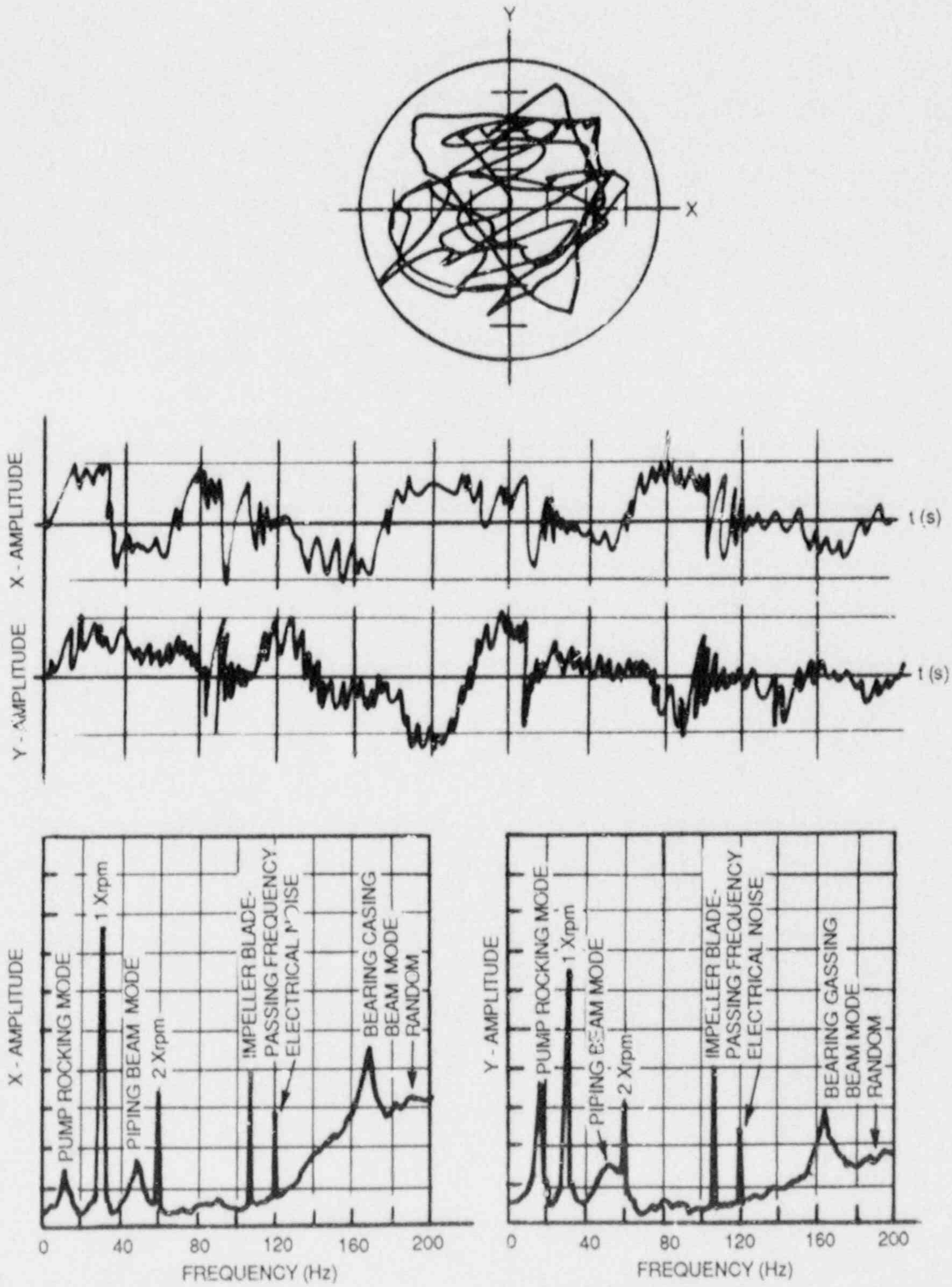

Fig. 4.4. Lissajous figure, time histories, and frequency spectra of pump data showing a combination of possible vibration indications.

and impacting from bearing wear. Figure 4.3 Lissajous pattern shows the double loop expected from the 2X-rpm frequency component along with the 1X-rpm frequency component and higher frequency random vibration caused by impacting. Although Fig. 4.3 shows random vibration caused by impacting occurring at <200 Hz, impacting indications may occur at much higher frequencies. Impact frequencies may be >20 kHz, but an upper frequency of 5 or 10 kHz is generally adequate for impact investigations. Further, the spectrum may show some broadening of the 1X rpm resulting from speed changes caused by rubbing and sliding. Figure 4.4 shows a typical Lissajous figure, time histories, and frequency spectra resulting from severe rubbing, sliding, and impacting. Although difficult to observe, this type of Lissajous pattern may contain harmonics of the rotational speed and random vibration along with rubbing, sliding, and impacting indications. The result may appear as random shaft motion but, most importantly, shows relatively large amplitudes. Figure 4.4 also shows other possible vibration responses that may be observed in the frequency spectra, thus illustrating the need to understand the vibration signature of the pump or other equipment and to determine which responses are useful for diagnostic and trending purposes.

The *ASME Boiler and Pressure Vessel Code*, Sect. XI, Subsect. IWP, states the rules and requirements for the in-service testing of Class 1, 2, and 3 centrifugal- and displacement-type pumps installed in light-water-cooled nuclear power plants. However, Article IWP-2000 states, "These tests are not designed to establish complete pump performance."[1] The in-service test requires the measurement of rotor displacement in the frequency range of one-half minimum speed to at least maximum pump shaft rotational speed along with other parameters. This discussion of the ASME Code requirements is included here to illustrate that the recommendations of this section are a logical extension of current in-service testing practice.

During in-service testing, it is recommended that rotor shaft vibration data and other data be recorded on a Frequency Modulated (FM) tape recorder with an expanded vibration frequency range limited only by proximity probe maximum response. With the availability of multichannel tape-recorded data, it becomes possible to study and trend many additional pump vibration characteristics in addition to meeting in-service testing requirements. These recommendations are not intended to imply that the use of hand-held vibration equipment should be discontinued. Although hand-held equipment is valuable for many applications, the data collection and analysis capabilities of hand-held equipment are limited in comparison to the wide range of available data collection and analysis hardware and software.

Table 4.2 contains a list of vibration causes and characteristics that are useful in relating vibration to failure data. Figure 4.5 illustrates possible trending of frequency spectra and shows the progression of bearing vibration to near failure. Other pump operating problems, such as cavitation and flow turbulence, may result in bearing vibration spectra similar to those shown in Fig. 4.5. However, assuming the transducer is installed near the bearing and data from other machine locations show no changes, then the cause of the change in vibration signature at this bearing location should be investigated, particularly if the rotor vibration amplitude is approaching maximum in-service test

Table 4.2. Vibration causes and characteristics

Cause	Frequency relative to pump speed[a]	Amplitude	Remarks
Unbalance	1X	Radial, steady	Most common cause of vibration
Bent shaft	1X, often 2X, sometimes 3X or 4X	Axial — high	Axial vibration up to 2 times radial
Sleeve bearings	1X	Primarily radial	Shaft and bearing amplitude same
Roller bearings	10—100X	Radial — low	Frequency depends upon type of bearing; use acceleration mode; consider cepstrum analyses
Journal bearings loose in housing	Subharmonics of shaft speed; 1/2 or 1/3X	Primarily radial	Looseness may develop only at operating speed and temperature
Mechanical looseness	0.1X–1X, could also be 2X, 3X, or 4X		Note subharmonics of loose journal bearings above; includes foundations
Misalignment			Axial amplitude 0.7 or higher of vertical or horizontal
Parallel	1X, 2X, 3X	Radial	
Angular	1X, 2X, 3X	Axial — high	
Oil whirl or whip in journal bearings	0.4—0.49X	Radial, unsteady	Oil whip excites balance resonance
Blade/vane pass	Number of blades/vanes times X; also harmonics	Radial	Also look for radial vibration in direction of discharge piping
Cavitation	Random	Fluctuating	Can be up to 2 kHz
Gears	High, related to other meshing frequencies (X times number of teeth) and harmonics	Radial — low	Cepstrum analysis can be valuable technique
Fluidic forces	1X or sub 1X		
Common rubs	High, can be multiples of 1X–10X		

[a]X is machine speed in revolutions per minute.

Notes to Table 4.2

Unbalance — Unbalance is the most common cause of vibration. It occurs when the center of mass of an object is not also its center of rotation. The apparent presence of a "heavy spot" causes forces to be applied that result in vibration. Unbalance produces a vibration reading whose amplitude is directly proportional to the amount of unbalance and whose frequency is the same as the running speed of the machine being measured.

For example, an 1800-rpm motor that is unbalanced will have vibrations at 1800 cpm (30 cycles per second), and the amount of vibration will be proportional to the amount of unbalance.

Bent shaft — In case of a bent shaft, a high axial vibration level at a frequency of single and sometimes double the shaft revolutions per minute will be noted. By observing the axial vibration at bearing A and bearing B, with the pickup held in the same direction on both bearings, the phase indication will be that of a static pair condition (reference mark at the same position for both bearings).

Note that for every rotation of the shaft a vertical impulse on the out-board and inboard end of each bearing will occur. This results in two impulses of vibration for each rotation of the shaft. If the bent shaft condition occurs as deflection, or deformation, at operating speeds, it can in many cases be corrected by adding balance weights at the center of the shaft. This will tend to straighten the shaft, and the mass centerline will then coincide with the rotation centerline. Normal two-plane balancing should be followed.

Misalignment — Misalignment of couplings and bearings results in a high axial vibration reading. This may be as much as 1.5 times the vertical or horizontal readings. This condition generally occurs at two times the running speed of the machine, although it may also occur at one (or even three) times the running or primary frequency.

Oil whip — A condition known as "oil whip" may occur in lightly loaded sleeve bearings. This instability is reflected by vibration frequency near, but always lower than one-half, the actual shaft speed. Correction of "oil whip" can be made by changing the bearing clearances or bearing design, changing the positions of the oil groove, increasing the shaft loading, or changing the oil characteristics, such as viscosity.

Cavitation — Cavitation is a somewhat common problem with centrifugal pumps and normally results whenever the pump is operated above its designed capacity or with inadequate suction pressure. Since the pump is "starved," the fluid coming into the pump will literally be pulled apart in an attempt to fill the void that exists. This creates pockets, or cavities, of nearly perfect vacuum which are highly unstable and which collapse or implode very quickly. Because of their impactive nature, these implosions excite the local natural frequencies of the pump housing, impellers, and other related pump parts; and, since these implosions may occur at random intervals at various locations within the pump or piping, the resulting vibration will be random in amplitude and frequency. Thus, a vibration caused by cavitation in a pump may cover a broad frequency range, where individual amplitudes and frequencies are constantly changing.

A distinctive noise often accompanies cavitation. Mild cavitation may have the sound of sand being pumped, whereas a more severe cavitation may sound as though gravel was being passed through the pump. In cases of extremely severe cavitation, it may sound as though rocks are being passed through the pump.

Notes to Table 4.2 (continued)

Common rubs — Rubs are typical phenomena to which a machine has no corrective response. In most cases a machine experiencing a full rub condition will progress to an immediate and catastrophic destruction of the machine. Rubs are typically introduced in machines by introducing contact between the rotor and a stationary element (labyrinth seals, diaphragms, etc.). Contact may be initiated by increased mass unbalance, thermal bow, or other mechanism causing the shaft to deflect or bend beyond the available clearances or by misalignment of the stationary elements.

In most cases, a full rub is preceded by a partial rub. Since the rotating system has no self-correcting capability, the rub must be defined and eliminated. Dismantling of the unit and examination usually are required to determine the extent of damage.

Looseness — Mechanical looseness, coupled with misalignment of couplings or bearings, generally occurs at a frequency of 0 to 1X the machine speed (rpm). The amplitude of the machine is normally dependent on the amount of looseness and the design of the machine.

Unbalanced rotational forces (no system is in perfect balance) and changing torque loads result in impacting of the base twice for each revolution. Excessive bearing clearance will produce the same results. Essentially, looseness allows more vibration to occur than what would otherwise be expected. This is particularly true in antifriction bearings, where small amounts of unbalance result in large vibrations at the first and second harmonics of the shaft speed. In most situations, this vibration will significantly decrease when the unbalanced force or misalignment is corrected.

Bearings — Excessive clearance in sleeve bearings can normally be determined by comparing the vibration level reading on the bearing housing using an accelerometer or velocity pickup, as close to the bearing as possible, and the vibration level of the shaft itself using a noncontact probe. Note that the pickups must be in the same plane and direction when taking both readings.

In many cases, manufacturers balance an antifriction bearing part by rotating it on the shaft, rather than in the bearing on which it will be finally assembled. If the rotational center of the antifriction bearings in a race is not coincident with the physical center of the outer races, a difference in the mass center and rotational center of the shaft will occur. This obliterates any prior balance performed. If bearing replacement, after original machinery installation, does not take this factor into consideration, an unbalanced condition may occur.

In the case of antifriction bearings, the scoring or brinelling of the balls or rolls will produce a high-frequency, low-amplitude vibration. It is best to use acceleration or velocity readings to detect faulty antifriction bearings. (A high-frequency but low-displacement signal has a high velocity, being a product of frequency and displacement, and higher acceleration, being a product of the displacement times the frequency signal squared.)

Verification of poor bearing performance is made by shutting off the machine and observing that the high bearing frequency remains as a unit coasts down in speed. This high-frequency signal will normally be retained until the machine comes to rest. The frequency indication is usually from 5 to 20 times the primary frequency of the part contained by the bearings.

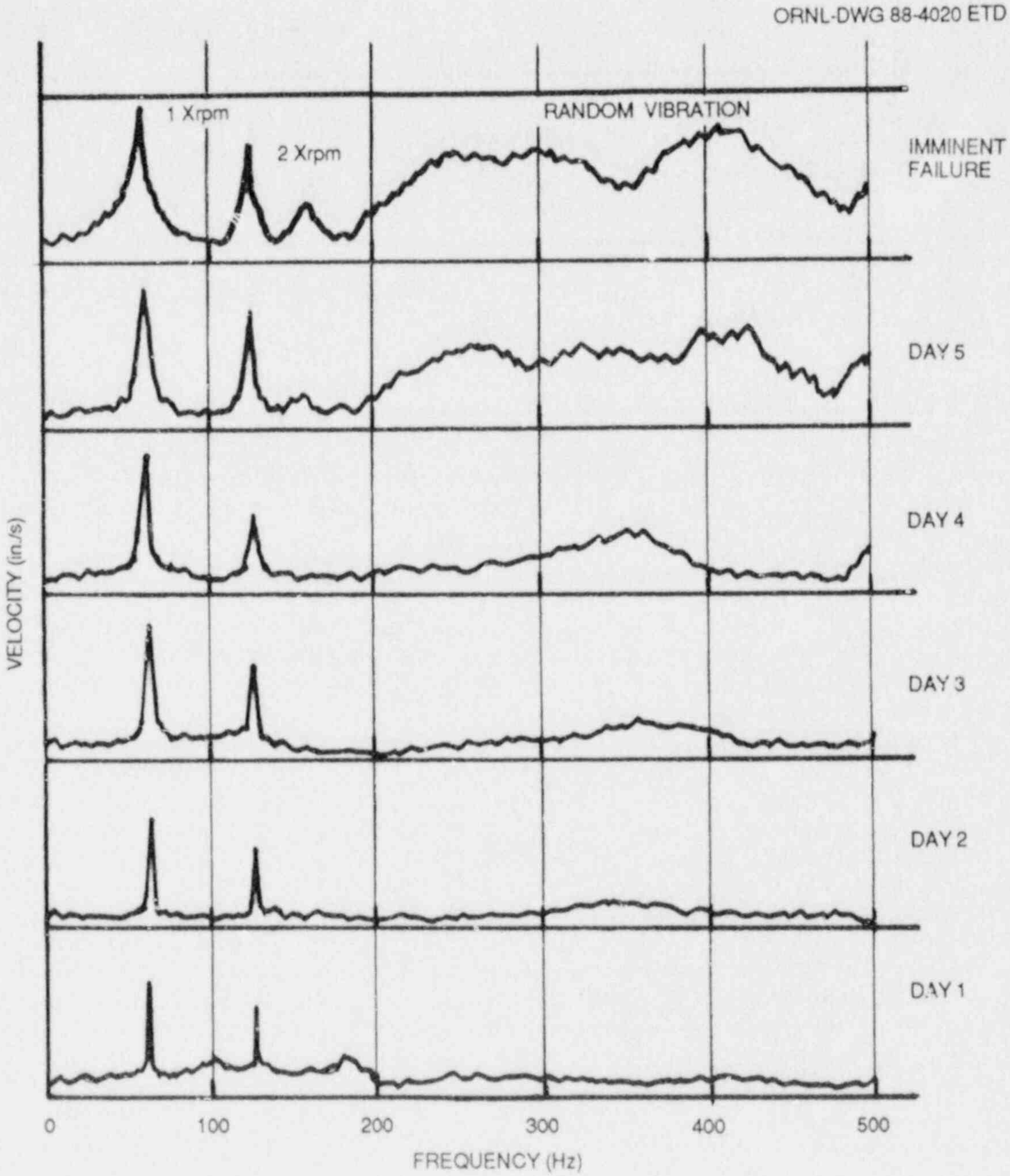

Fig. 4.5. Spectrum overlay of ball bearing vibration progressing to wear failure.

criteria. Using techniques similar to these, failure criteria can be generated, and pump conditions leading to failure can be identified and corrected.

Failure criteria in addition to in-service test criteria can be generated and refined based on experience and current studies. One such study is discussed in Vol. 1 of this report. Figures D.6—D.8 of Vol. 1 show some of the results of these studies. Particularly useful are data such as those reported in Fig. D.8, which provides allowable rotor vibration levels measured relative to the bearing cap.

In summary, the recommendations for pump vibration include permanent installation of shaft proximity probes and velocity or acceleration transducers that are capable of measuring pump casing vibration and recording data with a broad frequency response for investigating and trending multiple vibration characteristics.

4.2.9 IFD monitoring

IFD monitoring, sometimes called acoustic high-frequency or stress-wave emission monitoring, can be successfully applied to detect and predict rolling contact bearing, coupling, gear wear, or shaft seal deterioration. Acoustic frequencies are ones that are above the audible range or from 20 to ~300 kHz. In general, for AUXFPs, a piezoelectric sensor with a resonance near 100 kHz will provide the most useful data. Acoustic frequency responses are caused by rolling or sliding friction and metal deformation when metal components, such as ball bearings and gears are severely stressed. Severe stressing of components does not exist during normal machine operation; therefore, changes in acoustic responses are indicative of abnormal operation and potential failure. In the case of shaft seal failure, leakage between seal faces may produce high-frequency broadband random responses. These responses have been detected with IFD sensors and can be used to identify particular shaft seal failures. The attractiveness of IFD monitoring with respect to aging and service wear and failure detection is that responses show changes that precede low-frequency response changes. In general, IFD response changes may indicate a condition leading to a defect, and low-frequency response changes indicate the progression of an existing defect. Depending on bearing or gear design and application, changes in IFD responses may precede lower frequency vibration changes by days or weeks. Figure 4.6 illustrates the time relationship between conventional vibration monitoring signal amplitude and IFD emission signal amplitude. The root mean square (rms) value of the high-frequency signal shows an increasing amplitude before the low-frequency signal amplitude changes. Just prior to failure, IFD may show a decrease while low-frequency vibration increases. This finding is believed to be caused by the reduction in metal deformation at the onset of metal failure and mechanical defects.

Using signals from both low- and high-frequency sensors, the progression can be trended to provide the early detection of potential failure. For some actual bearing conditions, the sensitivity of IFD has been demonstrated. Changes in IFD emissions were identified well in advance of the detection of active defects. This sensitivity points out the need to identify for particular machines a level of IFD activity that is critical to avoid unnecessary early repairs. For example, referring to

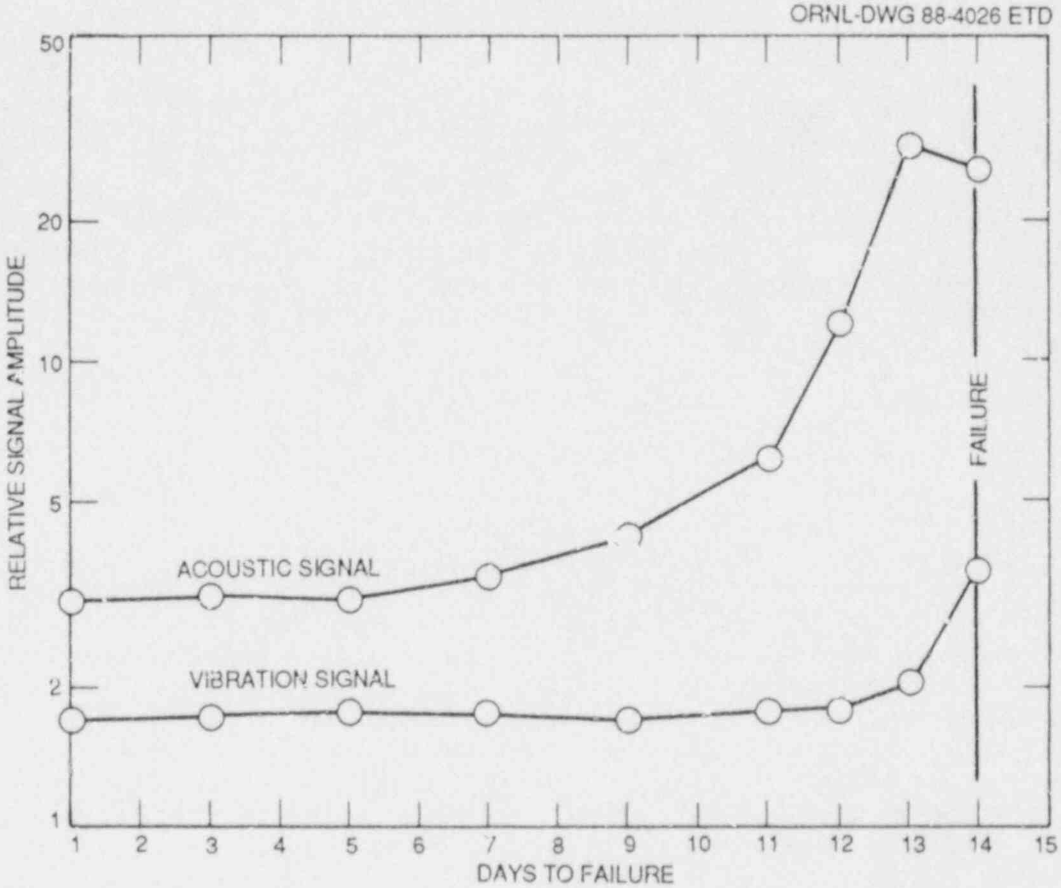

Fig. 4.6. Relative signal amplitude vs days to failure for acoustic and conventional vibration monitoring methods.

Fig. 4.6, the level of activity critical to operation may be specified initially as a factor of 4 increase in high-frequency signal level along with observable changes in low-frequency vibration responses. It is believed that critical emission signal levels can be identified by correlating and comparing data from shaft proximity probes, case accelerometers, and IFD sensors. Accelerometers are available with frequency responses up to 54 kHz, making possible some overlap of accelerometer and IFD data for both correlations and comparisons.

Because of the value of detecting machine abnormalities early, a piezoelectric sensor should be mounted as near as practical to each bearing, and these data, along with other data, should be used in studies to determine IFD emission activity relative to more conventional vibration monitoring. A realistic determination of signal emissions with respect to failure causes could lead to eliminating some of the conventional vibration monitoring equipment, thus reducing both monitoring and maintenance costs.

4.2.10 Liquid penetrant inspection

This inspection is intended to be performed during periodic inspection, surveillance, and maintenance. Highly stressed areas, such as fillets, grooves, keyways, and impeller bores should be inspected for such problems as cracks and surface indications. Liquid penetrant inspection is a well-proven and commonly used form of nondestructive testing. However, like all forms of nondestructive testing, the user must be experienced in the application of the method and in the interpretation of results.

4.2.11 Balance return-line flow monitoring

As described in Chap. 2, when the close running clearances of a pump wear, the internal leakage between stages increases and the pump's efficiency decreases. Since these wear-ring clearances also function as bearings, the shaft vibration increases as these clearances increase.

A measuring technique that has been used on boiler feed pumps to measure the change in clearance is to measure the increase of balance drum (or disk) return flow. The inner surface of a balance drum is exposed to high pressure (either full discharge pressure or an intermediate pressure), whereas the outer surface is at suction pressure. Consequently, a controlled leakage occurs through the close rotating clearances of the balance drum and its mating bushing. As with the interstage wear rings, leakage across the drum increases as the clearance increases. This leakage is termed "return flow" because it is returned to the pump suction. Although this technique only measures the flow through the balance drum clearances and not that through the interstage wear rings, it is generally accepted that an increase in balance drum clearance also reflects a similar change in the interstage clearances.

One method to measure the return flow uses an orifice installed between two flanges midway in the balance return line. Clamp-on ultrasonic flowmeters have also been installed on these lines. These meters are not as accurate as an orifice or venturi meter, but they do not require cutting into the line, nor do they produce a restriction and resulting pressure drop.

The flow through an annular clearance, such as a wear ring, is proportional to the clearance; therefore, the increase in flow is basically proportional to the clearance change. By first obtaining a baseline measurement of flow and then monitoring and trending the change, one can estimate the change in wear-ring clearance.

4.2.12 Audible noise inspection

This inspection, which requires the pump to be operating, can be combined with the regular visual inspection. It will require someone with experience in pump or machinery noise who is familiar with the pump and its operational history. A normally operating pump will produce a distinct audible noise signature, whereas an abnormally running pump that has developed a problem will often produce a change in audible noise.

Problems or failure causes associated with cavitation, rubbing, high shaft vibration, bearing wear, coupling wear, and so forth, can often be diagnosed by this method.

4.2.13 Bearing temperature monitoring

Bearings should be instrumented with thermocouples or resistance temperature detectors (RTDs) to measure bearing temperature. On AUXFPs equipped with antifriction bearings, the bearing housing must be drilled to accommodate a spring-loaded thermocouple. The thermocouple tip must rest tightly against the bearing outer race to enable fast response to any changes in bearing condition. For journal bearings, thermocouples should be permanently embedded in the bearing liner metal near the babbitt to backing interface. Embedment will require drilling through the bearing housing.

AUXFPs furnished with journal bearings have lubrication systems that include an oil reservoir, a shaft-driven oil pump, an auxiliary motor-driven oil pump, an oil cooler (oil-to-water heat exchanger), and interconnecting piping. Depending on the design of the lubrication system, additional temperature measuring equipment may be necessary to determine overall bearing performance with time. To determine or infer the heat generation in a bearing, the temperature change of the incoming and outgoing bearing oil should be determined. Therefore, it is recommended that the oil temperature leaving the bearings and the oil cooler be instrumented.

As a rolling element bearing wears, its temperature will increase because of the additional frictional heat generated. Bearing wear will also result in increased shaft vibration.

Bearing temperature and bearing oil temperature should not exceed values established by the pump vendors and the oil supplier. Temperature data gathered and recorded during an in-service test can be easily compared with baseline test data to determine whether a change in bearing performance has occurred. This data can be used to prompt additional data evaluation to pinpoint the cause of the change in bearing performance (e.g., bearing wear, shaft and bearing misalignment as a result of wear-ring wear, pump element flow instabilities, and unbalance). These changes are most apparent during start-up when the machine is heating from ambient conditions to normal operating conditions. Data manipulation is required to make comparisons because of changes in ambient temperatures and cooling water temperature. This task is easily accomplished using numerical techniques or graphical techniques. Changes in the slope, shape, or magnitude of a plot of temperature as a function of time will reveal a need for further investigation.

4.2.14 Rotor axial position monitoring

This measurement indicates the axial position of the rotor relative to the stationary pump parts. It can provide a wear indication of the thrust balancing device because a change in the clearance between the rotating balancing device and the stationary face or bushing can disrupt the hydraulic balancing capability of the balancing device. A change in axial position can also indicate the wearing of a thrust bearing, and in many instances, cavitation in the impellers can also disrupt the hydraulic balancing system of the pump. Both of these conditions can cause axial shuttling of the rotor. This measurement is often used on large fossil or process plant boiler feed pumps.

Most often, a proximity probe is installed to measure the rotor axial movement. The probe is installed on the outboard or thrust end of the pump against a relatively flat surface. The readout can be an analog or digital meter that may contain a relay to trip an alarm. The pump vendor should be consulted relative to the expected and maximum allowable rotor end float so that this condition can be properly monitored.

4.2.15 Bolt torque inspection

Many different fasteners are used in pumps to secure the various parts. Large fasteners secure the pump case to the support. Large fasteners are also used to clamp together the two halves of the pressure containment case. Smaller fasteners secure the shaft seal parts together. Besides the different fastener sizes, various materials are involved, all having different material and mechanical properties, such as corrosion resistance and yield strength.

Fastener wear or clamped-part wear often results from shaft vibration, thermal relaxation, improper clamped-part fits, or improper torquing during assembly. Undertorquing can lead to fastener wear and ultimate fatigue failure caused by the clamped parts rubbing or fretting against the fastener. Overtorquing can also lead to fastener failure caused by excessive preload to the fastener and clamped part, causing overload failures.

Fastener loosening or breakage has led to catastrophic failure of shaft seals, bearings, and even rotating assemblies. Improper torquing at gasketed surfaces, such as the pressure containment casing, suction or discharge flanges, stuffing box to casing joint, and bearing housing split lines, can lead to leaks caused by insufficient gasket seating load.

High shaft vibration is the single most common source of fastener loosening or breakage. Then again, casing or bearing housing vibration can be caused by loose support or housing fasteners. Component movement inside the pump caused by loose fasteners can often be diagnosed by monitoring rotor shaft vibration frequency spectrum plots. Loose parts will normally cause a one time or lesser multiple of running speed.

During the periodic inspection or whenever the pump is disassembled, bolt torques should be checked at disassembly and, of course, during reassembly. Many of the later pump manuals contain detailed torquing criteria for all fasteners. If the manuals are unavailable, this information should be obtained from the pump supplier. A suitable calibrated torque measuring device, such as a torque wrench, should be used to torque the fasteners to recommended pump vendor values.

Note that bolt torque values are a function of preload, bolt size, and material. If a thread lubricant is used, the [illegible] torque value must be adjusted to avoid overtorquing because the fastener will more easily tighten with the presence of a lubricant. In areas of high vibration, or when fastener loosening has been a problem, the use of a locking mechanism, such as a tab washer, lock washer, or double screw, may be warranted.

4.2.16 Leakage rate inspection

As noted in Sect. 3.2.19, leakage rate inspection consists of a visual approximation of the leakage rate. At installation, or during normal acceptable pump operation, the shaft packing should permit sufficient leakage to allow appropriate lubrication of the packing to keep the packing running cool. Leakage during this normal condition should be visually noted and used as a baseline parameter for reference later. Increases in the visually observed packing leakage rate should be carefully monitored and trended. Simple tightening of the gland nut may adjust leakage to acceptable levels, but care must be taken in order not to cut off leakage entirely with resultant overheating and damage to the packing and other stuffing box components. All packing adjustments should be recorded and trended along with the leakage data discussed previously. Based on this trending, aging and service wear degradation can be monitored and packing replacement indicated before overall pump failure occurs.

4.2.17 Lube oil analysis and quantity inspection

Many of the bearing problems reported involved oil contamination either by water or dirt. Research efforts begun years ago by the utilities industry have resulted in optimized lube oil analysis methods for light lubricating oils. Numerous laboratories located throughout the nation will perform analysis of lube oil. A thorough lube oil analysis program checks for appearance, water, flash point, viscosity, total acid number, and additive content.

The appearance test is purely visual. If free water is noted during the test, further testing is not warranted. Free water is highly undesirable in rotating equipment reservoirs because it can accelerate the formation of sludge and seriously lower the effective viscosity. On many occasions, the catastrophic failure of bearings has been attributed to free water in the oil. Flash point testing is performed whenever it is believed that light hydrocarbons may be present in the oil. If the oil is contaminated, the flash point of the oil is lowered. Viscosity is lowered by light hydrocarbon contamination.

The oil viscosity test is among the most important tests. Excessively low viscosity reduces oil film strength and deters the ability to prevent metal-to-metal contact. Low viscosity also hinders the ability of the oil to resist contamination and reduces its ability to seal. Excessively high viscosity impedes effective lubrication. Contaminants that thicken the oil may cause accelerated wear or corrosion of lubricated surfaces and may leave harmful deposits.

Acid formation can result from high temperature or overused and worn-out oil. Acidity can be considered a serviceability indicator that increases with progressive oxidation but can also be influenced by atmospheric contaminants.

Premium or good quality lubricating oils contain a phenolic oxidation inhibitor that breaks down after long-term operation of the oil after contamination by water or exposure to high operating temperatures. Oxidation inhibitors minimize the formation of sludge, resins, varnish, acids, and polymers.

Other tests, such as wear-particle analysis, can provide valuable information on component wear and incipient corrosion.

Lube oil quantity inspection must be an integral part of routine inspection and maintenance.

4.3 Parameter Monitoring to Establish Trends

The parameter monitoring methods recommended in the previous section should be implemented to determine operational readiness, to detect aging and service wear degradation, and to track this degradation. Existing instrumentation should be evaluated to determine whether this instrumentation will provide sufficiently accurate and repeatable data so that relatively small changes in performance can be measured to establish data trends.

We do not necessarily advocate the installation of "hard wired" instruments that directly wire the transducers to permanently installed back-mounted readout meters. However, we do recommend that pressure, temperature, and vibration measuring transducers be permanently installed on the pump (or piping systems) to minimize measuring errors. Data can then be collected using either existing readout meters (if they are sufficiently accurate) or collected with a hand-held portable and programmable electronic data recorder (a device that is becoming increasingly popular) to measure rotating equipment performance. These data can then be fed into a personal computer and analyzed. Numerous vendors design and produce software that can plot, analyze, and trend data. Software is available that can print caution or alarm conditions and mathematically extrapolate the data to predict the failure condition from the preestablished and program-entered alarm conditions. Software is also available that will perform vibration analyses, such as fast fourier transformation (FFT) analysis (otherwise known as spectrum analysis), which can be used to display vibration amplitudes vs frequency components. This technique is extremely valuable in diagnosing machinery problems and, when trended, in analyzing changes in the machine caused by aging and wear mechanisms. Monitored parameters that should be trended include

1. rotational speed,
2. vibration,
3. balance return-line flow,
4. developed head and delivered flow,
5. rotor axial position,
6. audible noise,
7. IFD response,
8. bearing temperature, and
9. motor power.

These trends should be analyzed and compared with acceptable established values, such as those provided by ASME XI (IWP), the American Hydraulic Institute, or the pump supplier, to predict the condition of operational readiness and the necessity to perform corrective maintenance on the diagnosed pump segment or part.

Data should be collected during each surveillance test and during plant start-ups or shutdowns whenever the AUXFPs are operated. No special testing outside of these requirements should be performed other than to diagnose special problems.

5. THE ROLE OF MAINTENANCE IN ALLEVIATING AGING AND SERVICE WEAR

5.1 Types of Maintenance

Three aspects of maintenance — predictive, preventive, and corrective are applicable to nuclear power plant safety systems. Figure 5.1 graphically displays the three modes of maintenance that have evolved with time. Now most nuclear utilities are primarily in transition between corrective and preventive maintenance.

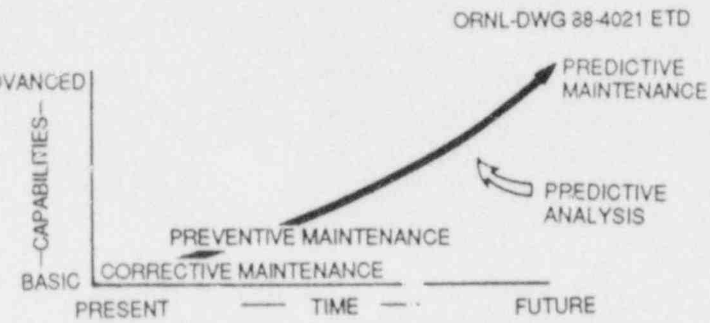

Fig. 5.1. Means to predictive maintenance.

5.1.1 Predictive maintenance

Predictive maintenance results from the combination of preventive maintenance and predictive analysis. Predictive analysis is the routine monitoring, trending, and analyzing of operating data for components and systems important to power plant productivity. By providing a key to planning a program of preventive maintenance through the baseline evaluation of equipment, predictive analysis ensures a streamlined, highly refined approach targeted to the operating trends and potential programs of the specific component. Combining the preventive maintenance approach with the unit-specific findings of predictive maintenance should result in

1. reduced component downtime;
2. forestalled maintenance outages;
3. forced outage avoidance;
4. optimized plans for upgrade, modernization, or plant life extension;
5. improved plant efficiency; and
6. reduced maintenance expenditures.

Once the utility adopts the predictive maintenance mode, it can adjust the scope and schedule of equipment inspection, repair, and/or replacement and thus achieve its maintenance goal by

1. focusing expenditures on activities with the highest probable impact on reliability,
2. eliminating unnecessary maintenance,

3. reducing actual maintenance costs through efficiency improvements,
4. reducing the cost of lost production, and
5. reducing insurance premiums resulting from fewer equipment failures.

If followed, recommendations provided in the NRC NPAR Program concerning inspection, surveillance, and monitoring will form good bases for the implementation of predictive maintenance practices in nuclear power plants.

5.1.2 Preventive maintenance

Plant maintenance is evolving from a corrective approach to a preventive one. Preventive maintenance involves routine service, inspection, and repair (or replacement) of equipment based on such factors as time, experience, vendor recommendations, codes, standards, and regulatory requirements. Utility initiatives in preventive maintenance have been valuable in both increasing plant efficiency and reducing forced outages. However, preventive maintenance can also lead to component problems, such as nonessential work. Without effective planning, management, and monitoring, the preventive approach can lead to higher than necessary maintenance costs. The critical goal now is to balance the progressive trend toward preventative maintenance with the potential cost of lost production. Without this balance, the utility may risk over or under investing in preventive maintenance with only marginal improvements in net power plant productivity. Achieving this balance or maintenance goal offers the minimum combined cost of production while maintaining component reliability. To assure long-term efficiencies in a preventive maintenance program, it is recommended that component-specific maintenance activities be developed by

1. applying engineering judgment and utility/station maintenance philosophy, by selecting, grouping, and evaluating known requirements to establish the most effective set of activities for the equipment; and
2. establishing baseline signature analysis of the equipment, monitoring the effectiveness of maintenance by periodic tracking of changes to the baseline condition, and taking appropriate action when significant changes occur.

5.1.3 Corrective maintenance

Corrective maintenance involves the actual repair of a disabled machine. It considers the corrective measures necessary to return a machine that is broken to safe and proper operation. Proper corrective maintenance relies upon all areas of maintenance, such as

1. review of condition monitoring results,
2. component preventive maintenance,
3. planned, streamlined procedures to minimize oversights or errors in assembly,
4. specialized tools,
5. adequate spare part inventories,

6. proper updating of equipment manuals, and
7. regular training of maintenance personnel.

If the nuclear plant is at power, corrective maintenance on AUXFPs or other safety components often requires working under a time limitation specified by the technical specifications. Typically, 72 h is allotted to complete the repair and return the component to service. Therefore, it is even more imperative that correct, updated, and efficient records, procedures, and spare parts be available to complete the repair to minimize component downtime and reduce the risk of plant forced outages.

5.2 Vendor Recommendations

As discussed in Vol. 1, vendor maintenance recommendations for AUXFPs are not very extensive. For example, one AUXFP vendor technical manual only includes the following regular maintenance information:

1. a general discussion of shaft seal stuffing box packing, including packing gland tightness adjustment instructions;
2. bearing lube oil change interval; and
3. bearing lube oil level setting.

Other instructions pertaining to cleaning, pump-to-driver alignment, bearing and coupling lubrication, bearing maximum temperature limits, and shaft vibration limits are provided under the sections entitled "Installation" and "Operation." However, the only information provided under the section specifically entitled "Maintenance" pertains to complete pump disassembly and overhaul. Inspection information includes

1. shaft runout tolerances,
2. bearing visual inspection guidelines, and
3. running clearance dimensions of rotating internal and stationary wear surfaces.

The information provided in the cited manual is typical of that provided in most AUXFP manuals. The lack of detailed regular or periodic maintenance instructions is not the fault of the vendor alone. Equipment specifications seldom indicate operating times or modes, and consequently, the vendor has no way of knowing the extent of regular maintenance to recommend. In retrospect, users often complain or ignore maintenance requirements that they believe are too stringent, frequent, or inconvenient to apply.

5.3 Current Utility Practices

The following describes typical maintenance now being performed on AUXFPs in nuclear power plants.

5.3.1 Regular maintenance

Utilities establish maintenance requirements for their equipment based on the pump vendor's technical manual, vendor technical bulletins, NRC inspection and enforcement bulletins, circulars, notices, INPO recommendations, and other information provided by industry.

A review of several utilities' maintenance activities has revealed that operations personnel regularly

1. inspect shaft seal leakage and adjust the gland as required,
2. inspect bearing oil level and adjust as required,
3. inspect for water leaks at the casing split, and
4. inspect for oil leaks in the lubrication system.

Furthermore, periodic maintenance includes changing the bearing lube oil or performing a lube oil analysis every 6 months. No complete disassembly, inspection, or maintenance is normally performed unless the pump develops a problem requiring this action.

5.3.2 Surveillance and condition monitoring

The plant technical specifications (part of the plant operating license), require that the AUXFPs, like all safety-related pumps, be tested normally each month to verify operational readiness. ASME Sect. XI Subsect. IWP (Refer to Vol. I, Appendix A) provides a listing of the parameters required to be measured. The general thrust of this test is directed toward a verification of the pump operational readiness.

The test is performed while the pump operates at miniflow and requires measurement of pump developed head, shaft vibration, speed, and bearing temperatures. Permanently installed instruments typically consist of (1) suction and discharge pressure gages, (2) motor current meter or turbine steam flow element (to measure pump power), and (3) a pump-delivered flow element. All other instrumentation is typically the portable type (e.g., vibration meters and an optical pyrometer).

Acceptance criteria for pump head is usually given in the plant technical specifications and ASME Sect. XI. Vibration limits are based on trending or change from a baseline, which is defined in ASME Section XI. Typically, the monthly test values are compared against acceptance criteria. The pump either passes the test unconditionally, rates an alert condition, or rates an alarm condition that results in a limited condition of operation (LCO). The utility will normally not perform diagnostic analysis unless an alert or alarm condition exists. Also, the test data are seldom used to establish a performance pattern or trend. The pump is tested only because regulations require it. If pump testing shows the pump to be in an alert range because of high shaft vibration, then utilities will obtain additional data, such as rotor vibration frequency spectrum plots, to diagnose the source of the problem. If the hydraulic performance is unacceptable, the pump will either be repaired or a safety analysis will be completed to justify the operation of the plant in its present condition.

5.4 Inspection, Surveillance, and Maintenance Practice Recommendations

Table 5.1 provides a detailed identification of recommended inspection and surveillance practices to identify and monitor potential failure causes. Note that this table lists categories of inspection and surveillance: nondisassembly, condition monitoring, and disassembly. The first category consists of routine actions to be carried out on a regular basis; the second follows from the recommendations given in Chap. 4; and the third involves disassembly and detailed examination as a part of periodic inspection, surveillance, and maintenance.

Maintenance actions, in summary, involve routine lubricating oil changes and additions, lubrication, shaft seal gland adjustments, and packing replacement. Other actions include cleaning, repairing, refurbishing, and replacing parts and components.

5.4.1 Regular nondisassembly inspection, surveillance, and maintenance

Regular inspection, surveillance, and maintenance involves the day-to-day preventive and corrective maintenance activities required to ensure machinery operational readiness. These activities include the ones that do not require pump disassembly. Many of these activities may be considered commonsense practice.

5.4.2 Surveillance and condition monitoring

Chapter 4 details the recommended monitoring methods to establish AUXFP operational readiness and to identify the condition of aging and service wear. These methods include vibration; balance return-line flow; rotational speed; delivered flow; developed head and rotor axial position; bearing temperature (sleeve, journal, and tilt pad thrust); and motor power monitoring and IFD (rolling element bearings).

We recommend the permanent installation of transducers into the various pump parts, such as proximity probes for vibration, speed, and rotor axial position; balance return-line flowmeter; bearing thermocouples; and pressure measuring transducers (pressure and flow differential pressure). Permanently mounted transducers will provide for more accurate and repeatable measurements. Transducer mounts for IFD measurements should be permanently installed on each bearing housing. These mounts normally consist of threaded studs that can be installed either by drilling and threading a short depth into the bearing housing or by cementing the mount to the housing with a strong adhesive especially made for transducer mounting.

Unless desired by the utility, hardwiring all of these transducers to a permanent readout metering system is not necessary. It is recognized that some of this instrumentation may exist in hardwired form and is already adequate for the duty. Portable instrumentation, such as hand-held "data loggers," is available and can be temporarily connected to the various permanently mounted transducers; then dynamic or static

Table 5.1. Recommended inspection and surveillance practices for AUXFPs

Segment	Part	Nondisassembly	Condition monitoring	Disassembly
Rotating elements	Shaft	Rotor binding inspection	Speed, rotor vibration, developed head, delivered flow	Visual inspection, runout inspection, penetrant test for surface indications
	Impeller	Rotor binding inspection, audible noise inspection	Speed, rotor vibration, developed head, delivered flow, balance return line flow, acoustic emission	Visual inspection, bore measurement, wear-surface clearance measurement, penetrant test for surface indications
	Thrust runner	Rotor binding inspection, audible noise inspection	Speed, rotor vibration, temperature, rotor axial position, acoustic emission	Visual inspection, penetrant test for surface indications
	Fasteners	Rotor binding inspection, audible noise inspection, visual inspection, bolt torque measurement	Speed, rotor vibration, rotor axial position, acoustic emission	Visual inspection, bolt torque measurement,
Nonrotating internals	Diffusers or volutes	Audible noise inspection	Developed head, delivered flow, rotor vibration, acoustic emission	Visual inspection, penetrant test for surface indications
	Wear surfaces	Rotor binding inspection, audible noise inspection	Developed head, delivered flow, rotor vibration, rotor axial position, balance return line flow	Visual inspection, wear-surface clearance measurement
	Fasteners	Rotor binding inspection, audible noise inspection, visual inspection, bolt torque measurement	Rotor vibration	Visual inspection, bolt torque measurement

Table 5.1 (continued)

Segment	Part	Nondisassembly	Condition monitoring	Disassembly
Pressure containment casing	Casing	Visual inspection, leakage inspection		Visual inspection
Mechanical subsystems	Bearings	Rotor binding inspection, audible noise inspection, oil level inspection, oil leak inspection, oil purity	Speed, rotor vibration, temperature, rotor axial position, acoustic emission	Visual inspection, wear-surface clearance measurement, rotor axial position measurement
	Shaft seals	Visual inspection, leakage inspection, audible noise inspection	Rotor vibration, temperature, acoustic emission	Visual inspection
	Thrust balancer	Rotor binding inspection, audible noise inspection	Rotor vibration, developed head, delivered flow, rotor axial position, balance return line flow, acoustic emission	Visual inspection, wear-surface clearance measurement
	Coupling	Rotor binding inspection, audible noise inspection, lubrication, leak inspection	Speed, rotor vibration, acoustic emission	Visual inspection
	Fasteners	Rotor binding inspection, audible noise inspection, bolt torque measurement	Rotor vibration	Visual inspection, bolt torque measurement
Support	Fasteners	Visual inspection, rotor binding inspection, audible noise inspection, bolt torque measurement	Rotor vibration	Visual inspection, bolt torque measurement

data can be recorded. These data then can be manually analyzed or used to develop trends or, more productively, entered into a personal computer and analyzed, with trends established through the use of pre-programmed software. Numerous programs and software are available and can perform frequency spectrum analysis, develop data trends, and perform alarm point extrapolation. This logging technique provides an automated permanent record of data and simplifies the trending and evaluation process. Also, it is much less expensive than the hardwiring method. This technique has been used extensively in the chemical and petrochemical industry for years and is now finding its way into the utility industry.

Data should be obtained during each required regulatory surveillance test. We do not advocate starting and operating an AUXFP for the sole purpose of obtaining these data unless additional diagnostic data are required for troubleshooting a pump problem.

It must be recognized that the implementation of such a formal program will require pump disassembly for the following purposes: transducer installation; possible purchase and maintenance of additional instrumentation; and personnel training in the installation, use, and evaluation of such hardware and the resulting test information. A successful program requires proper planning and thorough review before it is implemented.

Once the surveillance and condition monitoring program is implemented, the first step is to obtain a baseline evaluation of the equipment (for example, vibration, bearing temperature, and power) by using the monitoring methods recommended. Continued monitoring during the surveillance tests should measure the effectiveness of the entire maintenance program (i.e., predictive, preventive, and corrective) by tracking any changes to the baseline condition and taking appropriate action when significant changes occur.

5.4.3 Periodic inspection, surveillance, and maintenance

Periodic inspection, surveillance, and maintenance involves the disassembly, inspection, repair, and cleaning of an assembly after a designated operational time. AUXFPs are usually in a standby mode and consequently do not accumulate operating hours very quickly. Unlike high-speed boiler feed or chemical process pumps, most AUXFPs operate at <4500 rpm and therefore do not rapidly accumulate operational fatigue cycles.

Periodic inspection, surveillance, and maintenance intervals are usually established by the equipment supplier on the basis of experimental and development tests or by well-documented operating experience. Considering that AUXFPs do not accumulate many operating hours between refueling outages, we do not believe that complete disassembly and inspection after each refueling is warranted. Periodic inspection intervals, such as those associated with the aircraft industry, are based on specific component destructive testing. No comprehensive data base of this nature exists for AUXFPs. Consequently, it is difficult and unjustified to specify a distinct periodic inspection interval that must be practiced by each plant.

Each AUXFP design may warrant a different inspection interval that should be based on such criteria as mean time between failure evaluations. This inspection interval will therefore vary between plants.

The average operating time for an AUXFP, which also functions as a low-power feedwater pump, is estimated to be 500 to 750 h, for one refueling cycle (considered here to be 18 months). A periodic inspection, surveillance, and maintenance interval of 4 1/2 to 5 years or three refueling cycles would amount to 1500 to 2250 operating hours on a pump used for low-power feedwater operation (but substantially less for those not used for this service) and about 5 years of chronological time. The three-cycle refueling outage interval was selected as an engineering estimate for making the recommendations given here. It is not based on other evaluations, such as mean time between failure.

Although periodic inspection, surveillance, and maintenance has advantages, it also has many drawbacks. Needless disassembly of vital parts is a major disadvantage. Problems often arise when good parts are removed or replaced, no matter how much care is exercised. These problems may be a result of simply altering the normal fit. A prime example of a high probability of errors being made during the assembly-disassembly process is during bearing service. For example, fasteners can be overtightened or undertightened, and both nicks and burrs can result on critical surfaces. Dirt can also be introduced into the pump during these disassemblies and inspections, and care must be appropriately exercised. Experience has shown that frequent disassembly of an AUXFP will wear out the part and pump at a much higher rate than would be normally expected from the wear and aging process. Reference 2 discusses in greater detail the advantages and disadvantages of the periodic inspection, surveillance, and maintenance of machinery.

Table 5.2 provides inspection interval guidelines using 4 1/2 to 5 years as an inspection interval. We support complete periodic, inspection, and surveillance maintenance only for the purpose of developing a maintenance data base. The advantages of a regular periodic inspection maintenance program are outweighed by the problems one can experience during disassembly and reassembly. We recommend that after a data base is established, only the periodic inspection of the pump bearings, shaft seals, and bearing lube oil analysis be made. Bearing lube oil analysis is specified every 6 months. We believe that it is very important and will yield such data as bearing condition, moisture from packing leakage, and lubrication oil viscosity and cleanliness. Once a utility establishes a regular inspection and monitoring program, the optimum intervals for scheduled periodic inspection can be determined.

5.5 Maintenance Practice Evaluations

Table 5.3 lists maintenance practice evaluations for each failure cause identified in Chap. 2. The maintenance activities are those categorized as predictive, preventive, and corrective. Each maintenance practice is rated (high, medium, or low) with respect to effectiveness in preventing or correcting failures related to aging and service wear.

Table 5.2. Periodic inspection, surveillance, and maintenance interval guidelines for AUXFPs

Pump segment	Parts	Interval (refueling outages)
Rotating elements	Shaft	3
	Impellers	3
	Thrust runner	3
	Fasteners	3
Nonrotating internals	Diffusers or volutes	3
	Wear surfaces	3
	Fasteners	3
Pressure containment casing	Casing	3
	Suction nozzle	
	Discharge nozzle	
	Fasteners	3
Mechanical subsystems	Bearings	1/3[a], 3
	Shaft seals	3
	Thrust balancer	3
	Coupling	3
	Fasteners	3
Support	Fasteners	1–2

[a]Oil analysis.

Table 5.3. Maintenance practices evaluation

Failure cause	Maintenance practice effectiveness		
	Predictive	Preventive	Corrective
Bearing wear, corrosion, breakage	High	High	High
Shaft seal deterioration, breakage	Medium	High	High
Binding between rotor and stationary parts	Low	High	High
Impeller wear, breakage	High	Low	Low
Thrust balancer wear, galling, seizing	High	Low	Medium
Shaft breakage	High	High	High
Wear-surface wear, erosion, corrosion, seizing	High	Medium	Medium
Rotating element fastener loosening, breakage	Medium	High	High
Mechanical subsystem fastener loosening, breakage	Low	High	High
Thrust runner wear, breakage	High	Medium	High
Coupling wear, breakage	High	High	Medium
Nonrotating internals fastener loosening	Low	High	High
Leak at casing split	Low	High	High
Structural damage to stationary vanes (diffuser or volute)	Medium	High	Low
Support fastener loosening, breakage	High	High	High

6. SUMMARY AND CONCLUSIONS

AUXFP failure causes ranked in overall terms of importance were determined to be

1. bearing wear, corrosion, and breakage;
2. shaft seal deterioration and breakage;
3. binding between rotor and stationary parts;
4. impeller wear and breakage;
5. thrust balancer wear, galling, and seizing;
6. shaft breakage;
7. wear-surface wear, erosion, corrosion, and seizing;
8. rotating element fasteners loosening and breakage;
9. mechanical subsystem fasteners loosening and breakage;
10. thrust runner wear and breakage;
11. coupling wear and breakage;
12. nonrotating internals fasteners loosening and breakage;
13. leak at casing split;
14. structural damage to stationary vanes (diffuser or volute); and
15. support fastener loosening and breakage.

Measurable parameters that could be used to detect or monitor these failure causes include vibration, temperature, rotational torque, noise, appearance, bolt torque, leakage rate, speed, wear surface and critical fit clearance, motor power, lube oil purity, developed head, and delivered flow.

ISCM methods that were recommended to be most beneficial in assessing aging and service-wear-related failures include

1. rotor binding inspection,
2. visual inspection,
3. motor power monitoring,
4. rotational speed monitoring,
5. dimensional inspection,
6. pump pressure or developed head monitoring,
7. pump-delivered flow monitoring,
8. rotor vibration monitoring,
9. IFD monitoring,
10. liquid penetrant inspection,
11. balance return-line flow monitoring,
12. audible noise inspection,
13. bearing temperature monitoring,
14. rotor axial position monitoring,
15. bolt torque inspection,
16. leakage rate inspection, and
17. lube oil analysis inspection.

ISCM method evaluations were summarized in Chap. 4. Evaluation criteria included effectiveness, importance in terms of safety, and ease and cost of implementation. All ISCM methods were considered to have equally high effectiveness, with the exception of rotational speed monitoring, balance return-line flow monitoring, audible noise inspection, and bearing temperature and motor power monitoring, which were rated somewhat lower. Implementation cost for the predictive monitoring techniques is estimated to be high ($10 to $50K) if the pump must be disassembled or if the pump or system pressure boundary is affected by the instrumentation installation. Instrumentation hardware costs are considered to be low to medium ($5 to $10K) for all the instrumentation with the exception of that for delivered flow monitoring and balance return-line flow monitoring.

Permanent installation of instrumentation transducers into the respective pump parts is recommended to provide for accurate and repeatable measurements. Various recording and analysis hardware and software exist to obtain, process, analyze, and trend data. It is emphasized that a successful monitoring program requires proper planning and thorough review before it is implemented.

Many different parameters were identified for use in detecting and predicting various aspects of AUXFP aging and service wear. All of these parameters should be evaluated collectively to predict pump condition; that is, no single parameter should be used to track or trend pump aging and wear. The entire collection of parameters should be compared against each other to permit supportive correlations as to the accuracy of the results. (For example, one might use vibration monitoring, IFD monitoring, audible noise inspection, etc., to predict a bearing failure.) Interpretation of the parameter measurements can be simplified by using a "SMART" monitor, which is a computer-based system that can accurately and repeatably interpret the data obtained.

It should not be construed from this report that utilities are not following good inspection, surveillance, and maintenance practices. As stated in Chap. 4, ASME Sect. XI requires that utilities perform monitoring. In addition, utilities use, to some extent, many of the inspection and monitoring methods specified in this report. Some utilities are well on the way to developing good predictive maintenance programs.

Experience obtained during the past decade in pump technology has shown that hydraulic instabilities and unbalances can occur in pumps when operating at low flows. Operating pumps at higher flows (e.g., >25% BEP flow) is recommended to reduce the effect of hydraulic instability on performance. This also makes the test data more meaningful since, in reality, the pumps are not intended to operate at such low flows as miniflow (5–15% BEP flow) to perform their normal plant design functions. Increasing the flow rate should reduce vibration levels and consequently reduce longer-term wear and aging effects on pump parts such as bearings, seals, wear rings, etc. Obtaining test data at higher flows (i.e., >25% BEP flow) should also provide more accurate and repeatable measurements for the purpose of trending aging and service wear. Therefore, such things as the W-developed AUXFP test guidelines (App. G) should be evaluated or other alternatives for increasing flow rates should be considered to accommodate acceptably higher flow rate testing.

Three types of maintenance — predictive, preventive, and corrective — were reviewed in this report. Predictive maintenance results from the combination of preventive maintenance and predictive analysis. Predictive analysis involves monitoring, trending, and analyzing operating data; while preventive maintenance involves service, inspection, and repair (or replacement) of equipment, based on such factors as time, experience, vendor recommendations, codes and standards, and regulatory requirements. Vendor's typical AUXFP technical manuals for pump maintenance normally lack sufficiently detailed regular or periodic maintenance instructions. Utility regular maintenance essentially consists of external visual inspections, while periodic maintenance usually results only in changing bearing lube oil. No complete disassembly, inspection, or maintenance is performed unless the pump develops a problem requiring this action. Surveillance and condition monitoring is performed in accordance with the plant technical specifications following the guidelines specified by ASME Sect. XI and NRC Standard Review Plan (SRP) 3.9.6.

Regular inspection, surveillance, and maintenance activities embrace routine preventive and corrective maintenance actions; the inspections include rotor binding, audible noise, oil level, oil leak, coupling grease leak, and water leakage rate inspection. Periodic inspection, surveillance, and maintenance involve the disassembly, inspection, repair, and cleaning of an assembly after a designated operational time; intervals for performance are usually established by the equipment supplier on the basis of experimental and developmental tests or by well-documented operating experience. However, no comprehensive data base exists for AUXFPs to justify a specific time interval.

Interim schedules for periodic inspection, surveillance, and maintenance based on 4 1/2- to 5-year intervals are given in this report for items other than pump bearing and shaft seal inspections and lube oil analyses. The intervals stipulated for the latter are recommendations. Once a utility establishes an inspection, surveillance, and maintenance program, optimum intervals can be determined.

Good maintenance practices, besides utilizing inspection, surveillance, and monitoring techniques, rely on well-planned and streamlined procedures to minimize errors during the maintenance process, an adequate spare parts inventory, regular training of maintenance personnel, and continuous updating of equipment technical manuals.

Maintenance practice evaluations were summarized in Chap. 5. These evaluations considered the effectiveness of predictive, preventive, and corrective maintenance practices in preventing or correcting aging and service-wear-related failures.

The recommendations in this report should be transmitted to current code and standard development groups, such as the ASME Operation and Maintenance Committee, Working Group on Pumps and Valves, for possible future incorporation into codes and standards.

Further studies should consider the development of methods to monitor and trend shaft seal leakage, motor current, and turbine power. These parameters can potentially provide additional useful information concerning AUXFP aging and service wear. Studies should also consider developing "SMART" systems for monitoring and interpretation.

Finally, Vols. 1 and 2 of this report considered the pump only. Many of the problems reported for the AFW system involve the steam turbine drivers for pumps and control systems. Therefore, consideration should be given to evaluating problems encountered in these areas to enhance the overall reliability of AUXFPs and systems.

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Appendix A

DEFINITIONS

Absolute motion — Vibration measured with respect to an inertial (fixed) reference frame, which is often loosely referred to as "free space." Seismic transducers measure machine casing or structural absolute motion.

Acceleration — The time rate of change of velocity, which is often expressed as G, A, $\ddot{x}$, or d^2x/dt^2. Acceleration leads velocity by 90° in time and leads displacement by 180° in time. Typical units for acceleration are feet per second per second (ft/s^2), meters per second per second (m/s^2), or more commonly G's, zero-to-peak (where G = acceleration of gravity = 32.17 ft/s^2 = 9.81 m/s^2). Acceleration measurements are generally made with piezoelectric accelerometers and are typically used to evaluate high-frequency machine casing or bearing housing response characteristics.

Active pump — A pump that must perform a mechanical motion during the course of accomplishing a system safety function.

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.

Alignment — A condition whereby the axes of machine components are positioned according to design requirements. A measurement of the relative position of a machine component with respect to another. Relative alignment measurements can be made from bearing to bearing, rotor to rotor, rotor to bearing, bearing to casing, casing to foundation, casing to piping, etc. Various alignment requirements utilize different techniques of cold and hot machine measurement, including optical, mechanical (dial indicators), and electronic (proximity probes).

Ambient temperature — The average or mean temperature of the surrounding air that comes in contact with the equipment and instruments being tested.

Ammeter — An instrument used to measure current.

Amplitude — The magnitude of dynamic motion or vibration. Amplitude must be specified in terms of peak-to-peak, zero-to-peak, root mean square (rms), or average. Peak-to-peak equals the maximum excursion from the extreme positive peak to the extreme negative peak, and zero-to-peak equals one-half of peak-to-peak. For pure sine waves only, rms equals 0.707 of zero-to-peak, and average equals 0.637 of zero-to-peak. Root mean square is also called effective amplitude and is useful for acoustical studies. For vibration measurements, amplitude is usually expressed in terms of displacement, velocity, or acceleration.

Asynchronous — Vibration frequency components that are not related to shaft rotative frequency. Also referred to as nonsynchronous or nonintegral motion. See Synchronous.

Attenuation — The reduction in amplitude of a given signal without changing the characteristic waveform.

Axial position — The average position, or change in position, of a rotor in the axial direction with respect to some fixed reference. Ideally, the reference is a known position within the thrust bearing axial clearance or float zone, and the measurement is made with a proximity probe observing the thrust collar. However, the measurement can be made effectively by a probe observing some other integral axial shaft surface, if such surface is close to the thrust bearing. Also, the measurement of this position or change in position. Also called thrust position.

Blade passing frequency — A potential vibration frequency on any bladed machine (turbine, axial compressor, fan, propeller, etc.). It is represented by the number of blades (on a disk or stage) times the shaft rotative frequency.

Bow — A shaft condition such that the geometric shaft centerline is not straight. Usually, the centerline is bent in a single direction because of gravity sag, thermal warpage, etc; however, the bow may be three-dimensional (corkscrew).

Calibration — The process of adjusting an instrument or compiling a deviation chart so that the instruments reading can be correlated with the actual value being measured.

Cavitation — The formation of vapor bubbles in any flow that is subjected to an ambient pressure equal to or less than the vapor pressure of the liquid being pumped.

Compensated connector — A connector made of thermocouple alloys used to connect thermocouple probes and wires.

Corrective maintenance — Those maintenance activities involving the actual repair of a machine or component after failure is incurred.

Current — The rate of flow of electricity. The unit is the ampere (A), defined as 1 ampere = 1 coulomb per second.

Data acquisition system — The system consisting of analog multiplexers, sample-holds, analog-to-digital converters, and other circuits that process one or more analog signals and convert them into digital form for use by a computer.

Displacement — The change in distance or position of an object. Displacement is typically a peak-to-peak measurement of the observed motion and is usually measured in mils or micrometers. Eddy-current proximity probes measure displacement directly. Because integration of a velocity signal is necessary to obtain displacement, an acceleration signal requires double integration to yield a displacement measurement.

Dual-element sensor — A sensor assembly with two independent sensing elements.

Dual probe — A transducer set consisting of a proximity probe and velocity transducer installed radially at the same point (usually in a common housing) on the machine casing. Four separate measurements are provided by this transducer system. The proximity probe measures (1) shaft radial position within the bearing clearance and (2) shaft dynamic motion relative to the bearing. The velocity transducer measures (3) machine casing absolute motion. When the velocity signal is integrated to displacement and vectorially added to the shaft relative displacement, the summation represents (4) shaft absolute motion.

Eddy current — Electrical current that is generated (and dissipated) in a conductive material when such material intercepts the electromagnetic field of a proximity probe.

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

Failure cause — Degradation that results in 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.

Frequency — The repetition rate of a periodic wave within a unit time. This is normally expressed in units of revolutions per minute (rpm), cycles per minute (cpm), events per minute (epm), or cycles per second (cps or Hz). With respect to rotating machinery vibrations, two types of frequencies are of interest: (1) the shaft rotational frequency and (2) the various vibration frequencies (as measured by vibration transducers). Rotational frequencies are commonly expressed in terms of cpm, cps, or Hz or in terms of the shaft rotational frequency: 1X means one times rotational frequency, 2X means two times, 1/2X means one-half times, etc. Vibration frequency can also be expressed as a percentage of rotational frequency.

Frequency response — The amplitude and phase characteristics of a mechanical or electronic system with respect to frequency.

Harmonic — Sinusoidal quantity at a frequency that is an integer multiple of the fundamental frequency.

Hertz (Hz) — Unit of frequency measurement represented by cycles per second.

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.

Imbalance (unbalance) — Unequal radial weight distribution on a rotor system, a shaft condition such that the mass centerline (principal inertial axis) does not coincide with the geometric centerline. Also, the effective mass causing the rotor to be out of balance.

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.

Mechanical runout — A source of error on the output signal of a proximity probe transducer system; a probe gap change that does not result from either a shaft centerline position change or shaft dynamic motion. Common sources include out-of-round shafts, scratches, chain marks, dents, rust or other conductive buildup on the shaft, stencil marks, flat spots, and engravings.

Natural frequency — The frequency of free vibration of a system. The frequency at which an undamped system with a single degree of freedom will oscillate upon momentary displacement from its rest position by a transient force. The natural frequencies of multiple degree of freedom systems are the frequencies of the normal modes of vibration.

Noise — Any component of a transducer output signal that does not represent the variable intended to be measured.

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.

Oil whirl/whip — An unstable vibration of the free vibration category whereby the bearing has insufficient unit loading. This can occur because of various mechanisms, including excessive radial bearing clearance and a steady-state preload acting in the opposite direction to the bearing load. Under this malfunction condition, the shaft centerline dynamic motion is usually circular and forward (in the same direction as shaft rotation). Oil whirl vibration occurs at a frequency equal to average oil flow velocity within the bearing (i.e., 40 to 49% of shaft rotative speed). Oil whip occurs when the oil whirl frequency coincides with (and becomes locked to) a system natural resonance, often a rotor balance resonance. Note that the whirl/whip mechanism is not restricted to oil-lubricated bearings but can occur in any case where a fluid is between two cylindrical surfaces, for example, system whirl or whip in a steam seal.

Operational readiness — The capability of a pump to fulfill its function.

Orbit — The path of the shaft centerline motion during rotation. The orbit is observed with an oscilloscope connected to XY proximity probes. An orbit is sometimes referred to as a Lissajous pattern.

Peak-to-peak value — The difference between positive and negative extreme values of a signal or dynamic motion.

Power intensity — A relative measure of a pump's brake horsepower per stage. Mathematically, the power intensity factor is defined as

$$FPI = \frac{BHP}{D^3} ,$$

where BHP is the power (BHP) per stage and D is the impeller diameter (in.).

Predictive maintenance — Those maintenance activities involving the measure of parameters that indicate the performance or condition of a machine or component. Parameter changes from the norm can be used to predict and prevent failure.

Preventive maintenance — Those maintenance activities (e.g., schedules, lubrication, visual inspection, etc.) that involve preventive measures for the purpose of limiting facility outages.

Probe — A generic term that is used to describe many types of sensors.

Proximity probe — A noncontacting device that measures the displacement motion or position of an observed surface relative to its mounting location. Typically, proximity probes used for rotating machinery measurements operate on the eddy current principle and measure shaft displacement motion and position relative to the machine bearing(s) or casing.

Proximate cause — Final cause contributing to failure.

Radial — A direction on a machine that is perpendicular to the shaft centerline.

Radial position — The average location, relative to the RADIAL bearing centerline, of the shaft dynamic motion. This can be determined by evaluating the dc output signals of XY relative proximity probes.

Radial vibration — Shaft dynamic motion or casing vibration that is in a direction perpendicular to the shaft centerline.

Real time analyzer — A term used to describe an instrument that displays a vibration frequency spectrum. See Spectrum Display Unit.

Repeatability — The ability of a transducer or readout instrument to reproduce output readings when the same value is applied to it repeatedly, under the same conditions, and in the same direction. Also, the maximum deviation from the mean of corresponding data points taken from repeated tests under identical conditions. The word accuracy is often used incorrectly as a synonym for repeatability.

Resolution — The smallest change in applied stimulus that will produce a detectable change in the instrument output. Resolution differs from precision because it is a psychophysical term referring to the smallest increment of humanly perceptible output (rated in terms of the corresponding increment of input).

Resonance — The condition of vibration amplitude and phase change response caused by a corresponding system sensitivity to a particular forcing frequency. A resonance is typically identified by a substantial amplitude increase and related phase shift.

Seismic transducer — A transducer that is mounted on the case or housing of a machine and measures casing vibration relative to free space. An accelerometer is a seismic transducer that converts acceleration motion and/or gravitational forces capable of imparting acceleration into a proportional electrical signal. A velocity transducer is a seismic transducer that converts velocity motion into a proportional electrical signal.

Signal — An electrical transmittance (either input or output) that conveys information.

Sensitivity — The magnitude of the change in an output signal to a known change in the value of the measured variable. Also called scale factor.

Signature — The term usually applied to the vibration frequency spectrum that is distinctive and special to a particular machine or component, system, or subsystem at a specific point in time, under specific machine operating conditions, etc. Used for historical comparison of mechanical condition over the operating life of the machine.

Span — The difference between the upper and lower limits of a range expressed in the same units as the range.

Specific speed — The speed at which an impeller, geometrically similar to the one under consideration, would run if it were reduced in size to deliver 1 gal/min at 1-ft head. Mathematically, it is expressed as

$$N_S = \frac{N\sqrt{Q}}{H^{3/4}} ,$$

where N is the pump speed (rpm), Q is the design capacity (gal/min) at the BEP, and H is the total head (ft) per stage at the BEP.

Spectrum — Presentation of the amplitude of a signal as a function of frequency.

Spectrum display unit — An instrument that displays an XY presentation of vibration frequency (X) vs vibration amplitude (Y). This presentation is called a vibration frequency spectrum and is usually presented on a CRT, with hard-copy capability, plotter outputs, and/or computer interface connections.

Spectrum plot — An XY plot where the X axis represents vibration frequency and the Y axis represents vibration amplitude.

Stiffness — The springlike ability of mechanical and hydraulic elements to elastically deform under load (resistance to deformation). This concept is applicable to shafts, bearings, cases, and support structures. The units normally used are pounds per inch or newtons per meter.

Subharmonic — Sinusoidal quantity of a frequency that is an integral submultiple of a fundamental frequency.

Subsynchronous — Component(s) of a vibration signal that has a frequency less than the shaft rotative frequency.

Suction specific speed — Analogous to specific speed except that here it is essentially an index descriptive of the suction capabilities and characteristics of a given first-stage impeller. Mathematically, it is expressed as

$$S = \frac{N\sqrt{Q}}{(NPSHR)^{3/4}} ,$$

where

N = pump speed (rpm)
Q = design capacity (gal/min) at BEP for single-suction first-stage impellers or one-half the design capacity for double-suction first-stage impellers (at maximum diameter),
NPSHR = net positive suction head required (ft) (at BEP).

Supersynchronous — Component(s) of a vibration signal that has a frequency greater than the shaft rotative frequency.

Synchronous — Vibration frequency components that vary in direct proportion to changes in rotative frequency. Typically, but not always, synchronous components are whole-multiples, or integer fractions of rotative speed, and maintain that relationship regardless of speed (e.g., 1X, 2X, 3X, 1/2X, etc.).

Thermocouple — A temperature-sensing device comprised of two dissimilar metal wires that when thermally affected (heated or cooled), produce a proportional change in electrical potential at the point where they join.

Thermocouple type (ANSI symbol)	Material
J	Iron/Constantan
K	Chromel/Alumel
T	Copper/Constantan
E	Chromel/Constantan
R	Platinum/platinum 13% rhodium
S	Platinum/platinum 10% rhodium
B	Platinum 6% rhodium/platinum 30% rhodium
G*	Tungsten/tungsten 26% rhenium
C*	Tungsten 5% rhenium/tungsten 26% rhenium
D*	Tungsten 3% rhenium/tungsten 25% rhenium

*Not ANSI symbols.

Thermowell — A closed-end tube designed to protect temperature sensors from harsh environments, high pressure, and flows. They can be installed into a system by pipe thread or welded flange and are usually made of corrosion-resistant metal or ceramic material, depending upon the application.

Transmitter (two-wire) — A device that is used to transmit data via a two-wire current loop. The loop has an external power supply, and the transmitter acts as a variable resistor with respect to its input signal.

Ungrounded junction — A form of construction of a thermocouple probe where the hot or measuring junction is fully enclosed by and insulated from the sheath material.

Vane passing frequencies — A potential vibration frequency on vaned impeller compressors and other machines with vaned rotating elements. It is represented by the number of vanes (on an impeller or stage) times the shaft rotative frequency.

Velocity — The time rate of change of displacement. This is often expressed as V, $\dot{x}$, or dx/dt; velocity leads displacement by 90° in time. Typical units for velocity are inches per second or millimeters per second, zero-to-peak. Velocity measurements are usually obtained with a mechanically activated velocity transducer and are used to evaluate casing response characteristics. Electronic integration of an acceleration signal will provide a velocity signal.

1X — Notation for the signal component in a complex vibration signal that occurs at the rotational speed frequency. Also called synchronous.

2X, 3X, etc. — tation for the component(s) in a complex vibration signal having a frequency equal to an exact multiple of the rotative spee Also called harmonic, superharmonic, and supersynchronous.

XY — Perpendicular axes in a Cartesian coordinate system. Usually used as a reference for orthogonal (mutually perpendicular) radial vibration transducers.

Appendix B

OPERATING EXPERIENCE DATA BASES AND REPORTS

Failure data from LER, NPRDS, and various other data bases are summarized below. Data sources on AUXFP failures formed the basis of this report. Other data were provided as a supplement. Only data pertaining specifically to the pump are cited (i.e., drivers, etc., excluded).

Table B.1, AUXFP LER data (1973–83), is a reprint from Vol. 1 of this report. Refer to Vol. 1 for details.

Table B.2, AUXFP LER data (1984–86), is a supplement to Table B.1 and provides data from 1984–86. There were four additional events recorded in 1984–86.

Table B.3, AUXFP NPRDS data (1974–85), is a reprint from Vol. 1 of this report. Refer to Vol. 1 for details.

Table B.4, AUXFP NPRDS data (1986), is a supplement to Table B.3 and provides data from 1986. Only seven additional events were recorded in 1986.

Table B.5, HHSIP LER data (1973–86), contains supplementary data for the HHSIPs. These data represent HHSIPs (also called charging pumps in some plants) in addition to normal HHSIPs. Both pump types are similar in design to the AUXFP. Functionally, the charging pumps can be considered as continuously operating, while the HHSIPs are intermittent-duty pumps. There were 68 total events reported.

Table B.6, HHSIP NPRDS data (1974–85), contains supplementary data for the HHSIPs. There were 53 total events reported.

Table B.7, HHSIP NPRDS data (1974–85), (N < 3600 rpm), contains data that are derived from the same data source as that reported in Table B.6. The pumps that operate at <3600 rpm are considered the intermittent-duty pumps, which are functionally similar to the AUXFP. There were 27 total events reported.

Table B.8, Major outage producing failure causes for feed pumps, contains data that were derived from EPRI Report FP-754, entitled *Survey of Feed Pump Outages*,[1] and represents fossil boiler feed pumps, booster pumps, nuclear steam generator feed pumps, and booster pumps.

Table B.9, Typical failure causes, centrifugal pumps (domestic plants), contains data that represent petrochemical plant pump failure data and includes single-stage, multistage, vertical, and horizontal pumps.

Table B.10, Typical failure causes, centrifugal pumps (foreign plants), contains data that represent failure causes for four foreign petrochemical plants and would also include single-stage, multistage, vertical, and horizontal pumps.

Reference

1. E. Makay and O. Szamody, *Survey of Feed Pump Outages*, EPRI FP-754, Electric Power Research Institute, April 1978.

Table B.1. Summary of AUXFP-type failures reported in LERs

(1973–1983)

Item	Rate[a] (%)
Failed component	
Bearings	48
Packing and seals	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.

Source: M. L. Adams and E. Makay, *Aging and Service Wear of Auxiliary Feedwater Pumps for PWR Nuclear Power Plants, Volume 1, Operating Experience and Failure Identification*, NUREG/CR-4597, Volume 1 (ORNL-6282/V1), Oak Ridge Natl. Lab., July 1986.

Table B.2. Summary of AUXFP-type failures reported in LERs

(1984–1986)

Item	Occurrences	Rate (%)
Failed component		
Bearings	1	25
Packing and seals	0	0
Casing	1	25
Internal components	1	25
Impeller	0	0
Capacity	0	0
Shaft	0	0
Other	1	25
Total	4	100
Methods of detection		
Testing	2	50
Operation	2	50
Maintenance	0	0
Not stated	0	0
Total	4	100
Maintenance action		
Replacement	2	50
Repair	1	25
Modification	0	0
No repair required	1	25
Total	4	100
Identified cause		
Lack of lubrication or cooling	1	25
Maintenance error	0	0
Wear/end of life	0	0
Design error	1	25
Crud	0	0
Operator error	0	0
Other	2	50
Not stated	0	0
Unknown	0	0
Total	4	100

Table B.3. 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.

Source: M. L. Adams and E. Makay, *Aging and Service Wear of Auxiliary Feedwater Pumps for PWR Nuclear Power Plants, Volume 1, Operating Experience and Failure Identification*, NUREG/CR-4597, Volume 1 (ORNL-6282/V1), Oak Ridge Natl. Lab., July 1986.

Table B.4. Summary of AUXFP-type failures reported in NPRDS data base

(1986)

Item	Occurrences	Rate (%)
Failed component		
Packing/gasket	4	58
Bearings	3	42
Internal components	0	0
Shaft	0	0
Total	7	100
Methods of detection		
Incidental observation	0	0
Surveillance testing	1	14
Routine observation	4	58
Audiovisual alarm	1	14
Operational abnormality	0	0
Special inspection	1	14
Total	7	100
Maintenance action		
Repair/replace	7	100
Modify	0	0
Total	7	100
Failure cause		
Wear	6	86
Lubrication	1	14
Binding	0	0
Aging	0	0
Abnormal stress	0	0
Total	7	100

Table B.5. Summary of HHSIP failures reported in LERs (1973–1986)

Item	Occurrences	Rate (%)
Failed component		
Bearings	5	8
Packing and seal	8	12
Casing	2	3
Internal components	22	32
Impeller	6	9
Capacity	7	10
Shaft	7	10
Other	11	16
Total	68	100
Methods of detection		
Testing	30	44
Operation	22	32
Maintenance	6	9
Not stated	10	15
Total	68	100
Maintenance action		
Replacement	23	34
Repair	13	19
Modification	2	3
No repair required	10	15
No data available	20	29
Total	68	100
Identified cause		
Lack of lubrication or cooling	5	8
Maintenance error	3	4
Wear/end of life	22	32
Design error	14	21
Crud	1	2
Operator error	7	10
Other	9	13
Not stated	3	4
Unknown	4	6
Total	68	100

Table B.6. Summary of HHSIP failures reported in NPRDS (1974–1985)

Item	Occurrences	Rate (%)
Failed component		
Packing/gasket	25	47
Bearings	10	19
Internal components	16	30
Shaft	2	4
Total	53	100
Methods of detection		
Incidental observation	1	2
Surveillance testing	11	21
Routine observation	36	68
Audiovisual alarm	0	0
Operational abnormality	3	5
Special inspection	2	4
Total	53	100
Maintenance action		
Repair/replace	51	96
Modify	2	4
Total	53	100
Failure cause		
Wear	33	63
Lubrication	3	5
Binding	4	8
Aging	3	5
Abnormal stress	10	19
Total	53	100

Table B.7. HHSIP failures reported in NPRDS data base for <3600 rpm

(1974–1985)

Item	Occurrences	Rate (%)
Failed component		
Packing/gasket	12	44
Bearings	5	19
Internal components	10	37
Shaft	0	0
Total	27	100
Methods of detection		
Incidental observation	0	0
Surveillance testing	7	26
Routine observation	14	52
Audiovisual alarm	1	4
Operational abnormality	3	11
Special inspection	2	7
Total	27	100
Maintenance action		
Repair/replace	26	96
Modify	1	4
Total	27	100
Failure cause		
Wear	17	63
Lubrication	4	15
Binding	2	7
Aging	0	0
Abnormal stress	4	15
Total	27	100

Table B.8. Ten major outage-producing failure causes for feed pumps

No.	Pump failures: component, symptom, or technology
1	Seals
2	Vibration: pump, piping, foundation
3	Axial balancing device
4	Journal bearing
5	Cavitation
6	Impeller breakage
7	Wear-ring: rapid wear
8	Unstable head curve
9	Shaft broken/damaged
10	Thrust bearing

Source: Reprinted with permission from E. Makay and O. Szamody, *Survey of Feed Pump Outages*, EPRI FP-754, Electric Power Research Institute, April 1978.

Table B.9. Typical failure causes reported for U.S. petrochemical plants

(Centrifugal pumps)

Failure cause	Ranked by frequency of occurrence
Shaft seal deterioration, breakage	1
Bearing wear, corrosion, breakage	2
Shaft failures	3
Binding between rotor and stationary parts	4
Coupling wear, breakage	5
Impeller wear, breakage	6
Wear surface wear, erosion, corrosion, seizing	7
Casing leakage	8

Source: Reprinted with permission from H. P. Bloch, *Practical Machinery Management for Process Plants, Vol. 1, Machinery Reliability*, Gulf Publishing Company, 1982.

Table B.10. Typical failure causes reported for several foreign petrochemical plants

(Centrifugal pumps)

Failure cause	Ranked by frequency of occurrence
Shaft seal deterioration, breakage	1
Bearing wear, corrosion, breakage	2
Impeller wear, breakage	3
Shaft failures	4
Casing and internal structural damage	5
Couplings	6

Source: Reprinted with permission from H. P. Bloch, *Practical Machinery Management for Process Plants, Vol. 1, Machinery Reliability*, Gulf Publishing Company, 1982.

Appendix C

INSTALLATION LIST

Table C.1. Installation list, Ingersoll-Rand

Installation	Pump size	Qty.	Capacity (gal/min)	Head (ft)	Speed (rpm)	Temp. (°F)
Indian Point	3HMTA-9	2	400	3118	3570	120
Surry	4HMTA-6	1	735	2726	4200	120
Surry	3HMTA-8	2	370	2726	3560	120
Surry	4HMTA-6	1	735	1716	4200	120
Surry	3HMTA-9	2	370	2726	3560	120
Indian Point	3HMTA-9	2	400	3118	3570	120
Beaver Valley	4HMTA-6	1	730	2691	4200	120
Main Yankee	3HMTA-5	1	530	2529	4400	120
Donald C. Cook	4HMTA-6	2	900	1137	4350	100
Donald C. Cook	3HMTA-8	2	450	1137	3560	100
Beaver Valley	3HMTA-8	2	370	2691	3560	120
Main Yankee	3HMTA-8	1	500	2529	3575	120
Arkansas Nucl. One	4HMTA-9	2	780	2569	3560	120
Arkansas Nucl. One	5HMTA-4	1	1150	2340	3560	120
Three Mile Island	3HMTA-8	3	470	2553	3560	100
Three Mile Island	4HMTA-6	1	940	2553	4250	100
Sequoyah	5HMTA-5	2	920	2529	3950	120
Sequoyah	3HMTA-9	4	440	2834	3570	120
Millstone	4HMTA-5	1	600	3656	4200	100
Millstone	2HMTA-8	2	300	2962	3560	100
North Anna	4HMTA-6	2	735	2815	4200	100
North Anna	3HMTA-8	4	370	2815	3560	100
Farley	4HMTA-7	2	700	3641	3960	95
Farley	2HMTA-10	4	350	3641	3560	95
Watts Bar	5HMTA-6	2	1000	2927	3850	120
Watts Bar	3HMTA-9	4	500	2927	3577	120
Shearon Harris	4X9N-7	4	850	2934	4000	125
Shearon Harris	3HMTA-9	8	425	2934	3500	125
Ginna	2HMTA-10	2	200	3354	3560	120
Commanche Peak	5HMTA-6	2	1140	3158	4050	110
Commanche Peak	4HMTA-9	4	570	3158	3560	110
ASCO (Spain)	4X9NH-9	2	840	3338	4000	125
ASCO (Spain)	2HMTA-10	4	240	3438	4000	125
Callaway	6HMTA-6	5	1060	3405	3800	95
Wolf Creek	4HMTA-9	10	535	3405	3560	95
WNP	6HMTA-6	2	1400	3128	3850	90
WNP	4X9NH-10	4	600	3197	3850	90
KRSKO (Yugoslavia)	4X9N-9	1	810	3398	3900	100
KRSKO (Yugoslavia)	2HMTAM-10	2	370	3354	4000	100
St. Lucie	4HMTA-7	1	540	N/A	3725	120
St. Lucie	2HMTA-10	2	300	N/A	3590	120
Georgia Power	6HMTA-5	2	1155	3169	3950	150
Georgia Power	4HMTA-9	4	510	3169	3960	150

Table C.2. Installation list, Bingham

Installation	Pump size	Quantity
McGuire	3X4X9E MSD	4
McGuire	4X6X10B MSD	2
North Anna	3X6X9C MSD	1
North Anna	3X6X9C MSD	2
North Anna	3X6X9C MSD	1
North Anna	3X6X9C MSD	2
Beaver Valley	3X6X9C MSD	2
Beaver Valley	3X6X9C MSD	1
Millstone	3X6X9C MSD	2
Millstone	4X6X10 MSD-D	1
Catawba	3X6X9E MSD	4
Catawba	4X8X10-1/2B MSD-D	2
Midland	4X8X10-1[illegible]D-D	2
Midland	4X8X1[illegible]D	2
Bellefonte	4X8[illegible]	2
Bellefonte	3X6[illegible]	4
South Texas	4X6X[illegible]	8

Table C.3. Installation list, Pacific Pumps

Installation	Pump size	Capacity (gal/min)	Head (ft)
Keewaunee	1 1/2-in. UNI	240	2850
Prairie Island	1 1/2-in. UNI	220	2920
Zion	3-in. JTC	495	3099
Zion	4-in. JTCX	990	3099
Braidwood	4-in. SRBF1DS	990	3350
Byron	4-in. SRBFIDS	990	3350

Table C.4. Installation list, Byron-Jackson

Installation	Pump size	Qty.	Capacity (gal/min)	Head (ft)	hp	Speed (rpm)	Temp. (°F)
Diablo Canyon	3X6X9E DVMX	2	490	3000	500	3510	100
Diablo Canyon	4X6X9D DVMX	1	930	3000	900	4000	100
Diablo Canyon	3X6X9D DVMX	2	490	3000	500	3570	100
Diablo Canyon	4X6X9D DVMX	1	930	3000	900	4000	100
Calvert Cliffs	3X6C9E DVMX	2	700	2490	600	3990	100
Hutchison Island	3X6X9B DVMX	2	325	2660	350	3570	120
Hutchison Island	3X6X9B DVMX	1	325	2660	350	3570	120
Arkansas	3X6X9B DVMX	2	575	2800	600	3575	100
Davis-Besse	4X6X9D DVMX	2	1050	2500	800	3600	100
Almaraz	4X6X9D DVMX	4	450	3200	1000	2960	100
Almaraz	4X6X9D DVMX	2	900	3200	650	3900	100
San Onofre	4X6X9D DVMX	1	860	2842	800	3570	100
San Onofre	4X6X9D DVMX	1	860	2842	800	3570	100
San Onofre	4X6X9D DVMX	1	860	2842	800	3570	100
San Onofre	4X6X9D DVMX	1	860	2842	800	3570	100

Appendix D

LOW-FLOW TESTING

AUXFPs are considered "active" safety-related emergency pumps. The plant technical specification stipulates that these pumps must be tested monthly to meet hydraulic and mechanical acceptance criteria. AUXFPs typically have a design-developed head equal to steam generator safety valve set pressure plus system line losses. Surveillance testing or any operation of these pumps while the plant is at power requires special precautions to protect against interference with normal operation.

Periodic surveillance testing is now performed on the pump while pumping from the condensate storage tank through the miniflow bypass line (which normally contains a miniflow orifice and isolation valves) and returning to the condensate storage tank. Appendix E, Fig. E.2 shows a typical AFW flow diagram for a W PWR plant. The minimum flow for these pumps was typically established only to prevent pump overheating and thus is normally 10 to 15% of BEP flow. (For purposes of this discussion, "low flow" is defined as those flows equal to ~10–15% of pump BEP flow.)

The pump miniflow circuit provides a convenient flow path through which the pump can be loaded and testing can be accomplished. Currently, ASME Sect. XI, Subsect. IWP, requires that the pump performance baseline parameters be established first. The baseline measurements are very critical because pump acceptance criteria depend on performance changes from the baseline, although the minimum acceptable developed head must always be greater than an absolute minimum established curve, which is normally included in the plant technical specifications. Testing frequency is now once per month. Table IWP-3100-2, Subsect. IWP, provides parameter ranges that must be maintained to satisfy pump operational readiness. Should the pump performance fall within the "alert range," per Table IWP-3100-2, normal test frequency is doubled. 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. Correction shall be either replacement or repair 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. IWP-3111 discusses reference value determinations following replacement or repair actions.

Experience gained in both the laboratory and the field during the past decade has shown that instabilities and unbalances can occur in pumps at low flows. There are reported cases of pump vibration and unusual flow pulsation frequencies and amplitudes occurring during low-flow testing. Vibration-induced damage to pumps and valves has been known to occur during extensive operation of pumps at low flow.

Pump vibration is typically highest when the pump is operated at low flow, as in the case of surveillance testing. On a new or refurbished machine, that is, a pump with tight wear-ring clearances, new bearings, etc., the effects on vibration caused by the hydraulic instabilities would be lower than for worn pumps. However, as the pump ages and the critical clearances increase, the damping effect of these clearances is reduced, and the effect on vibration caused by these instabilities becomes more pronounced and measurable. This could result in prematurely declaring the pump to be in the alert or required action range. Operating the pump at higher flows (e.g., >25% BEP flow) would reduce the effect of hydraulic instability on pump performance, thus making the test data more meaningful because, in reality, the pump is not intended to operate at such low flows to perform its normal plant design function. Reducing the vibration levels by operating at higher flows will also reduce the longer-term aging and wear effects on such parts as wear rings, bearings, seals, etc.

The increase in internal clearances, such as wear-ring and balance drum clearances that occur through pump aging, results in increased stage-to-stage leakage, which reduces pump hydraulic efficiency. Figure 2.7 shows the effect of wear on the head-capacity curve of a centrifugal pump. At any given head, the net capacity is reduced by the increase in leakage. The leakage is proportional to the square root of the stage-to-stage pressure differential. Note that in the low-flow portion of the curve the change in head, after wear has taken place, is small compared with the larger change taking place near expected BEP or runout flow. Because, in reality, these pumps are to perform their function at the higher flows, the measurement of head change at low flow will not provide an accurate indication of the actual influence of wear and degradation.

Periodic testing at low flows is now conducted to detect gross pump degradation such as that caused by impeller structural damage or other hydraulic component failure. Low-flow testing will not provide a good indication of gradual pump performance deterioration that can be expected to result from the normal wear and aging process. Full-flow testing will more accurately detect these gradual changes.

In summary, obtaining test data at higher flows results in more consistent and dependable data for determining and trending pump wear. In addition, aging and service wear rates should be reduced.

Appendix E

TYPICAL AUXILIARY FEEDWATER SYSTEM DESCRIPTION FOR WESTINGHOUSE PWR PLANT

Function

The AFW system serves as a backup system to supply feedwater to the secondary side of the steam generators whenever the normal main feedwater system is not available in order to maintain the steam generator as the principal reactor shutdown heat sink. This system also functions as an alternate to the main feedwater system during start-up, hot standby, and cooldown. As an engineered safeguards system, the AFW system is directly relied upon to provide core cooling during off-normal or emergency transients such as loss of normal feedwater or secondary system piping failure. Thus, feedwater is continuously supplied to the steam generators following main feedwater isolation or main feedwater pump turbine trip. Figure E.1 is a simple schematic of the system, while Fig. E.2 is a more detailed schematic.

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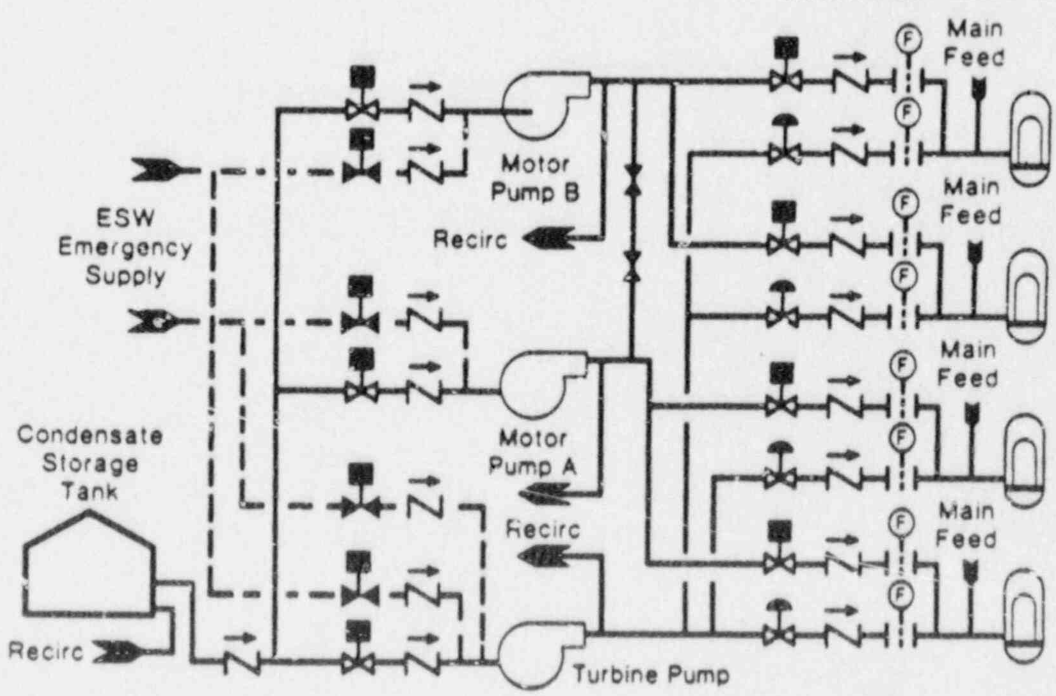

Fig. E.1. AFW system simplified schematic.

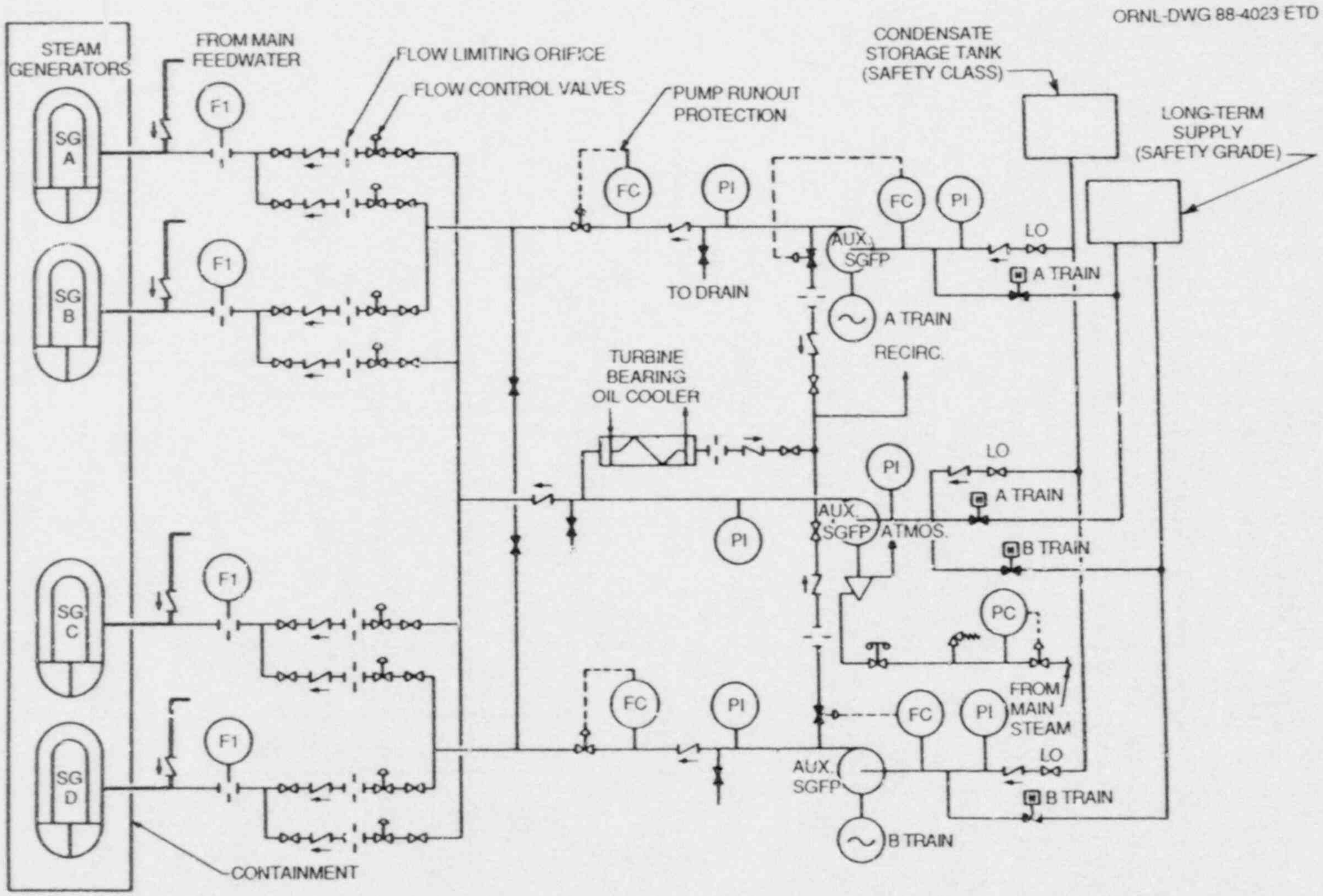

Fig. E.2. Flow diagram (typical) AFW system four-loop plant.

Design Basis

The AFW system is designed to meet the cooling requirements for the following emergency conditions:

1. loss of main feedwater system,
2. loss of site electric power,
3. steamline rupture,
4. feedline rupture,
5. control room evacuation, and
6. loss of coolant accident.

To meet these requirements, the systems are to incorporate the following recommended features:

1. Two sources of cooling water.
2. Two 100% capacity trains of electric-motor-driven pumps, etc.
3. A third 200% capacity turbine-driven pump.
4. Only two of the four steam generators are needed for a safe cooldown.
5. Alternate and redundant power supplies, control systems, and instrumentation and piping systems are provided to meet the design requirements for engineered safety features.
6. The system must perform adequately with a single active or passive failure.

Minimum flow requirements for safe shutdown require at least one motor-operated AUXFP to establish hot shutdown following any accident or transient condition. The AFW system then provides adequate cooling capacity to cool the plant from hot standby to a reactor coolant hot leg temperature of 350°F. This temperature will permit extended cooling using the residual heat removal system. A cooldown from hot standby to 350°F can be achieved in ~5 h.

Automatic start-up of the system following initiation of the auxiliary feedwater actuation signal will deliver water to the steam generator within 1 min.

System Description

General description (Fig. E.1)

The AFW system consists of two motor-driven pumps, one steam-turbine-driven pump, and associated piping, valves, instruments, and controls. Each pump can deliver water to all four steam generators. Normal flow is from the condensate storage tank containing feedwater quality water, through the motor-operated isolation valves, to the AUXFPs. Backup sources typically include the safety class essential service water (ESW) system. Flow rate to the steam generators is controlled by pneumatic diaphragm or motor-operated flow control valves downstream of pump discharges. Pumps start automatically on sensed loss of heat sink events.

The system is located within the auxiliary building, with adequate missile barrier protection provided.

Motor-driven pumps

The two motor-driven AUXFPs are multistage horizontal centrifugal pumps. They are powered by two separate 3600-rpm, 60-Hz motors that draw power from different busses. Emergency power is from physically separate emergency diesel generator busses. Maximum pump capacity, not including recirculation flow, is typically about 500 gal/min, at a maximum developed head of 3200 ft.

Turbine-driven pump

The one steam-driven AUXFP is also a multistage horizontal centrifugal pump. Maximum pump capacity, not including recirculation flow, is typically about 1000 gal/min at a maximum developed head of 3450 ft. The steam is supplied from main steam loops B and C upstream of the main steam isolation valves. The steam supply lines contain isolation valves and bypass valves. The turbine is designed to operate over a wide range of steam pressures, ~100 to ~1200 psig at a maximum steam temperature of 580°F. The turbine is controlled by two independent, redundant electrical-hydraulic governors, either of which may control the turbine. An emergency overspeed governor shuts off the main steam supply by closing the trip/throttle valve at 110% rated speed. The turbine lube oil system is cooled by auxiliary feedwater from the first stage of the pump. The turbine exhausts to the atmosphere. Auxiliary steam can be lined up to the turbine but is used only for testing.

Auxiliary feedwater supply

Normal supply. Normal supply is from a 450,000-gal stainless steel condensate storage tank containing feedwater-grade makeup water. Water temperature is maintained at least 50°F by steam heating coils. Minimum required volume is 200,000 gal. The flow by gravity from the tank through a common supply header divides into a separate supply line to each AUXFP. Flow is controlled through motor-operated isolation valves.

Emergency supply. Emergency supply is from two ESW system headers. The first ESW header divides into a separate supply line to Motor Pump A and to the turbine pump through two motor-operated isolation valves. The second ESW header divides into a separate supply line to Motor Pump B and to the turbine pump through two motor-operated isolation valves.

Auxiliary feedwater pump discharges

The turbine pump discharge divides into two headers. Each header further branches into two feed lines, one going to each steam generator. Flow is controlled through pneumatic diaphragm valves. Downstream of the flow control valves, each line joins with a feed line from the associated motor-driven pump into a single auxiliary feed header, which connects into each main feed line just upstream of containment penetration.

Motor Pump A discharge divides into two lines that supply the common auxiliary feed header to steam generators B and C through motor-operated flow control valves. Motor Pump B similarly supplies steam generators A and D through motor-operated valve flow control valves, respectively.

The motor-driven pump discharges may be cross-connected through a line normally shut by locked-shut isolation valves.

Continuous recirculation from each AUXFP to the condensate storage tank is maintained for pump protection. The separate recirculation lines combine into a single return. Manual isolation valves in each individual recirculation line are normally open.

Downstream of each supply source, pump, and flow control valve are nonreturn valves and manual isolation valves that prevent backflow and allow for system isolation. Each pump and motor-operated/pneumatic isolation or control valve may be operated manually at a local station.

Instrumentation and control

Instrumentation is provided that informs the operator in the control room of the state of the system. Instrumentation is also provided locally, which enables operation of the system in the event of evacuation of the control room.

Alarms are also provided in the control room for minimum level in the condensate tank, low suction pressure in both condensate and alternate suction lines, low turbine stop valve steam pressure, and an alarm annunciator that indicates when the remote control is overridden by local control.

In addition to the manual actuation of the AUXFPs, the following signals provide for automatic actuation:

1. Motor-driven AUXFP automatic actuation signals:

   A. 2/4 low-low level in any one steam generator,
   B. trip of all main feedwater pumps,
   C. blackout sequence, and
   D. safeguards sequence (initiated by safety injection signal).

2. Turbine-driven AUXFP automatic actuation signals:

   A. 2/4 low-low level in any two steam generators, and
   B. 2/4 undervoltage reactor coolant pump busses (loss of off-site power).

Whenever any AUXFP starts, the steam generator blowdown and sampling systems are automatically isolated.

The minimum number of pumps must be supplied with power and the required flow to the steam generators must be established automatically under a loss of station power or a loss of normal feed system.

System Operation

Normal conditions

Plant startup. During a normal startup, the AFW system can be used in manual control to maintain steam generator water level until sufficient steam is generated to allow startup of the steam-driven main feed pumps. Suction is from the condensate storage tank.

Power operation. The system is not used in normal operation. The AUXFPs may be tested by pumping into main feed lines while at power.

Plant cooldown. During normal plant cooldown, the system is used to supply feedwater to the steam generators for decay heat removal and reactor coolant system cooldown. The rate of auxiliary feedwater flow is regulated from the main control room to maintain steam generator level. After the primary coolant temperature reaches 350°F, the residual heat removal system is placed in operation, and the AFW system is no longer required. Suction is taken from the condensate storage tank.

Abnormal conditions. The system is normally lined up for operation with the exception of AUXFPs. Upon emergency start of the AUXFPs, feedwater from the condensate storage tank will be automatically pumped into steam generators. Whenever the auxiliary feedwater actuation signal (AFAS) is initiated, the steam generator blowdown and sample lines are automatically isolated. These blowdown and sample isolation valves may not be opened until the AFAS reset pushbutton is reset. If loss of normal vital power supply has also occurred, the motor-driven AUXFPs will be powered by separate emergency diesel generators and will be started in accordance with the engineered safety feature safeguards timing sequence to prevent overloading the emergency diesels. In case of failure or loss of condensate storage tank water supply, the normally closed ESW supply isolation valves will automatically open.

In the event of a main feed line break, flow restrictors in the AUXFP discharge lines will sufficiently limit flow to the break so that cooldown flow is maintained to other steam generators for removal of reactor decay heat. In the event of a main steam line break inside containment, the flow restrictors will sufficiently limit flow to the faulted steam generator to keep containment pressure below the design limit.

Adequate control functions and indications are available at the auxiliary shutdown panel and the local component control stations to operate the AFW system upon loss of the main control room. An undesirable valve position or pump status selected on the main control panels may be overridden by local/remote control switching and duplicate selector switches on the auxiliary shutdown panel.

Appendix F

AUXFP MINIMUM-FLOW RATE CRITERIA

Many technical articles, textbook discussions, and research studies have been completed in an attempt to understand and develop specification criteria for minimum pump flow. Traditionally, minimum flow was established on the basis of pumped fluid temperature rise. Pumped fluid temperature rise is a result of the hydraulic inefficiency of the pump that occurs when shaft power is converted to water power. The temperature rise in a pump with water flowing through it is

$$\Delta T = \frac{H(1 - \eta)}{778\eta} ,$$

where ΔT is the temperature rise (°F/min), η is the pump hydraulic efficiency, and H is the pump head (ft).

Near the shutoff point, the efficiency of a centrifugal pump is almost zero, and most of the energy will go into heating the liquid in the pump. Unless a minimum flow passes through the pump to carry away the heat, overheating will cause liquids with low vapor pressure to boil in the low-pressure regions of the pump and will cause the pump rotating element to seize. A general rule of thumb to establish minimum continuous flow, based on ΔT, is to limit temperature rise to 15°F/min.

However, currently it is generally understood by pump manufacturers and users that temperature rise is only one concern. Centrifugal pumps will demonstrate a condition of inlet-flow breakdown at some point on their characteristic curve. This breakdown point results in a mixed flow condition at the impeller eye, with flow simultaneously moving in and out of the inlet. This condition has been described in a variety of ways such as backflow, stall, and inlet recirculation. Volume 1, Appendix D, of this report provides a detailed description (with illustrations) of this phenomenon.

Vane separation and recirculation can be expected to occur at the impeller exit and inlet when a pump is operating at flow rates considerably below the BEP. The percentage of design-point flow rate at which recirculation will commence is dependent on many factors, including the following:

1. impeller to volute or diffuser match,
2. inlet vane angles and position of the leading edge,
3. impeller radius ratio,
4. shaft diameter ratio,
5. power intensity,
6. specific speed,
7. suction specific speed, and
8. net positive suction head (NPSH) available margin.

Simplifying matters, it is widely believed that the most significant factors affecting AUXFPs are power intensity and suction specific speed.

Most of the guidelines that have been published stipulating recommended minimum flow for pumps present minimum flow rates (or percentage of BEP flow rate) as a function of specific speed. Guidelines from references are included [Figs. F.1 (Ref. 1), F.2 (Ref. 2), and F.3 (Ref. 2)]. Figures F.1 and F.2 present minimum flow as a function of specific speed and suction specific speed, respectively. Figure F.1 reflects the impact of

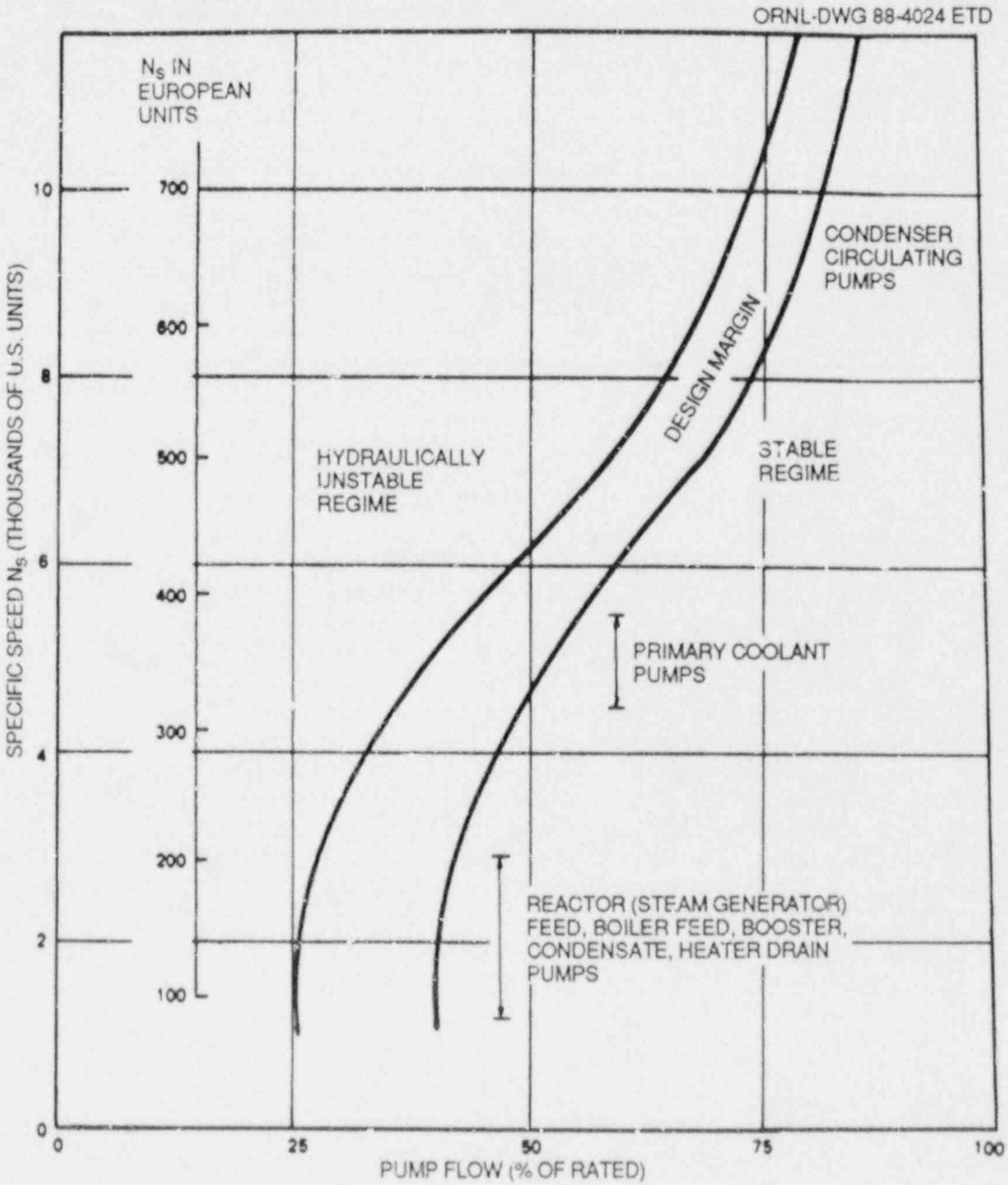

Fig. F.1. Anticipated useful operating ranges for pumps used in large nuclear and fossil power generating units.

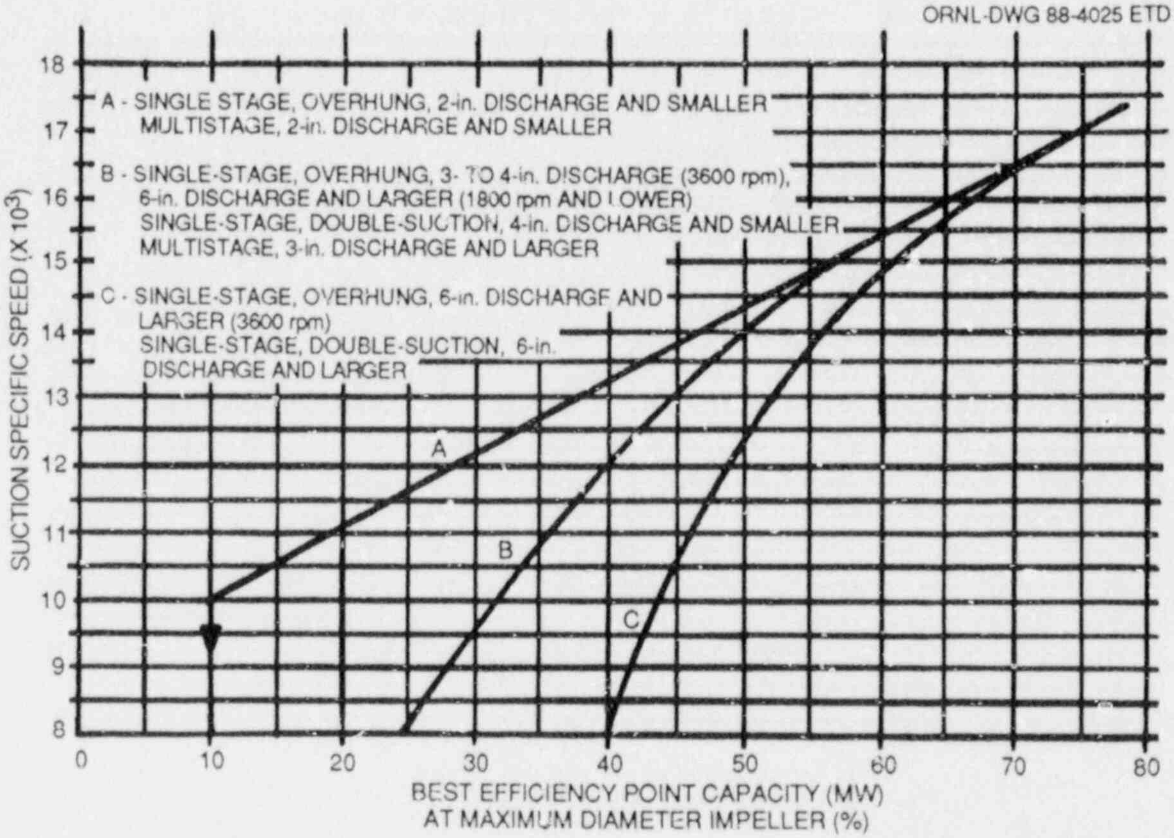

Fig. F.2. Minimum continuous stable flow, percentage of BEP capacity at maximum diameter impeller. Adapted with permission from C. C. Heald and R. Palgrave, "Backflow Control Improves Pump Performance," *Oil Gas J.* (Feb. 25, 1985).

power intensity on minimum flow. These criteria result in a minimum continuous flow rate of >25% of BEP. Similar comments can be found in Refs. 3 to 6 in addition to Refs. 1 and 2, already noted.

However, the final selected minimum flow value should be mutually agreeable to both the user and manufacturer and should reflect the experience gained for the specific pump configuration. It will be important to monitor discharge pressure pulsations and shaft vibration when operating at these low flows. Note that specification API 610 defines minimum continuous flow as "the lowest flow at which the pump can operate without exceeding the noise and vibration limits imposed by this specification."

One point to keep in mind is that the AUXFPs operate on miniflow intermittently and not continuously. The overheating effect of excessive temperature could be considered a short-term problem, while the effects of hydraulic internal recirculation will result in a longer-term aging and wear issue. The question of what is adequate flow for continuous vs intermittent operation is now constantly asked, with special emphasis on intermittent operation. The ability for a pump to operate at low flow rates for various periods of time depends on many different factors, some

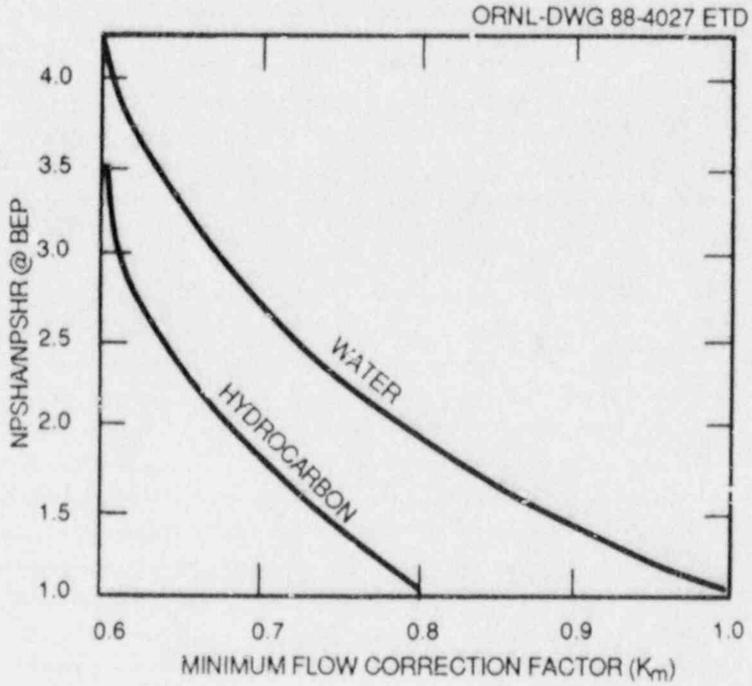

Fig. F.3. Minimum flow correction factor.

of which are listed below:

1. suction specific speed,
2. specific speed,
3. percentage of BEP flow,
4. power intensity (specific speed),
5. available NPSH margin,
6. rotor design,
7. bearing design,
8. shaft design,
9. casing and support design, and
10. seal design.

No proven formulas are available to determine how long a specific pump design can operate at given flow rates. Only through further testing and accumulation of data can this determination be made.

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Appendix G

TEST GUIDELINES FOR FULL-FLOW TESTING

At present, periodic testing at power is performed at relatively "low flows" by utilizing the pump miniflow path. Increasing the flow through the pump miniflow path would most likely require considerable hardware changes and may decrease the reliability of the system to perform its intended functions.

An alternative "full-flow" periodic test that requires no hardware changes was proposed by W a number of years ago. The W-proposed test, which is summarized below, would allow the pumps to deliver flow into the steam generators during power operation. This type of periodic test imposes additional temperature transients on the steam generators and has a small transient impact on the steam generator level control system. However, these additional temperature transients and level transients appear to be well within the design specifications for the W steam generator and steam generator level control systems, respectively. Also, this type of test would require that the source of auxiliary feedwater (typically, the condensate storage tank) during testing be condensate quality and that all noncondensate quality sources of feedwater be securely isolated during testing to protect the long-term structural integrity of the steam generator.

In conclusion, the W-proposed "full-flow" test procedure appears to be a viable alternative to low-flow miniflow testing and can be implemented at minimal cost because it requires no system or hardware design changes.

These guidelines address W-designed plants. No attempt has been made to determine their applicability to other plant designs.

The test guidelines, included in the following pages, are proposed by W as an alternative for upgrading periodic AUXFP testing and to address the latest industry concerns. However, it must be recognized that this procedure should be evaluated by all utilities on an individual plant basis before implementation.

estinghouse-Proposed AUXFP Test Guidelines

1.0 Each AUXFP should be demonstrated operable at least once every quarter by verifying at least one point on the characteristic (head-flow) curve of the pump. The portion of the head-flow curve that is verified should ensure that the pump can deliver its design flow. The AUXFPs should be tested individually one at a time.

1.1 If it should be necessary to isolate an AUXFP by closing pump discharge valves to test the pump or some other portion of the system, only one AUXFP should be so isolated or tested at any given time. During this period of isolation, the other AUXFPs should remain operable, and the system alignment should be such that an automatic start signal will actuate the remaining pump(s), allowing them to deliver flow into the steam generators.

1.2 The operability of an AUXFP should be demonstrated by manually starting the pump and allowing the pump to deliver flow into the steam generator feedwater lines, provided that the precautions and limitations discussed in item 1.3 below are observed. Such a test actually demonstrates the capability of the system to deliver flow into the steam generators and should be performed quarterly.

1.3 To demonstrate the ability of the AUXFPs to deliver into the steam generators, each pump should be tested individually. Each pump should be permitted to deliver flow into all steam generators to which it is normally aligned. For regular quarterly testing purposes, the flow rate delivered into any one steam generator should not exceed 500 gal/min. Only condensate quality water may be injected into the steam generators.

The remote manual flow control valve(s) in the branch discharge line(s) leading from the pump to the steam generator(s) may be closed immediately preceding the test and thereafter slowly opened progressively during the test until the flow rate to any steam generator reaches the 500-gal/min limitation. This procedure should provide that the aforementioned flow limit is observed and should minimize the transients experienced by the steam generator and the flow and level control systems.

With the pump running, the operators should observe or record pump discharge pressure, pump suction pressure, discharge flow, recirculation flow, flow to each steam generator, turbine inlet pressure, and pump speed (turbine pumps), in addition to other pertinent test data, and should verify that these performance indicators conform to expectations. After stopping the pump, the remote manual control valves in the branch discharge lines should be returned to the full-open position.*

1.4 Following quarterly pump testing, the operator should verify that each valve (whether it be manual, power-operated, or automatic) in the auxiliary feedwater flowpaths is in its correct position.*

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*Standard technical specifications (STS) require this check only for valves not locked, sealed, or secured in position.

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2 TITLE AND SUBTITLE
Aging and Service Wear of Auxiliary Feedwater Pumps for PWR Nuclear Plants. Volume 2. Aging Assessments and Monitoring Method Evaluations

3 LEAVE BLANK

4 DATE REPORT COMPLETED
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5 AUTHOR(S)
D. M. Kitch, J. S. Schlonski, P. J. Sowatskey, W. V. Cesarski

6 DATE REPORT ISSUED
MONTH | YEAR

7 PERFORMING ORGANIZATION NAME AND MAILING ADDRESS (Include Zip Code)
Oak Ridge National Laboratory
P.O. Box 2009
Oak Ridge, Tennessee 37831-8063

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)

Failure causes attributable to aging and service wear of auxiliary feedwater pumps (AUXFPs) are given and ranked in terms of importance. Cause identifications are made on the bases of experience, post service examinations, and in situ assessments.

Measurable parameters related to failure causes are identified. ISCM methods are also identified, evaluations are made based on Westinghouse experience, and recommendations are given. The methods are intended to yield required capabilities for establishing operational readiness, as well as for detecting and tracking degradation.

The role of maintenance in mitigating aging and service wear effects is discussed, and the relationship of maintenance to ISCM methods is identified. Predictive, preventive, and corrective maintenance practices are discussed and evaluated.

Appendixes are included that contain failure data base information, AUXFP installation lists, discussion of low-flow testing, auxiliary feedwater system description, AUXFP minimum-flow-rate criteria, and proposed guidelines for full-flow testing.

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, hydraulic instability, 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

U.S. GOVERNMENT PRINTING OFFICE: 1988-548-118/80048

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