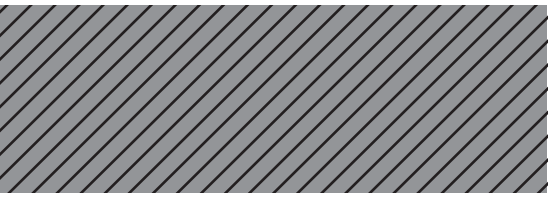
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EPRI TR-107252
Final Report
October 1997

Centrifugal and Positive Displacement Charging Pump Maintenance Guide



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REPORT SUMMARY

Centrifugal and Positive Displacement Charging Pump Maintenance Guide

In pressurized water reactors, charging pumps provide charging flow to the RCS and are also required to be available in the event of a loss of coolant accident (LOCA) event. Pump availability combined with modes of operation have made pump maintenance both frequent and costly. This report provides techniques to assist utilities in reducing maintenance costs for these pumps. Areas for improving reliability of these pumps include: excessive seal leakage, definition of acceptable seal leakage, packing life, and shaft cracking.

INTEREST CATEGORIES

Maintenance practices
Nuclear plant operations and maintenance
Engineering and technical support

KEYWORDS

Maintenance
Pumps
Packing
Seals

BACKGROUND NMAC-member utilities requested additional guidance on maintaining and improving the reliability of charging pumps. The need to reduce maintenance costs has given impetus to the requirement to better understand and implement successful maintenance practices.

OBJECTIVES To provide the latest information on successful maintenance practices for centrifugal and positive displacement pumps

APPROACH This guide was developed by a team of pump industry and field service personnel along with plant owners and personnel who maintain these pumps.

RESULTS The report provides comprehensive guidance on maintaining and improving the reliability of nuclear plant charging pumps. The document addresses charging pump design and construction, presents an overview of pump theory and design, and construction considerations for both centrifugal and positive displacement pumps and their major components. The report provides detailed guidance on:

- Predictive and preventative maintenance, including condition-based maintenance and data collection techniques that can be used to predict failures
- Charging pump maintenance, including generic procedures for performing recommended actions
- Post-maintenance startup, operation, and testing
- Westinghouse and Combustion Engineering charging pump systems
- Reliability-centered maintenance
- Shaft alignment
- Pump design enhancements

EPRI PERSPECTIVE This guide can assist utilities in improving charging pump maintenance.

Centrifugal and Positive Displacement Charging Pump Maintenance Guide

TR-107252

Final Report
October 1997

Prepared by:
Life Cycle Engineering, Inc.
4360 Corporate Road, Suite 100
North Charleston, SC 29405

Principal Investigators:
Tom Pearson
Peter Douglas
Don Gurley

Prepared for
Nuclear Maintenance Applications Center
1300 Harris Boulevard
Charlotte, NC 28262

Operated by
Electric Power Research Institute
3412 Hillview Avenue
Palo Alto, CA 94304

EPRI Project Manager
Randal Kerr
Nuclear Power Group

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ABSTRACT

Charging pumps used in pressurized water reactors (PWR) are components critical to nuclear plant operations. Two basic types of pumps, centrifugal and positive displacement (piston), are used in charging pump applications.

This guide presents comprehensive methods to achieve increased charging pump reliability through the application of proper predictive and preventive maintenance practices. The document addresses charging pump design and construction, presents and overview of pump theory and design, and construction considerations for both centrifugal and positive displacement pumps and their major components. The report provides detailed guidance on:

- Predictive and preventative maintenance, including condition-based maintenance and data collection techniques, that can be used to predict failures
- Charging pump maintenance, including generic procedures for performing recommended actions
- Post-maintenance startup, operation, and testing
- Westinghouse and Combustion Engineering charging pump systems
- Reliability-centered maintenance
- Shaft alignment
- Pump design enhancements

This guide contains information that is useful to utility personnel who are involved in maintenance-related activities. It is intended for use by maintenance craft and maintenance management personnel who are interested in improving the performance of their pumping equipment. Maintenance engineers, system engineers, engineering technicians, planners, and performance monitoring personnel will find this guide an important resource when updating maintenance procedures, providing technical direction, preparing work control documents, and evaluating the condition of installed equipment.

ACKNOWLEDGMENTS

This guide was prepared for the Nuclear Maintenance Applications Center (NMAC), under Electric Power Research Institute (EPRI) project number RP-3814-16. Many individuals and organizations provided assistance and information to make this publication possible.

The Ingersoll-Dresser Pump Company, APV Gaulin, Inc., and Union Pump Company—manufacturers of nuclear charging pumps—and Garlock, Inc., John Crane, Inc., and UTEX Industries are to be commended for providing the information necessary to make this guide complete and useful, and for granting permission to reproduce illustrations and use their technical data.

NMAC and the authors of this guide would like to particularly recognize the following members of the Technical Advisory Group for their timely and detailed contributions to this project:

Joe Badalamenti	Union Pump Company
John Bristol	APV Gaulin, Inc.
François Cattant	Electricité de France
Eric Colvin	Westinghouse Energy Center
Bill Darbyshire	Pacific Gas & Electric Co. – Diablo Canyon
Tim Donovan	Garlock, Inc.
Michael Duckworth	Ingersoll-Dresser Pump Company
Lee Hentz	Duke Power Co. – McGuire Nuclear Station
Grover Hettel	Entergy Operations, Inc. – Arkansas Nuclear One
John Huber	Baltimore Gas & Electric – Calvert Cliffs
Kent Huber	BW/IP International, Inc.
Bob Joines	Ingersoll-Dresser Pump Company
Steve Kochert	Union Electric Co. – Callaway Nuclear Station
John Koeplin	ComEd – Byron Nuclear Power Station
Steven Lemberger	John Crane, Inc.
Steve Mays	Duke Power Co. – Catawba Nuclear Station
Fred Pippert	UTEX Industries
Terry Price	Arizona Public Service Co. – Palo Verde
David Sabigia	ABB Combustion Engineering
Dean Shah	Consolidated Edison Co. – Indian Point
Thomas R. Smith	Public Service Electric & Gas – Salem Nuclear Station
Greg Thoraldson	Prairie Island Nuclear Station
Jack Wade	Entergy Operations, Inc. – Waterford 3 SES
Paul Weeks	Westinghouse Spartanburg Service Center

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INTRODUCTION

Charging pumps are safety-related equipment, designed to provide normal charging service to the reactor coolant system (RCS) and to act as the high-head safety injection pump. Charging pumps are part of the chemical and volume control system (CVCS) and provide makeup water to the RCS during transient conditions, provide high-pressure seal water for the reactor coolant pump seals, and allow for chemical additions to the RCS.

Two basic types of charging pumps are in use throughout the industry: centrifugal and positive displacement pumps. In plants utilizing both centrifugal and positive displacement charging pumps, flow is automatically controlled by the charging pump flow controller that maintains pressurizer water level at the setpoint programmed for the reactor coolant average temperature. For plants utilizing only positive displacement pumps, charging flow is either constant and letdown flow is varied or charging flow is varied with constant letdown flow to control the pressurizer water level.

A pump application matrix, presented as Table 1-1, lists the types of charging pumps that are installed in nuclear plants. This matrix proved useful in the development of this manual, and it is included here for use by plant engineers and technicians. This information should aid plant personnel in establishing a dialogue with like-pump users to discuss pump problems and solutions. Figure 1-1 shows charging pump utilization, which gives the relative population of pump types.

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Table 1-1
Charging Pump Plant Installations

Site	Utility	NSSS	Centrifugal Pump Types			Positive Displacement Pump Types types		
			Quantity	Pump Mfg.	Size	Quantity	Pump Mfg.	Size
Arkansas Nuclear One 1 Arkansas Nuclear One 2	Entergy Operations, Inc.	B&W CE	*3*	Byron Jackson	DVMX 3x4x8-9 Stage	*3*	Manton Gaulin	2640 NP18-3TPS
Beaver Valley 1 Beaver Valley 2	Duquesne Light Co.	WE WE	*3* *3*	Dresser Pacific Dresser Pacific	2.5RLIJ-11 Stage 2.5RLIJ-11 Stage			
Braidwood 1 Braidwood 2	ComEd	WE WE	*2* *2*	Dresser Pacific Dresser Pacific	2.5RLIJ-11 Stage 2.5RLIJ-11 Stage	*1* *1*	Manton Gaulin Manton Gaulin	NP45-3 NP45-3
Byron 1 Byron 2	ComEd	WE WE	*2* *2*	Dresser Pacific Dresser Pacific	2.5RLIJ-11 Stage 2.5RLIJ-11 Stage	*1* *1*	Manton Gaulin Manton Gaulin	NP45-3 NP45-3
Callaway 1	Union Electric Co.	WE	*1* *2*	Ingersoll Rand Dresser Pacific	2x10-12CAM500X1 2.5RLIJ-11 Stage	*1*	Union Pump	2-7/16x5QX-300
Calvert Cliffs 1 Calvert Cliffs 2	Baltimore Gas & Electric Co.	CE CE				*4* *3*	Armco Steel Armco Steel	J-531-M6DF J-531-M6DF
Catawba 1 Catawba 2	Duke Power Co.	WE WE	*2* *3*	Dresser Pacific Dresser Pacific	2.5RLIJ-11 Stage 2.5RLIJ-11 Stage	*1* *1*	Union Pump Union Pump	2-7/16x5QX-300 2-7/16x5QX-300
Comanche Peak 1 Comanche Peak 2	Texas Utilities Electric Co.	WE WE	*2* *2*	Dresser Pacific Dresser Pacific	2.5RLIJ-11 Stage 2.5RLIJ-11 Stage	*1* *1*	Union Pump Union Pump	2-7/16x5QX-300 2-7/16x5QX-300
Cook 1 Cook 2	Indiana/Michigan Power Co.	WE WE	*2* *2*	Dresser Pacific Dresser Pacific	2.5RLIJ-11 Stage 2.5RLIJ-11 Stage	*1* *1*	Union Pump Union Pump	2-7/16x5QX-300 2-7/16x5QX-300
Crystal River 3	Florida Power Corporation	B&W	*4*	Sulzer Bingham	3x5x7.5 MSD-9 Stage			
Davis Besse 1	Toledo Edison Co.	B&W	*2*	Sulzer Bingham	3x5x7.5CP-12 Stage			
Diablo Canyon 1 Diablo Canyon 2	Pacific Gas & Electric Co.	WE WE	*3* *2*	Dresser Pacific Dresser Pacific	2.5RLIJ-11 Stage 2.5RLIJ-11 Stage	*1* *1*	Union Pump Union Pump	2-7/16x5QX-300 2-7/16x5QX-300

Table 1-1
Charging Pump Plant Installations (continued)

Site	Utility	NSSS	Centrifugal Pump Types			Positive Displacement Pump Types types		
			Quantity	Pump Mfg.	Size	Quantity	Pump Mfg.	Size
Farley 1 Farley 2	Southern Nuclear Operating Co.	WE WE	*3* *3*	Dresser Pacific Dresser Pacific	2.5RLIJ-11 Stage 2.5RLIJ-11 Stage			
Fort Calhoun 1	Omaha Public Power District	CE				*3*	Manton Gaulin	2640 NP18-3TPS
Ginna	Rochester Gas & Electric Corp.	WE				*3*	Ajax Iron	T-125
Haddem Neck	Northeast Utilities	WE	*3*	Dresser Pump	2Z	*1*	Ajax	T-75
Harris	Carolina Power & Light Co.	WE	*3*	Dresser Pacific	2.5RLIJ-11 Stage			
Indian Point 2 Indian Point 3	Consolidated Edison Co. New York Power Authority	WE WE				*3* *3*	Union Pump Union Pump	2-7/16x5QX-300 2-7/16x5QX-300
Kewaunee	Wisconsin Public Service Corp.	WE				*4*	Ajax Iron	T-150
Maine Yankee	Maine Yankee Atomic Power	CE	*3*	Sulzer Bingham	3x5x7.5 MSD-9 Stage			
McGuire 1 McGuire 2	Duke Power Co.	WE WE	*2* *2*	Dresser Pacific Dresser Pacific	2.5RLIJ-11 Stage 2.5RLIJ-11 Stage	*1* *1*	Union Pump Union Pump	2-7/16x5QX-300 2-7/16x5QX-300
Millstone 2 Millstone 3	Northeast Utilities	CE WE	 *3*	 Dresser Pacific	 2.5RLIJ-11 Stage	*3* *3*	Armco Steel Manton Gaulin	J-531-M6DF 2640 NP18-3TPS
North Anna 1 North Anna 2	Virginia Power Co.	WE WE	*3* *3*	Dresser Pacific Dresser Pacific	2.5RLIJ-11 Stage 2.5RLIJ-11 Stage			
Oconee 1 Oconee 2 Oconee 3	Duke Power Co.	B&W B&W B&W	*4* *5* *3*	Ingersoll Rand Ingersoll Rand Ingersoll Rand	2.5VHTB-24 2.5VHTB-24 2.5VHTB-24			
Palisades	Consumers Energy	CE				*2* *1*	Manton Gaulin Manton Gaulin	2640 NP18-3TPS NP45-3

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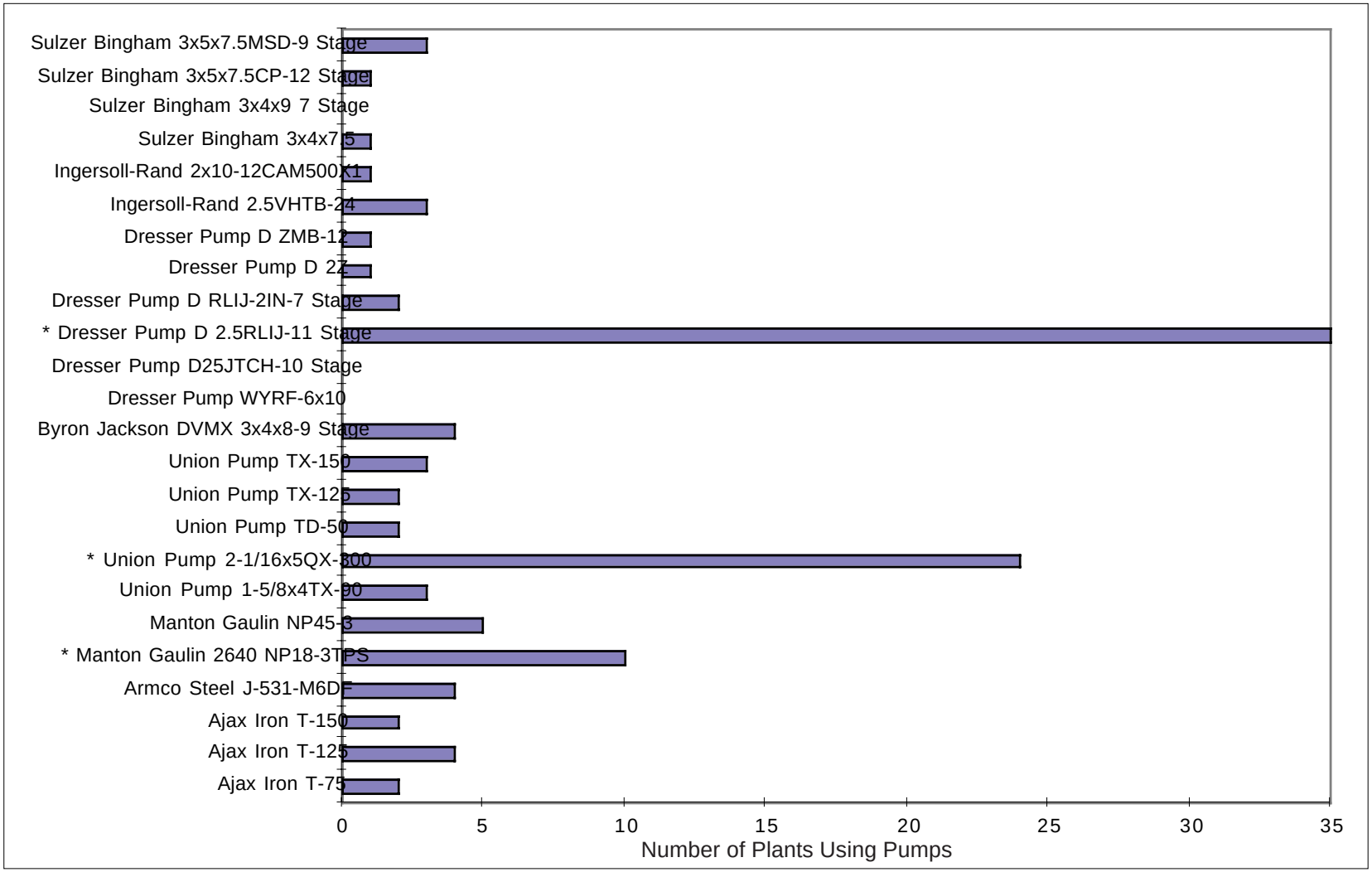
Table 1-1
Charging Pump Plant Installations (continued)

Site	Utility	NSSS	Centrifugal Pump Types			Positive Displacement Pump Types types		
			Quantity	Pump Mfg.	Size	Quantity	Pump Mfg.	Size
Palo Verde 1	Arizona Public Service Co.	CE				3	Manton Gaulin	2640 NP18-3TPS
Palo Verde 2		CE				3	Manton Gaulin	2640 NP18-3TPS
Palo Verde 3		CE				3	Manton Gaulin	2640 NP18-3TPS
Point Beach 1	Wisconsin Electric Power Co.	WE				3	Ajax Iron	T-125
Point Beach 2		WE				3	Ajax Iron	T-125
Prairie Island 1	Northern States Power	WE				5	Ajax Iron	T-125
Prairie Island 2		WE				3	Ajax Iron	T-150
Robinson	Carolina Power & Light Co.	WE				3	Union Pump	TX-150
Salem 1	Public Service Electric & Gas	WE	2	Dresser Pacific	2.5RLIJ-11 Stage	1	Union Pump	2-7/16x5QX-300
Salem 2		WE	2	Dresser Pacific	2.5RLIJ-11 Stage	1	Union Pump	2-7/16x5QX-300
San Onofre 1	Southern California Edison	WE	2	Dresser Pump	Dresser ZMB-12	1	Union Pump	TD-50
San Onofre 2		CE				5	Manton Gaulin	2640 NP18-3TPS
San Onofre 3		CE				3	Manton Gaulin	2640 NP18-3TPS
Seabrook 1	North Atlantic Energy Service Co.	WE	2	Dresser Pump	2.5RLIJ-11 Stage	1	Union Pump	2-7/16x5QX-300
Sequoyah 1	Tennessee Valley Authority	WE	2	Dresser Pump	2.5RLIJ-11 Stage	1	Union Pump	2-7/16x5QX-300
Sequoyah 2		WE	2	Dresser Pump	2.5RLIJ-11 Stage	1	Union Pump	2-7/16x5QX-300
South Texas 1	Houston Lighting & Power Co.	WE	2	Dresser Pump	RLIJ-2IN-7 Stage	1	Union Pump	1-5/8x4TX-90
South Texas 2		WE	2	Dresser Pump	RLIJ-2IN-7 Stage	1	Union Pump	1-5/8x4TX-90
St. Lucie 1	Florida Power & Light Co.	CE				3	Armco Steel	J-531-M6DF
St. Lucie 2		CE				3	Union Pump	TX-125
Summer	South Carolina Electric & Gas Co.	WE	3	Dresser Pump	2.5RLIJ-11 Stage			
Surry 1	Virginia Power Co.	WE	3	Byron Jackson	DVMX 3x4x8-9 Stage			
Surry 2		WE	3	Byron Jackson	DVMX 3x4x8-9 Stage			

Table 1-1
Charging Pump Plant Installations (continued)

Site	Utility	NSSS	Centrifugal Pump Types			Positive Displacement Pump Types types		
			Quantity	Pump Mfg.	Size	Quantity	Pump Mfg.	Size
Three Mile Island 1	GPU Nuclear Corp.	B&W	*3*	Sulzer Bingham	3x4x7.5			
Turkey Point 3 Turkey Point 4	Florida Power & Light Co.	WE WE				*3* *3*	Union Pump Union Pump	TX-150 TX-150
Vogtle 1 Vogtle 2	Georgia Power Co.	WE WE	*2* *2*	Dresser Pump Dresser Pump	2.5RLIJ-11 Stage 2.5RLIJ-11 Stage	*1* *1*	Union Pump Union Pump	2-7/16x5QX-300 2-7/16x5QX-300
Waterford 3	Entergy Operations, Inc..	CE				*4*	Manton Gaulin	2640 NP18-3TPS
Watts Bar 1	Tennessee Valley Authority	WE	*2*	Dresser Pump	2.5RLIJ-11 Stage	*2*	Union Pump	2-7/16x5QX-300
Wolf Creek	Wolf Creek Nuclear Operating Corp.		*2*		2.5RLIJ-11 Stage	*1*	Union Pump	2-7/16x5QX-300
Zion 1 Zion 2	ComEd	WE WE	*2* *2*	Dresser Pump Dresser Pump	2.5RLIJ-11 Stage 2.5RLIJ-11 Stage	*1* *1*	Union Pump Union Pump	2-7/16x5QX-300 2-7/16x5QX-300

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* Most prevalent pump installations

Figure 1-1
Charging Pump Type Utilization by Plant

The primary focus of this manual is on the most industry-prevalent charging pumps: Ingersoll-Dresser's Pacific centrifugal pumps, and Union and Gaulin positive displacement pumps. However, many aspects of this guide can be applied to any centrifugal or positive displacement piston pump. Figures 1-2 through 1-6 show typical charging pump installations of the abovementioned pump types and models.

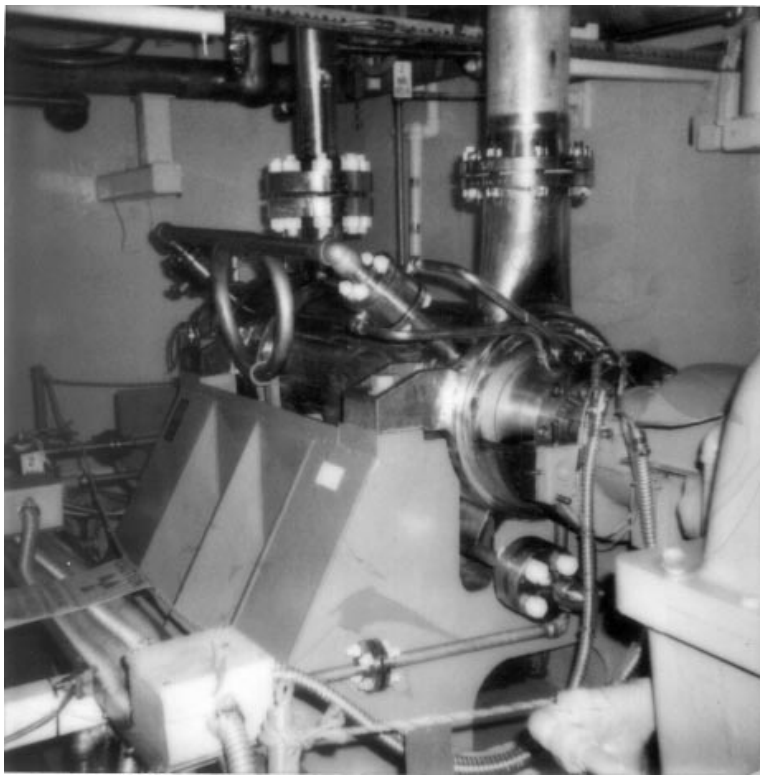


Figure 1-2
Pacific Centrifugal Charging Pump, McGuire Installation



Figure 1-3
Pacific Centrifugal Charging Pump, Salem Installation

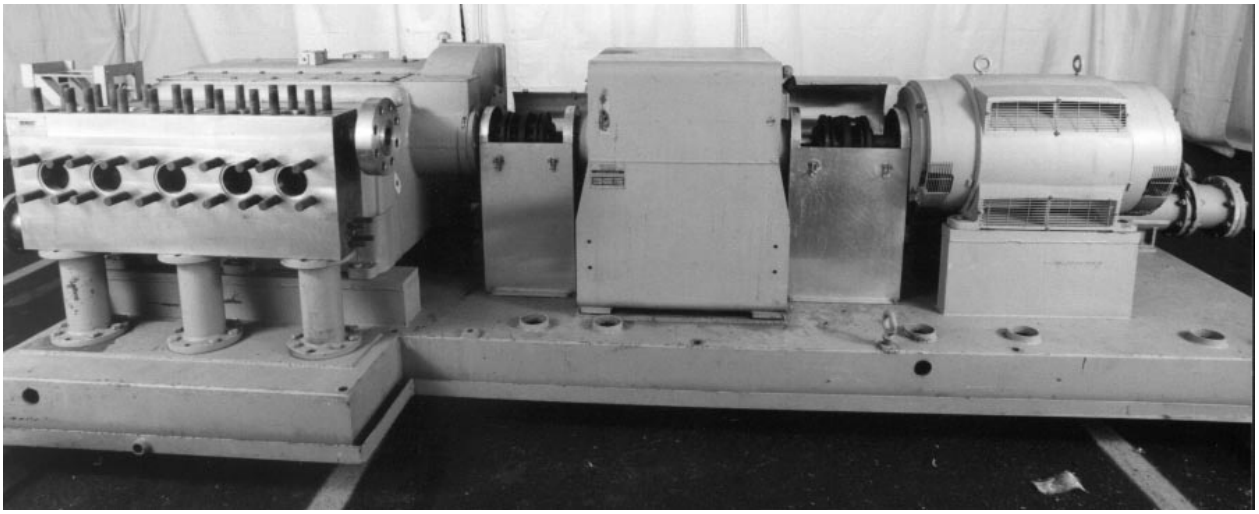


Figure 1-4
Overview of Union Model QX-300 Quintiplex Reciprocating Pump with Stripped Fluid End, Showing Layout of Pump, Gyrol Fluid Drive, and Westinghouse Motor

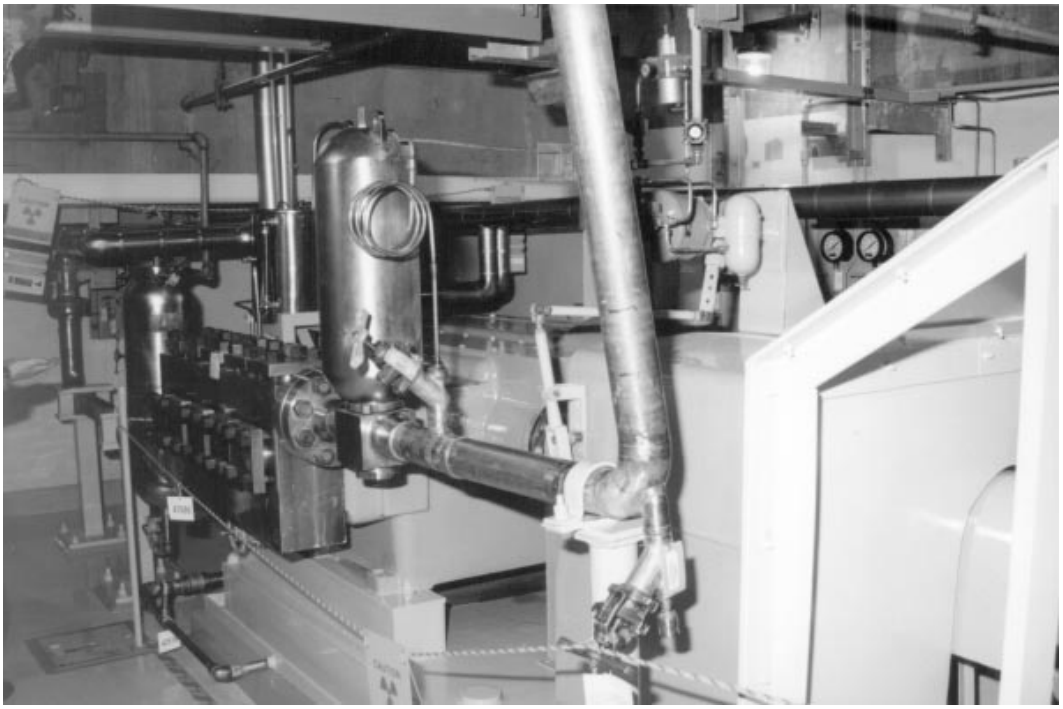


Figure 1-5
Union Positive Displacement Pump, Salem Installation

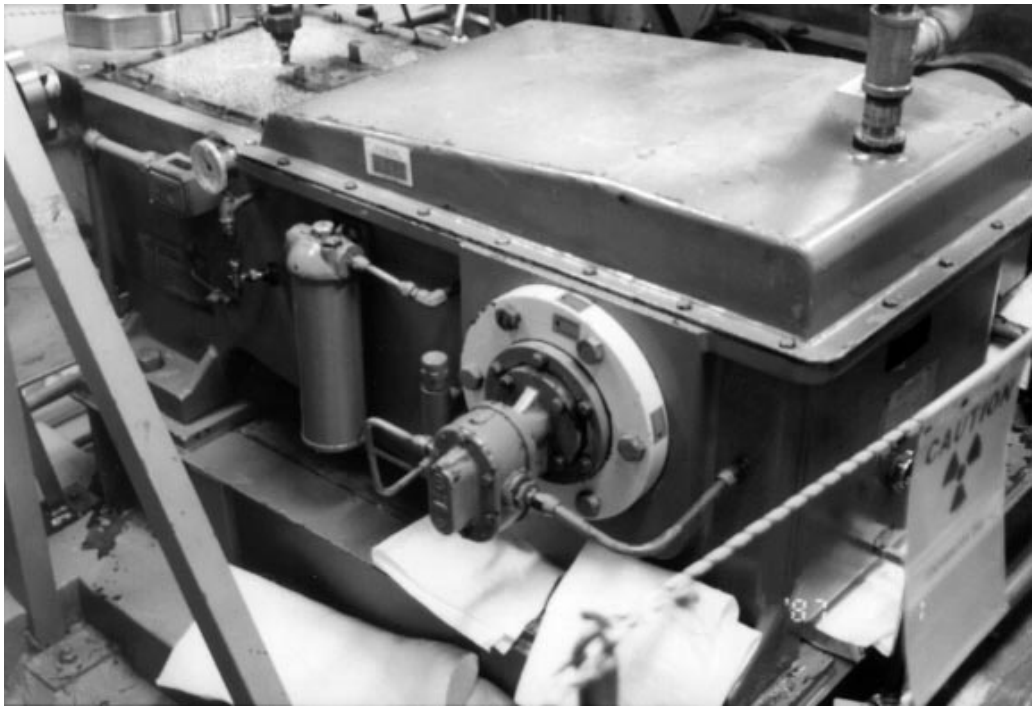


Figure 1-6
Power End View, Gaulin Triplex Reciprocating Charging Pump, Showing Crankshaft-Mounted Oil Lubrication Pump, Arkansas Nuclear One Installation

1.1 Organization of This Guide

Chapter 2, Charging Pump Design and Construction, presents an overview of pump theory and design, and construction considerations for both centrifugal and positive displacement pumps and their major components. Discussions on bearings, couplings, and drivers are provided to further develop an understanding of the systems approach to maintenance. This chapter explains the relationship between pump components and their functions, thus establishing the foundation for maintenance decisions.

Chapter 3, Predictive and Preventive Maintenance, describes condition-based maintenance and data collection techniques that can be used to predict failures. These actions are presented as equipment maintenance plans and include both predictive and preventive actions designed to maintain pump reliability.

Chapter 4, Charging Pump Maintenance, discusses maintenance actions that can be used to prevent failures. Generic procedures for performing the recommended actions are presented to guide maintenance personnel in evaluating or developing their own specific procedures for plant use.

Chapter 5, Post-Maintenance Startup, Operation, and Testing, presents the steps that should be considered after shutdown maintenance has been performed.

Appendices A and B present system discussions for the Combustion Engineering– and Westinghouse-designed plants. System information on plant startup, normal operation, and abnormal operation is discussed.

Appendix C contains a general discussion of reliability-centered maintenance (RCM). Shaft alignment is discussed extensively in Appendix D. Appendix E provides pump design enhancements.

1.2 System Overview

This section provides a general overview of charging/letdown system operation. Although all plants use some form of a makeup water system, the specific operating details can differ from plant to plant. See Figure 1-7 for a typical Combustion Engineering charging system and Figure 1-8 for a typical Westinghouse system.

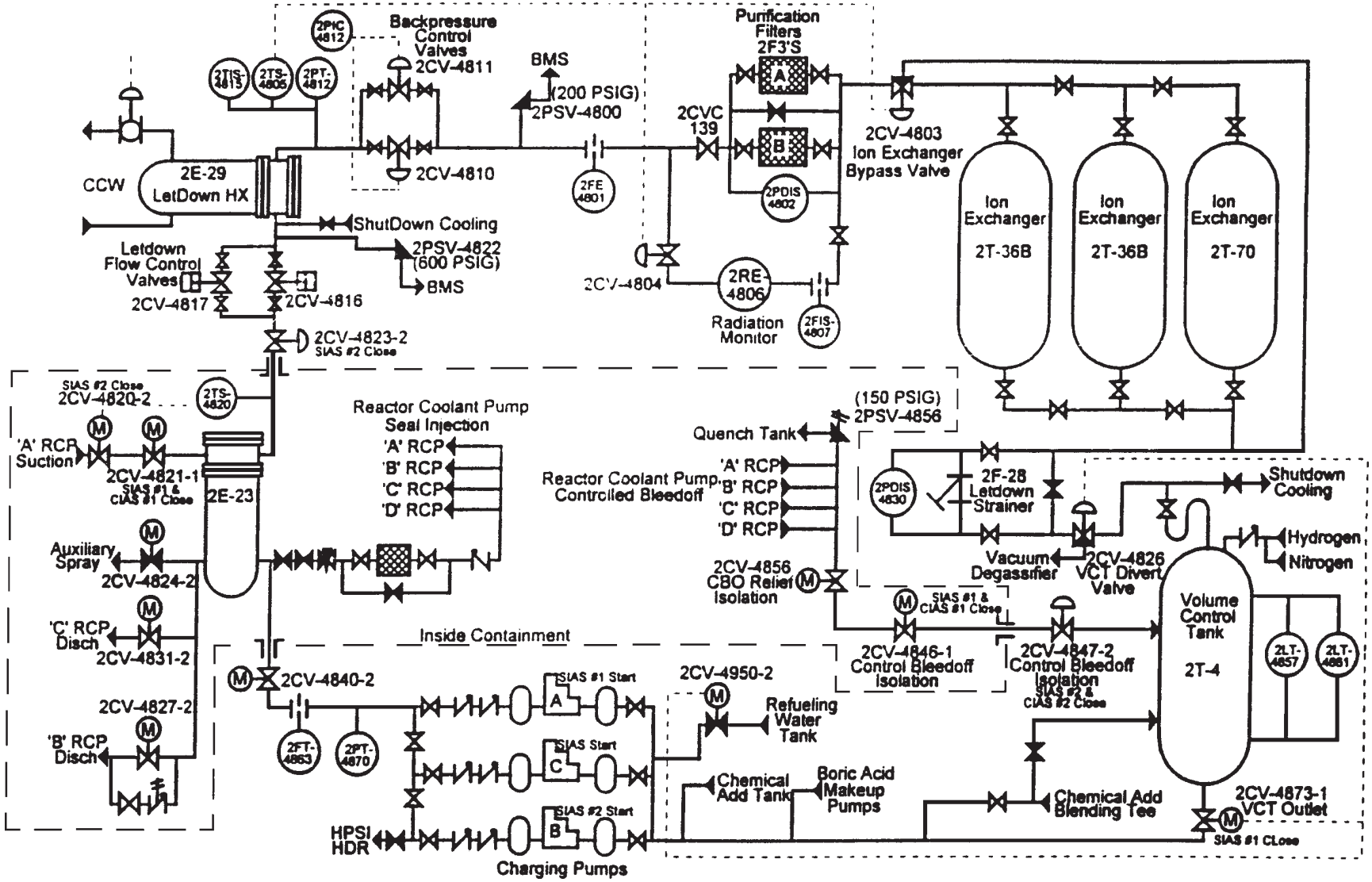


Figure 1-7
Typical Combustion Engineering CVCS Schematic

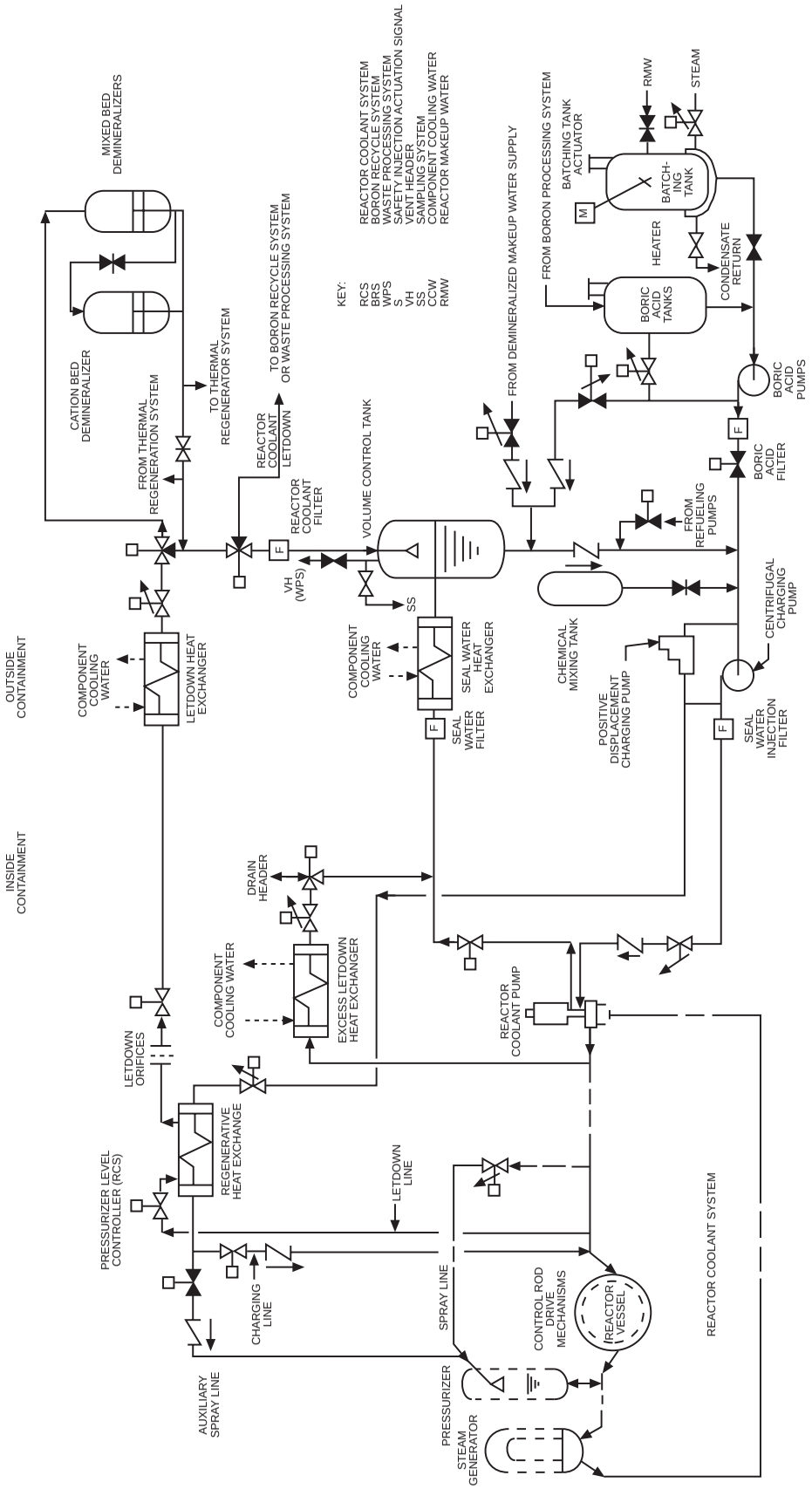


Figure 1-8
Typical Westinghouse CVCS Schematic

During power operation, a continuous feed-and-bleed stream is maintained to and from the reactor coolant system (RCS). Letdown water leaves the RCS and flows through the shell side of the regenerative heat exchanger, where it gives up its heat to makeup water being returned to the RCS. The letdown water then flows through orifices or valves, where its pressure is reduced, then through the letdown heat exchanger, followed by a second pressure reduction through the low-pressure letdown valve.

After passing through a mixed bed demineralizer, where ionic impurities are removed, the water flows either through the thermal regeneration resins or directly through the reactor coolant filter and into the volume control tank (VCT). The vapor space in the VCT contains hydrogen, which dissolves in the coolant. Any fission gases present are removed from the system by venting of the VCT.

All charging pumps take coolant from the VCT. Westinghouse units pump through two parallel paths: to the RCS through and to the seals of the reactor coolant pumps. The streams divide; some water flows into the RCS, and the remainder leaves the pumps as seal leakage. From the pumps, the leakage water goes to the seal water heat exchanger and then returns to the VCT for another circuit. If the normal letdown and charging path through the regenerative heat exchanger is not operable, water injected into the RCS through the reactor coolant pump seals is returned to the VCT through the excess letdown heat exchanger.

Combustion Engineering units pump through the tube side of the regenerative heat exchanger into the RCS. Controlled bleed-off from the reactor coolant pump seal is through normal RCS and return to the VCT.

Makeup fluid to the chemical and volume control system (CVCS) comes from the following sources:

- Demineralized and deaerated water supply, when the concentration of dissolved neutron absorber is to be reduced
- Boric acid tank, when the concentration of dissolved neutron absorber is to be increased
- A blend of demineralized and deaerated water and concentrated boric acid to match the reactor coolant boron concentration for normal plant makeup
- Refueling water storage tank for emergency makeup of borated water
- Chemical mixing tank for small quantities of hydrazine for oxygen scavenging or lithium hydroxide for pH control

1.3 System Operation

1.3.1 Plant Startup

Charging flow is initiated and the normal letdown flow path is established with all orifices opened. Pressure in the RCS is automatically regulated by throttling the low-pressure letdown valve. When a pressure adequate to meet reactor coolant pump net positive suction head (NPSH) and pump seal requirements has been attained, the pumps are operated intermittently to sweep any entrained gases from the RCS during the venting operation.

After filling and venting, the pressurizer liquid temperature is raised to a level where the pressure in the pressurizer is adequate for reactor coolant pump NPSH and seal requirements. The charging pump flow and low-pressure letdown valve are controlled manually to obtain the desired pressurizer water level. The charging flow control then maintains the normal pressurizer water level.

A hydrogen blanket is established in the VCT by raising the tank water level and purging with nitrogen gas. As the water level is lowered, a gas blanket of hydrogen is introduced. Reactor coolant makeup control is operated to adjust the boron concentration and thus maintain the desired control rod position. Chemicals are added from the chemical mixing tank as required to control the reactor coolant dissolved oxygen and pH. When the reactor pressure has been increased sufficiently, the standby orifices are closed to maintain the letdown flow within the normal limits. The reactor coolant makeup control is then set for the desired reactor coolant boron adjustment.

1.3.2 Normal Operation

The charging or letdown flow controller keeps the pressurizer water level at the setpoint programmed for the reactor coolant average temperature that is being maintained. At a constant power level, the letdown flow equals the charging flow plus the flow through the reactor coolant pump seals. The letdown flow is first cooled by the charging stream in the regenerative heat exchanger; then the letdown flow is reduced to approximately 130°F in the letdown heat exchanger.

The letdown flow passes through one of the mixed bed demineralizers, through the reactor coolant filter, and into the VCT. Hydrogen overpressure in this tank maintains the specified concentration of hydrogen in the reactor coolant. From the VCT, a charging pump delivers the reactor coolant to the regenerative heat exchanger, where it is reheated before being returned to the RCS.

During periods of plant loading, the reactor coolant expands as its average temperature rises. The pressurizer absorbs most of this expansion as the level controller raises the level setpoint during load increases. The remainder of the coolant is letdown to the VCT. During this period, the letdown flow remains constant, and the charging flow is reduced by the pressurizer level control signal, resulting in an increased temperature at the regenerative heat exchanger outlet. The temperature controller downstream from the letdown heat exchanger increases the component cooling water flow to maintain the desired letdown temperature. During periods of plant unloading, the charging flow is increased to make up for the coolant contraction that is not accommodated for by the programmed reduction in pressurizer level.

2

CHARGING PUMP DESIGN AND CONSTRUCTION

Pumps add kinetic and potential energy to a liquid to move it from one point to another, thus doing work. Charging pumps are designed to operate within a set of parameters that are defined by the system requirements. The required pumping characteristics (pressure and flow) provide the basis for theoretical pump design. The pumps that are installed in the charging systems were selected because they meet system requirements and other design, construction, installation, and operational criteria relative to cost, performance, reliability, and maintainability.

Figure 2-1 shows a typical charging system schematic.

The characteristics of the high-head system in which the pump must perform are described in Table 2-1.

Table 2-1
Charging System Characteristics

	Westinghouse	Combustion Engineering
Discharge pressure	2400 psi	
Suction pressure	35-50 psi	
Flow (minimum)	60 gpm	44 gpm
Flow (maximum)	550 gpm	132 gpm
Temperature (maximum)	300°F	
Temperature (nominal) letdown	130°F	
Friction head losses	50 ft @ 150 gpm	
Elevation difference (suction to discharge)	30 ft	

The system suction head (H_{sys}) consists of:

- The elevation difference between the VCT and the charging piping/loop connection
- The pressure difference between the VCT and the charging piping/loop connection
- The friction loss in the piping, valves, and fittings

The system suction head can be measured in feet (ft) of water or pounds per square inch (psi) of pressure. For example, a column of water 2.3 feet high (between 40°F and 70°F) produces a pressure at its base of 1 psi.

This relationship varies with the density (temperature) of the water according to the following equation:

$$\text{Pressure (psi)} = (\text{head (ft)} \times \text{s.g.}) \div 2.3 \qquad \text{(Eq. 2-1)}$$

where s.g. = specific gravity = 1 / density

Solving Equation 2-1 for head:

$$\text{Head (ft)} = (\text{Pressure (psi)} \times 2.3) \div \text{s.g.}$$

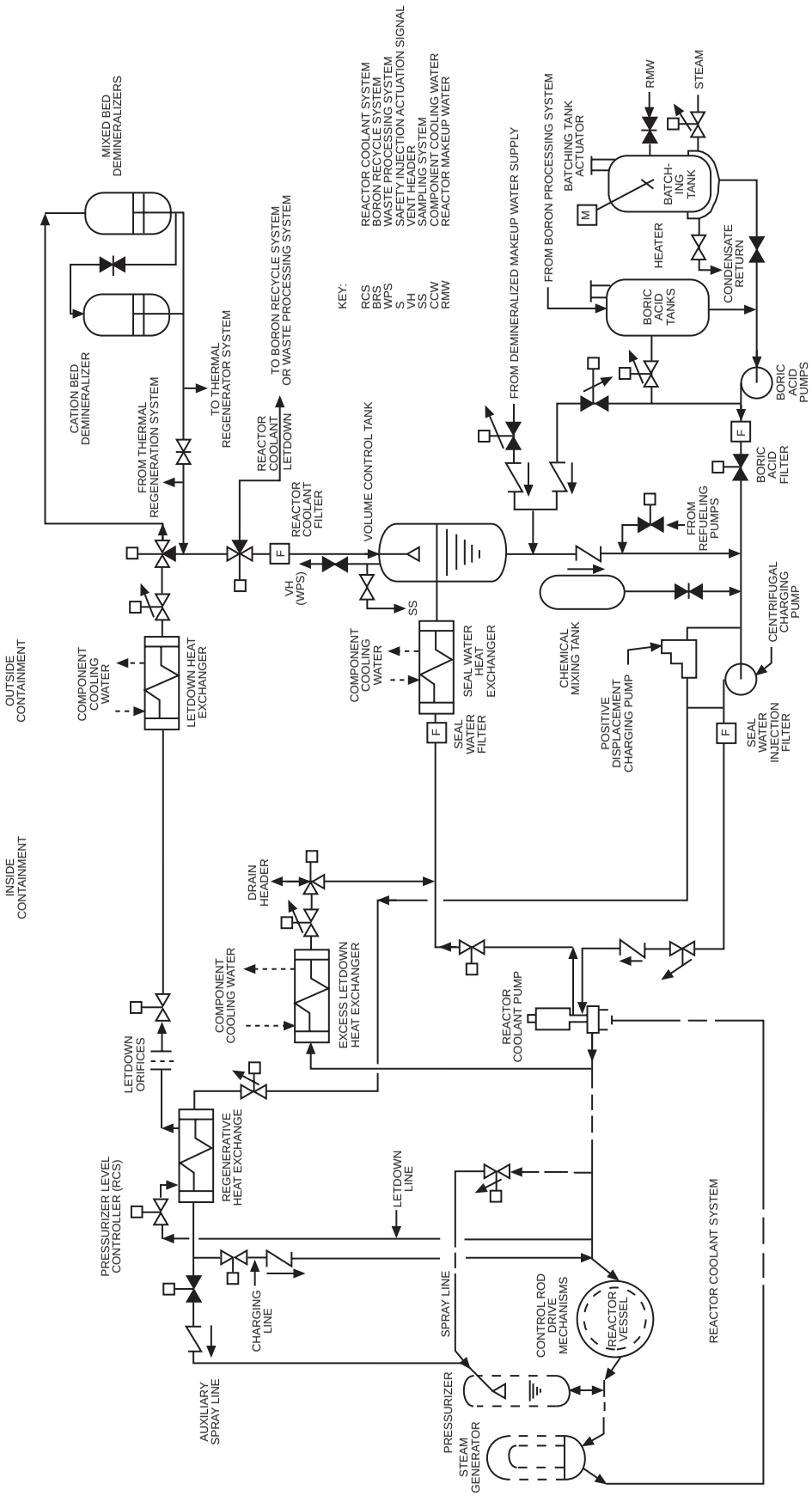


Figure 2-1
Typical Charging System

When figuring the head of a pump, it is important to convert pump pressure to feet at the pumping temperature.

The relationship between head and temperature for a pressure of 500 psi is shown in Figure 2-2.

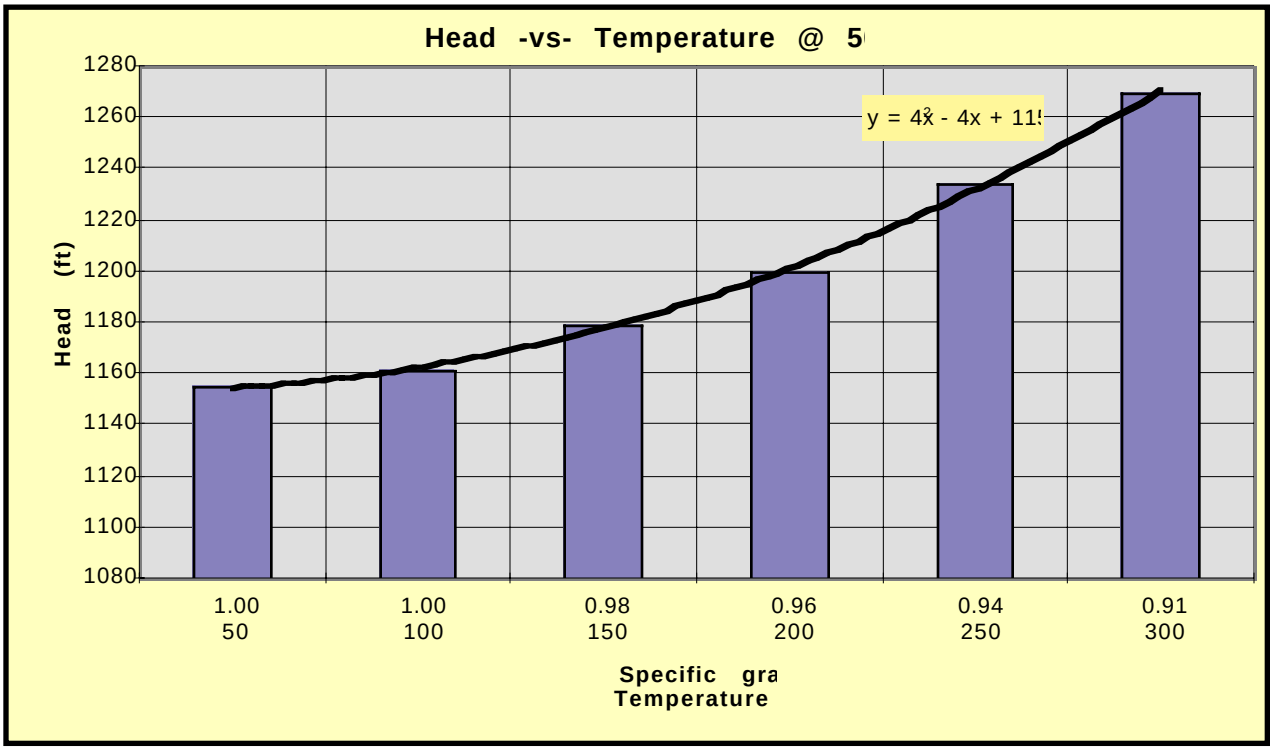


Figure 2-2
Head and Temperature Relationship

Figure 2-3 can be used to compensate for pumping temperatures between 50°F and 300°F when determining pump head (ft) from a pressure gauge (psi) reading. For example, a pressure gauge reading of 2000 psi at 300°F would equate to a pump head of 5100 ft; while the same pressure reading at 50°F would equate to a pump head of about 4600 ft.

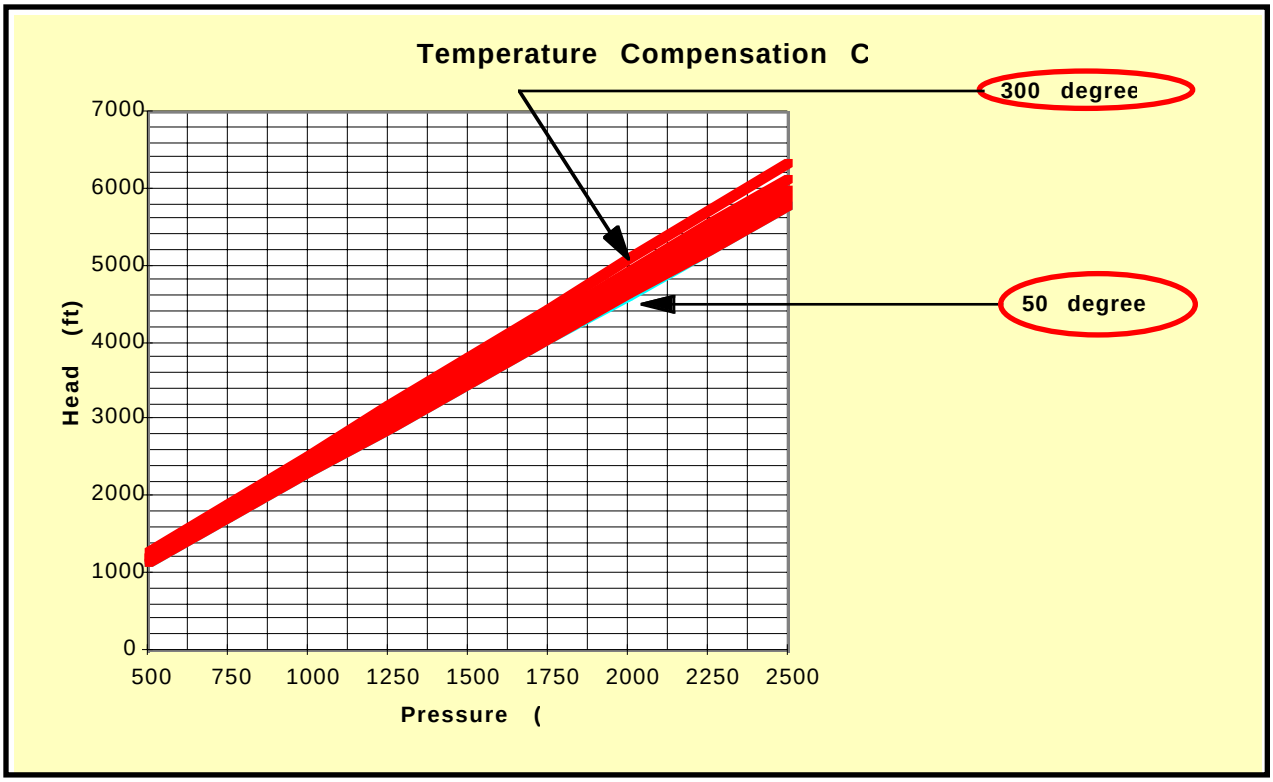


Figure 2-3
Pressure / Head Temperature Compensation Curve

The elevation difference between the VCT and the charging piping / loop connection is called the *static head* of the system and has two components, static discharge head and static suction head (or lift, if the suction liquid level is below the pump centerline), both measured from the pump centerline. The difference between suction pressure and discharge pressure (expressed in feet) can be added to the elevation difference to produce the total static head.

Total Static Head (in ft) = (Sdh - Ssh) + (DP - SP) (Eq. 2-2)

Where: Sdh = static discharge head (ft)

Ssh = static suction head (ft)

DP = discharge pressure (ft)

SP = suction pressure (ft)

Using Equation 2-2 and the data provided in Table 2-1 and Figure 2-3:

Total Static Head (at 130°F) = 30 ft + 5470 ft = 5500 ft

Head losses due to friction in the piping, valves, and fittings are called *friction head losses*, and are expressed in feet of liquid pumped necessary to overcome the friction losses caused by flow of the liquid through the system. Entrance losses (due to turbulence) and exit losses (due to the loss of the velocity head at the termination of the discharge line) can be included as friction losses. Friction head losses vary as the square of the flow.

For a representative flow, friction losses can be calculated using friction factors for each component, pipe, valve, heat exchanger, etc., in the system using data from the *Piping Handbook* (Nanyar, 1992) or the *ME Handbook* (Baumeister, 1987). Friction head losses for other flows can then be approximated using the following square relationship:

$$h_{fx} = h_{fo} \times (F_x / F_o)^2 \qquad \text{(Eq. 2-3)}$$

where: F_o = the flow for which the losses are known

F_x = any other flow

h_{fo} = the friction head loss at F_o

h_{fx} = the friction head loss at F_x

For example, if the sum of the component, entrance, and exit friction head losses in a system were calculated to be 50 ft at the design flow rate of 150 gpm, then, by Equation 2-3:

for $F_x = 50$ gpm, $h_{fx} = 50$ ft x $(50 \text{ gpm} / 150 \text{ gpm})^2$; $h_{fx} = 6$ ft loss

and

for $F_x = 500$ gpm, $h_{fx} = 50$ ft x $(500 \text{ gpm} / 150 \text{ gpm})^2$; $h_{fx} = 556$ ft loss

Friction head losses based on the above example can be extrapolated from Figure 2-4.

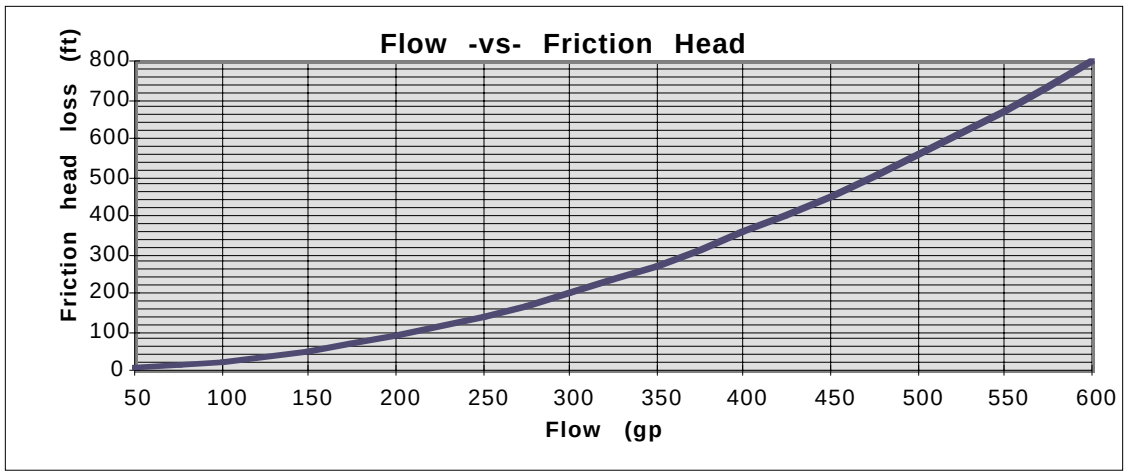


Figure 2-4
Friction Head Losses

The system head is the sum of the total static head and the friction head. Combining the static heads, pressure differences, and friction head losses plotted against the flow results in the system head curve shown in Figure 2-5.

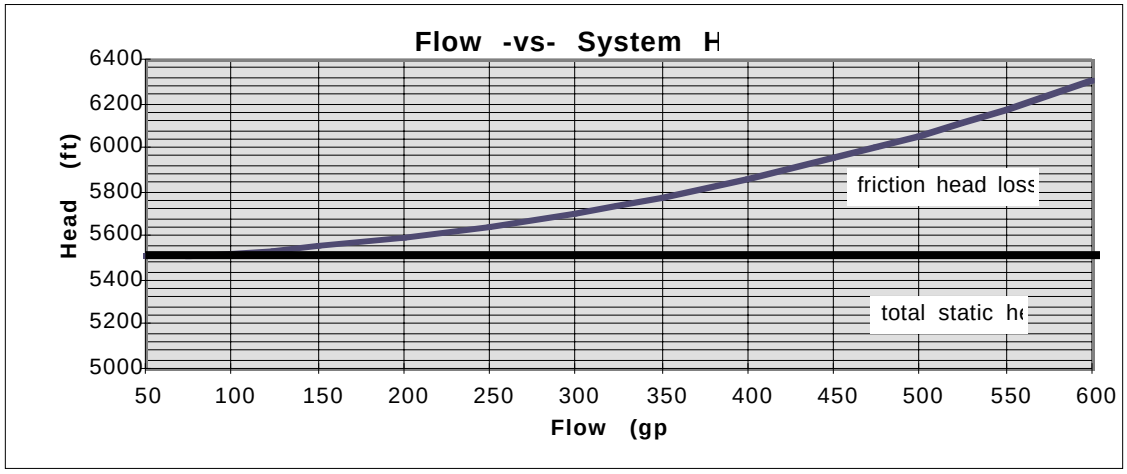


Figure 2-5
System Head Curve

There are five physical factors that can act upon a fluid to affect its behavior. All of the physical actions of fluids in all systems are determined by the relationships of these factors to each other. These factors are:

- Gravity, which acts at all times upon all bodies, regardless of other forces
- Atmospheric pressure, which acts on any part of a system exposed to open air

- Specific applied forces, which might or might not be present, but are entirely independent of the presence or absence of motion
- Inertia, which comes into play when there is a change from rest to motion or vice versa, or when there is a change in direction or in rate of motion
- Friction, or head loss, which is always present whenever there is motion

Any fluid has a velocity above which laminar flow cannot exist in a stable condition. This is the critical velocity. If the velocity of a fluid is slowly increased, a point is reached where turbulence begins. Increased turbulence results in increased head loss. The Darcy-Weisbach pipe-friction equation exists in several forms, but is commonly written as:

$$h_f = f \times (L \div D) \times (V_2 \div 2g)$$

The piping of centrifugal pumps, particularly the suction piping, can seriously affect the operating efficiency and life expectancy of the pump. Poorly designed suction piping can result in the entrainment of air or vapor into the pump and can cause cavitation, which displaces liquid from within the pump casing, results in vibration, and throws the pump out of balance. The cavitation alone can result in severe erosion of the impeller, and the out-of-balance condition can set up a slight eccentric shaft rotation, which eventually wears out the pump bearings and seals in a centrifugal pump. It affects the internal check valve and fluid cylinder life in positive displacement pumps. The suction and discharge piping should be supported independently of the pump so that very little load is transmitted to the pump casing.

As discussed in Chapter 1, the most prevalent pumps installed in U.S. plant charging systems are the Ingersoll-Dresser centrifugal and the Gaulin and Union positive displacement pumps. A brief discussion of the centrifugal charging pump is found in Section 2.1 and the positive displacement pump is discussed in Section 2.2. A brief discussion of the charging pumps found in EdF is in Section 3.

2.1 Ingersoll-Dresser (Pacific Model) Centrifugal Charging Pump

Centrifugal pumps depend entirely on changes in velocity to add energy to the fluid; thus, a discussion on centrifugal pump design becomes a discussion on velocities occurring at various points within the pump. For practical purposes, centrifugal pump design is based on a one-dimensional analysis of velocity. For radial flow pumps, the head (H) produced by the impeller can be expressed as:

$$H = \mathrm{N}\ (U_2V_2 - U_1V_1)/g \qquad \text{(Eq. 2-4)}$$

where H = head (ft)

N = coefficient of hydraulic efficiency (.85 to .95)

U_2 = circumferential velocity of impeller discharge (ft / sec)

V_2 = average absolute fluid velocity at impeller discharge (ft / sec)

U_1 = circumferential velocity of impeller suction (ft / sec)

V_1 = average absolute fluid velocity at impeller suction (ft / sec)

g = 32.174 (ft / sec^2)

Because the rotational component of flow at the impeller inlet is negligible, Equation 2-4 can be reduced to:

$$H = \mathrm{N}\ U_2 V_2 / g \qquad \text{(Eq. 2-5)}$$

Therefore, Equation 2-5 provides a practical method for calculating total head for any centrifugal charging pump used in the industry.

A large portion of the velocity energy developed by the impeller is converted to pressure by a set of stationary diffusion vanes or by a volute designed into the casing surrounding the impeller. An 11-stage centrifugal pump is shown in Figure 2-6.

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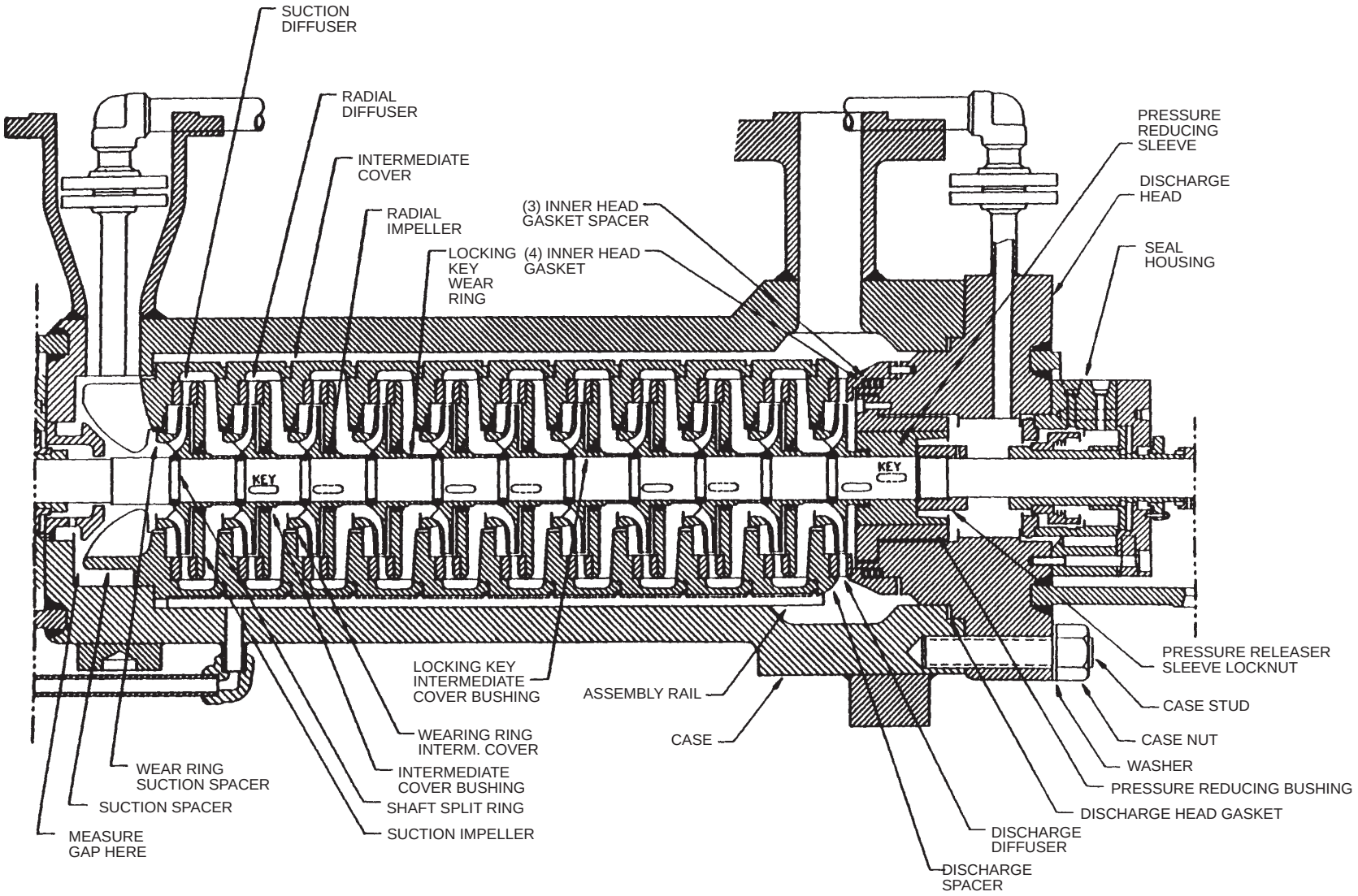


Figure 2-6
Centrifugal Charging Pump (Pacific Model RLIJ 11-Stage)

2.1.1 Impeller

Centrifugal pumps impart forces on the fluid by a set of rotating vanes (the impeller). The rotating impeller causes the fluid to be discharged at its periphery at a higher velocity than at its center. Basically, the impeller is designed to move the fluid at the velocity that would be obtained if the fluid were to fall a distance (in feet) equal to the pump design head.

The principle of dynamic similarity, when applied to centrifugal pumps, expresses the fact that two pumps geometrically similar to each other will have similar performance characteristics. The term *specific speed* is the concept that links the three main parameters of these performance characteristics (capacity, head, and rotation speed) into a single term.

In its basic form, the specific speed is an index number and can be calculated as follows:

$$N_s = \frac{N\,Q^{1/2}}{H^{3/4}}$$

where: N_s = specific speed

N = rotation speed in rpm

Q = capacity in gpm

H = head in feet (head per stage for a multistage pump)

The formula remains unchanged, whether the impeller is the single- or double-suction type. It is customary, therefore, when referring to a definite value of specific speed to mention what type of impeller is in question. While it is possible to calculate the specific speed for any given operating condition of head and capacity, the definition of specific speed assumes that the head and capacity used in the equation refer to those at the best efficiency of the pump. It should be noted that this index number is independent of the rotation speed at which the pump is operating.

The physical characteristics and the general outline of impeller profiles are intimately connected to their respective type-specific speeds. Thus, the value of the specific speed immediately describes the approximate impeller shape. Similarly, the specific speed of a given pump is also definitely reflected in the shape of the pump characteristic curves. While not dimensionless, specific speed is generally expressed simply as a number because its practical application is such that units are of no importance. The practical range of specific speed is approximately 500 to 15,000.

Impeller velocity requirements can be determined by the equations:

$$S = 1/2\ at^2 \qquad \text{(Eq. 2-6)}$$

and

$$V = at \qquad \text{(Eq. 2-7)}$$

In Equation 2-6, fluid falling from a height (*S*) of 4000 feet would reach zero elevation in the time (*t*) of 15.75 seconds due to the gravitational acceleration (*a*) of 32.2 ft / sec^2. Substituting 15.75 seconds for *t* in Equation 2-7, the fluid would achieve a velocity (*V*) of just over 500 ft / sec, and that is the same velocity that an impeller would have to impart to a fluid to pump it to a 4000-foot elevation.

Figure 2-7 shows the velocities that must be attained by impellers when pumping against pressure heads of 0 to 6000 feet. For reference, the Pacific model 2.5" RLIJ-11 stage charging pump is designed for a differential head of 5800 feet (requiring about 600 ft / sec impeller velocity).

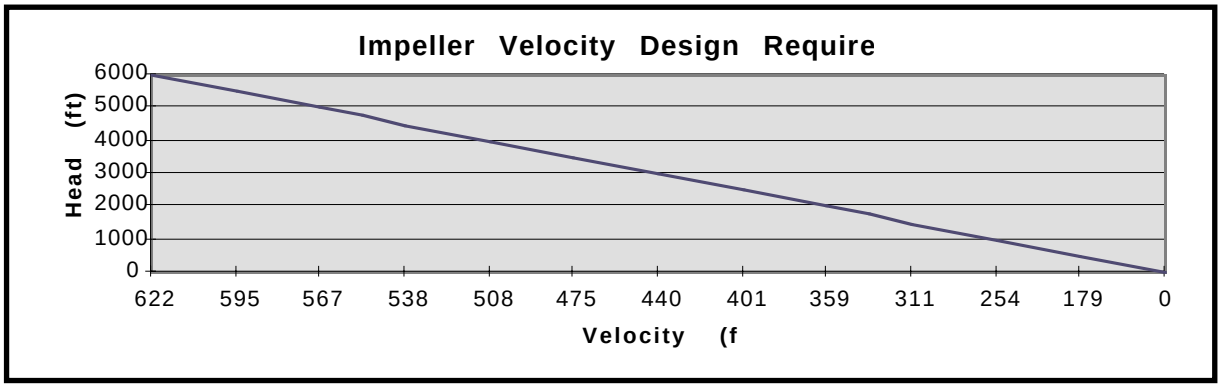


Figure 2-7
Head vs. Impeller Velocity

In Pacific barrel pumps, the impellers are assembled in series. The flow of fluid is from the first stage to the second stage, and so on, to the last stage impeller and through the discharge nozzle. The heads of the impellers are additive, and the total head developed is the sum of the heads developed by each impeller. Thus, it is possible to divide the total head required into stages, using a number of high-efficiency, small-diameter impellers. A single-stage pump could be designed to provide the velocity and pressure characteristics of the multi-stage pump described in Figure 2-8; however, its impeller would have to be three feet in diameter and turn at 3600 rpm.

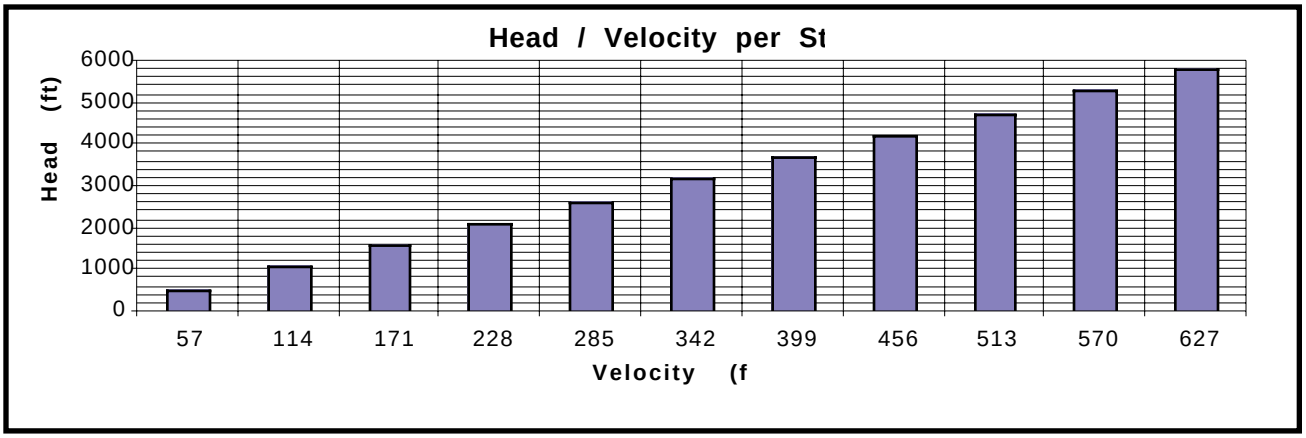


Figure 2-8
Head / Velocity of Fluid Per Stage (11-Stage Pump)

2.1.2 Seals

The purpose of the mechanical seal is to control the leakage from the stuffing box to the atmosphere along the rotating shaft. As the pump rotates, the pressure in the impeller housing is higher than the pressure in the stuffing box. This pressure difference tends to force the liquid toward the low pressure side. It is a common misconception that mechanical seals are designed for zero leakage. Actually, all mechanical seals are designed to leak; however, the leakage is invisible because it evaporates due to heat generation as the fluid passes between the seal faces. The amount of leakage is dependent on seal size, design, pressure, temperature, and the liquid being sealed.

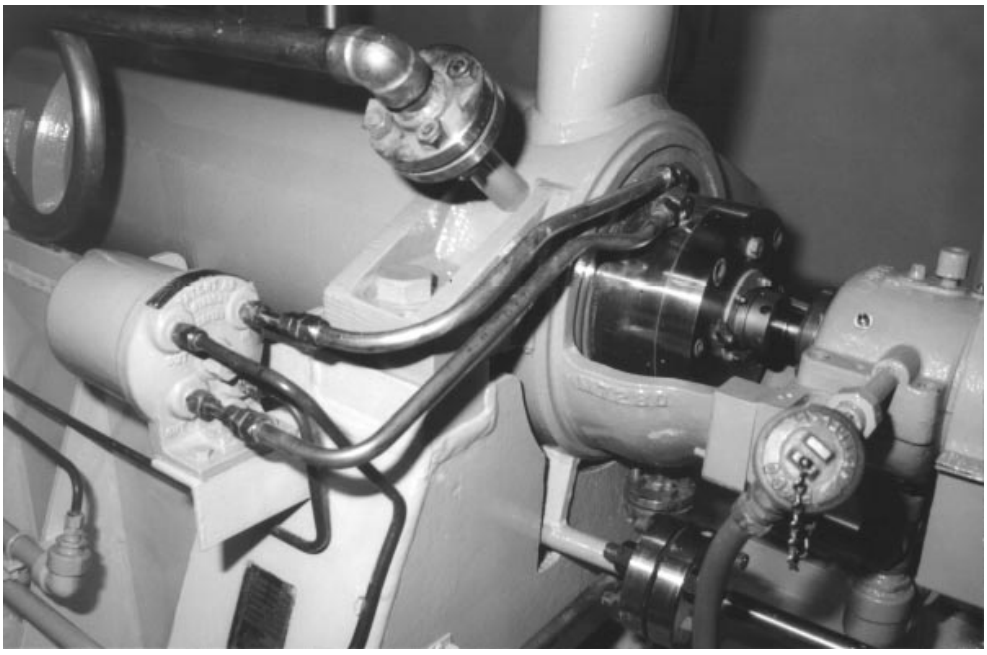


Figure 2-9a
Closeup of Pacific Charging Pump Inboard / Outboard Seals and Bearing Housings

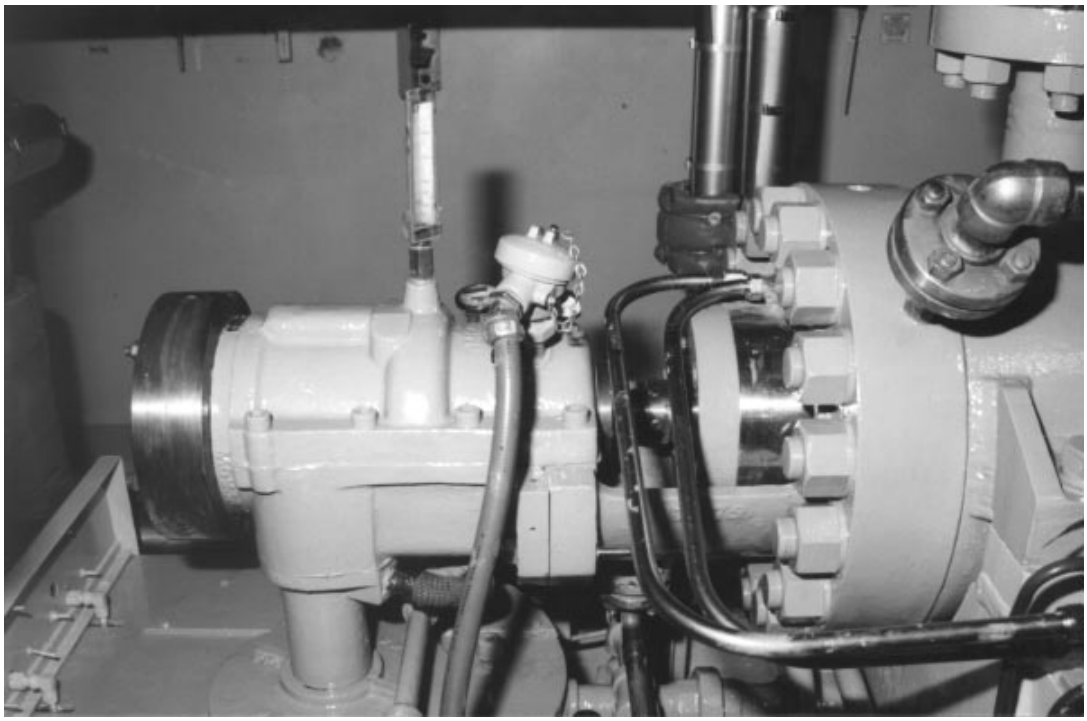


Figure 2-9b
Closeup of Pacific Charging Pump Inboard / Outboard Seals and Bearing Housings



Figure 2-10
Mechanical Seal
(Courtesy John Crane, Inc.)

Although mechanical end face seals vary in design, they operate on the same basic principle. Fluid is to be sealed in a housing through which a rotating shaft extends. The easiest way to understand the concept involved is to visualize a shaft with a shoulder running against a housing wall as shown in Figure 2-11a. The dynamic sealing takes place between the end faces. This shoulder-on-the-shaft approach explains the principle but does not make a practical seal.

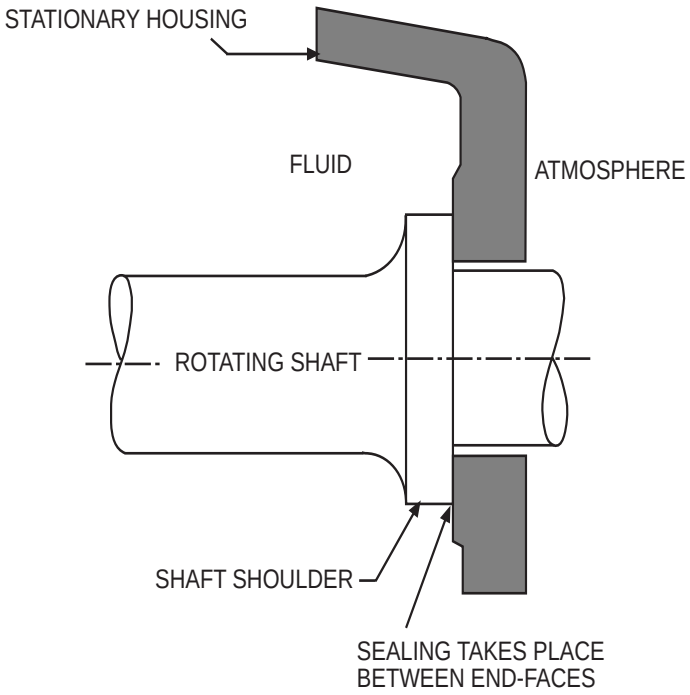


Figure 2-11a
Mechanical Seal Shaft / Housing Interface

To achieve commonality of materials and other operating advantages, the shaft shoulder is replaced by a separate piece, and another separate piece is inserted into the housing wall as shown in Figure 2-11b. These two pieces, the primary ring and the mating ring, are the heart of the mechanical seal.

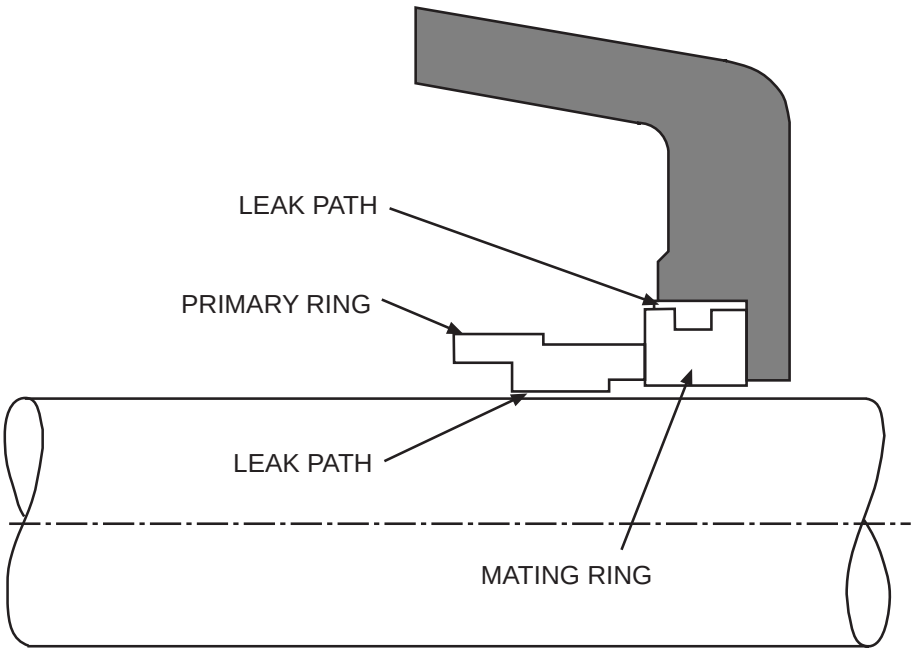


Figure 2-11b
Primary Seal and Mating Ring Interface

The precision lapped faces of the rings are automatically kept in contact by, in addition to the fluid pressure, the insertion of a spring as shown in Figure 2-11c. A single spring or multiple springs can be used.

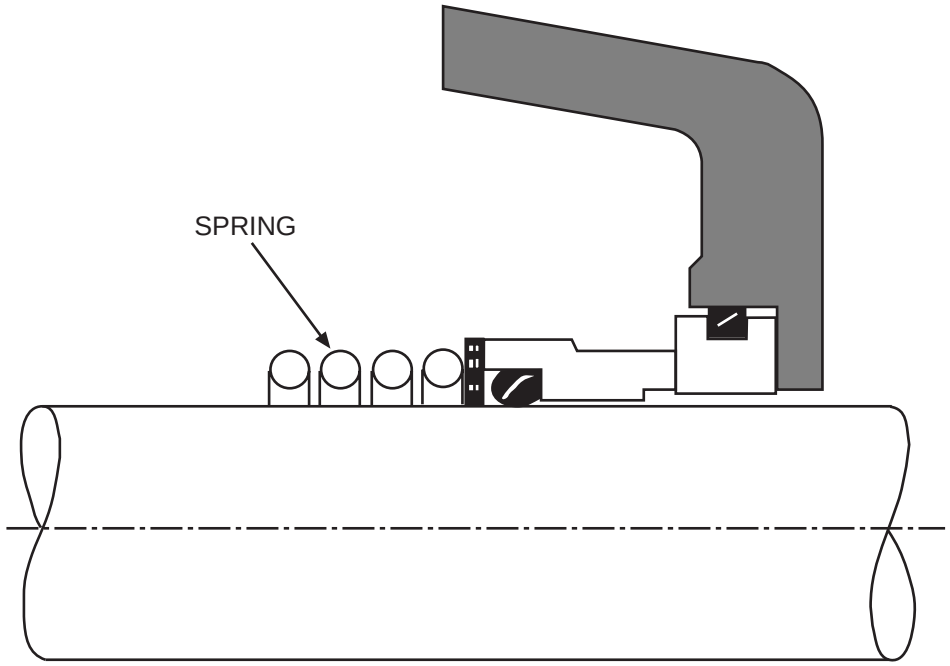


Figure 2-11c
Mechanical Seal Spring

Leakage paths around the rings are sealed with secondary seals such as O-rings, wedges, or bellows, as shown in Figure 2-11d.

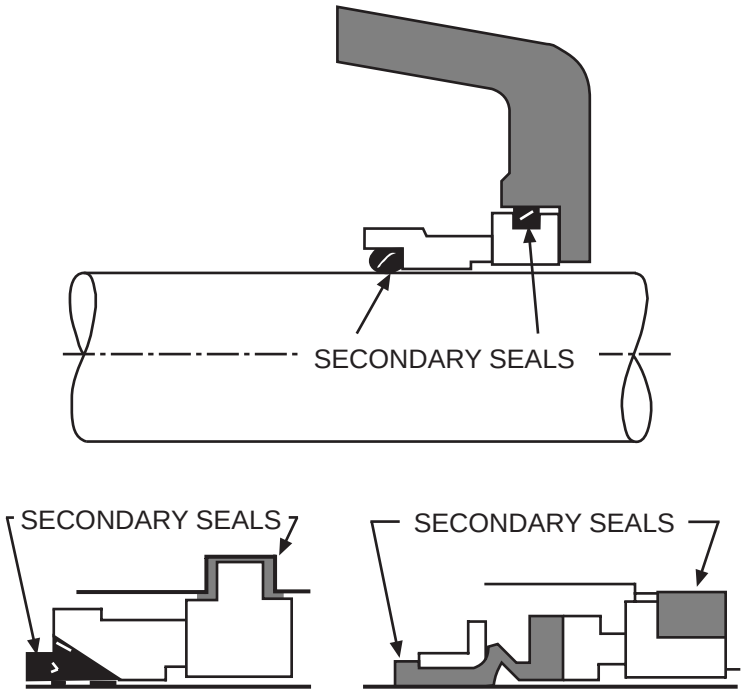


Figure 2-11d
Types of Secondary Seals

In the seal head, the retainer is fixed to the shaft with set screws. The retainer retains the springs, disc snap ring, and primary ring and provides a positive drive to the primary ring. The primary ring is held in the retainer by a dent and groove principle, which also serves as an anti-rotation device. The spring is placed in the spring holes of the retainer and compressed to its working height, which is a predetermined dimension, to provide the correct spring load to the sealing face. Figure 2-11e shows the completed seal assembly.

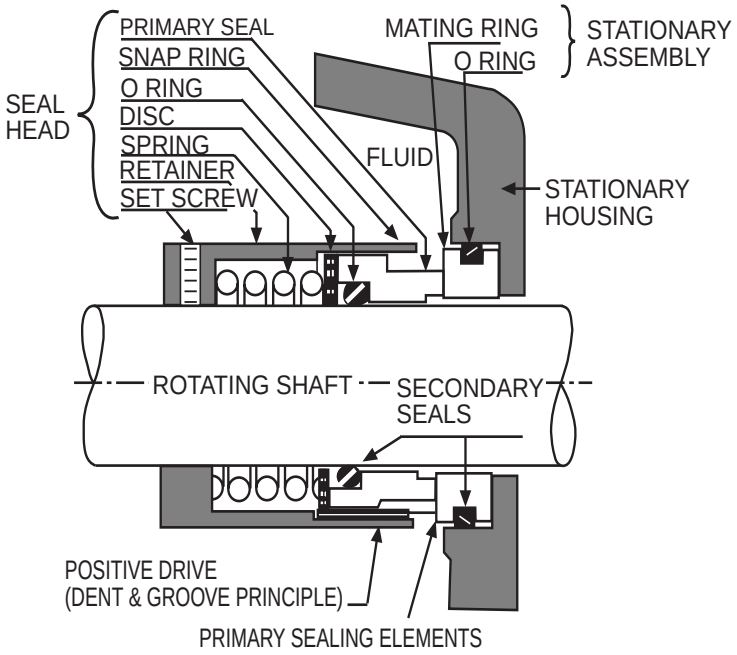


Figure 2-11e
Mechanical Seal Assembly

The seal that is used in the charging pump uses a rubber bellows instead of an O-ring in the seal head, a different type of retainer, a single spring, and a drive band that holds the bellows in place. Figure 2-12 shows the components of the seal.



Figure 2-12
John Crane Mechanical Seal Exploded View
(back, from left) Spacer, Spring, Retainer, Primary Ring
(front, from left) Bellows, Drive Band, Disc, Mating Ring

There are three sealing areas that are common to all basic mechanical seal designs:

- At the mating surface of the primary and mating rings
- Between the stationary component and the gland plate
- Between the rotating component and the shaft or sleeve

In a cartridge seal assembly, where the seal is installed on a sleeve and a gland plate is used, there are two additional sealing points:

- Between the face of the stuffing box and the gland plate
- Between the shaft and sleeve

Other components of the mechanical seal and their functions are listed below:

- Retainer - retains the springs and provides positive drive between the retainer and the primary ring
- Springs - ensures contact between the mating ring and primary ring during startup and shutdown, when there is a lot of end play and / or a lack of liquid pressure
- Disc - ensures equal spring force distribution to the primary ring
- Anti-X-ring - ensures that the O-ring does not extrude between the shaft or sleeve and primary ring
- Snap ring - ensures that the primary ring does not detach from the retainer under spring pressure
- O-ring - performs as a secondary seal to prevent leakage between the primary ring and the sleeve or shaft
- Retainer set screws - ensures positive drive between the retainer and shaft or sleeve and ensures proper positioning of the seal head

Some of the features considered in the design of mechanical seals are seal balance, face pressure, pressure-velocity (PV), power loss, and temperature control.

Seal balance is the ratio of the pressure area closing the seal rings divided by the pressure area opening the seal rings. However, because the effect of balance on the seal cannot be strictly separated from other effects, the optimum balance ratio is a subject of debate. In general, seals with a higher balance are more stable, but have higher wear and be more prone to heat checking. Seals with lower balance ratios wear better, but are more prone to instability and have higher leakage rates. The balance ratio is calculated as follows:

$$b = a_c \div a_o$$

where: b = balance

a_c = hydraulic closing area

a_o = hydraulic opening area or face area

The pressure in the stuffing box acts in all directions and forces the primary ring against the mating ring. The pressure acts on the closing area (a_c) so the closing force on the seal face is

$$F_c = pa_c$$

where: F_c = hydraulic closing force

p = stuffing box pressure

To control the pressure between the seal faces, the ratio between the opening and closing forces can be adjusted. One way is to hold the opening area constant and reduce the closing area by adding a step to the shaft or installing a sleeve on the shaft. All balanced seals are designed with a shoulder, except the metal bellows type seal.

Face pressure is one of the most important factors in the success or failure of a mechanical seal. Hydraulic pressure between the seal faces tends to separate the mating ring and primary ring. This pressure distribution is considered a pressure wedge and is assumed to be linear in all calculations. The actual face pressure (P_f) is the sum of the hydraulic pressure (P_h) and the spring pressure (P_{sp}):

$$P_f = P_h + P_{sp} = \Delta P(b\text{-}k) + P_{sp}$$

where: ΔP = pressure differential across the seal faces

b = seal balance

k = pressure gradient factor, normally = 0.5

P_{sp} = spring pressure

$$P_{sp} = F_{sp} \div a_o$$

where: F_{sp} = seal spring load

a_o = seal face area

Pressure-velocity values were developed by the seal manufacturers as a measure of adhesive wear. As the primary ring rotates against the mating ring, they are affected by the actual face pressure and rotational speed. Pressure-velocity (PV) can be calculated as follows:

$$PV = P_f \, x \, V_m$$

where: P_f = face pressure

V_m = velocity of the mean face diameter

Power loss is the amount of power a seal consumes during startup and continuous operation. It can be estimated by using the following equation for the continuous operation. For startup, this value is approximately five times larger.

$$N_f = (PV) \times f \times a_o$$

where PV = pressure times velocity

f = coefficient of friction between two materials

a_o = area of the smaller contacting face

It is a common problem among pump designers to neglect the amount of power needed to start and run a seal.

Control of the temperature between the seal faces is an important task because the wear is a function of the temperature. Heat between the seal faces causes the liquid to vaporize, and the lack of a liquid film can cause severe problems, such as thermal distortion, accelerated wear, and heat checking of the sealing faces. These problems result in excessive seal leakage.

The best way to avoid flashing problems is to keep the temperature of the sealed product 30–40°F below the vapor point at the faces. Increasing the seal chamber pressure above the vapor pressure normally can accomplish the same thing. Increasing the pressure is accomplished by a bypass line from discharge. Changing temperature can be accomplished by external flush, jacketed seal chamber, bypass from discharge through a heat exchanger, cooled and recirculated seal chamber liquid, quenching, or some form of dual-seal arrangement.

There are three important factors in the seal coolant selection process. The coolant must:

- Be free of abrasives
- Be compatible with the fluid being pumped
- Flow at the calculated rate

The temperature at the seal faces is a function of the heat that is generated by the seal and the heat gain or loss from the flush. The heat generated at the faces from sliding contact is equal to the power loss of the seal. If this amount of heat is removed at the same rate as it is generated, the temperature will not rise. The cooling flow rate can be calculated in gallons per minute (gpm) as follows:

$$gpm = Q \div [\, 500\ (SpHt)\ (SpGr)\ \Delta T\,]$$

where: Q = heat input

500 = conversion factor

$SpHt$ = specific heat of coolant

SpGr = specific gravity of coolant

ΔT = temperature rise

2.1.3 Shaft

The shaft and the shaft bearing assembly are designed for each application with the goal of obtaining a wet critical speed above the operating speed, including reverse rotation conditions (runaway). Pump shaft diameters are usually larger than actually needed to transmit the torque because their size is dictated by the maximum permissible or desirable shaft deflection. This deflection is itself chosen to prevent possible contact at the wearing surfaces while maintaining reasonable clearances that do not affect pump efficiency too unfavorably. The first critical speed of a shaft is related to its deflection.

Pump shafts are usually protected from erosion, corrosion, and wear at the stuffing boxes and leakage joints and in the waterways by renewable sleeves. The most common shaft sleeve function is that of protecting the shaft from wear at a stuffing box.

The shaft is supported at each end by a self-aligning sleeve bearing, and the axial thrust load is carried by a Kingsbury thrust bearing on the discharge end of the pump. The bearings are pressure lubricated by an internal gear-type oil pump driven by a worm gear on the main shaft. A motor-driven gear-type auxiliary oil pump is typically provided for starting and emergency service.

Shaft cracking failures had occurred in 16 applications by the end of 1994. Five other pump shafts have seized (not considered shaft cracking failures). The Westinghouse Owners Group has conducted a study on shaft failures and is in the process of testing and qualifying a new shaft material based on their findings that shaft life can be extended by using material with superior corrosion resistance, lower notch sensitivity, lower FATT, and better fatigue strength. Three materials are under test: Nitronic 50, Custom Age 625 Plus, and the current shaft material, A276 Type 414, 1150°F temper. Pending acceptability of the qualification test results, Ingersoll-Dresser is expected to incorporate new material into shaft production in 1997.

2.1.4 Gear Drive

High-speed gear drives are specifically designed to supply speeds in excess of those that can be obtained economically and safely from ordinary prime movers. The continuous efficient operation of high-speed, totally enclosed gear drives depends on the following factors:

- Adequate mounting
- Proper alignment with the prime mover and driven machine
- Proper lubrication

- Proper cooling
- Proper maintenance

Most gear drives have a priming inlet to allow priming the oil pump before startup to ensure that the bearings and gears are properly lubricated. Some models have a thermometer installed at the priming inlet connection. If this is the case, the thermometer can be temporarily removed to prime the oil pump and then be reinstalled. Ordinarily, the pump does not require manual priming, but in the case of a long shutdown, it is considered good practice to prime the pump before starting the unit. Most models are supplied with a water cooling system to maintain bearing oil temperatures. High-speed gear drives can be operated with bearing temperatures as high as 175°F.

High-speed gear drives typically have a simplified lubrication system, where the gears and bearings are lubricated with the same oil. For uni-directional drives, the lubricating pump is generally mounted on the low-speed end cap and is connected to the low-speed shaft by either a pin-type or flexible coupling. The pump can be connected directly to the low-speed shaft, or it can be connected through a set of gears to obtain proper operating speeds.

The oil in high-speed gear drives serves two purposes. In addition to acting as a lubricant, it also serves as a cooling medium to remove heat from the gears and bearings. Oil spray regulating valves are provided in some models, which proportion the oil between the bearings and gears. Oil pressure to the bearings between 3–10 pounds normally provides satisfactory operation. Due to the pitch line velocities being relatively high, air currents are set up in these types of drives. This tends to “beat” air into the oil, especially at the mesh of the gearing where the oil ejected from the spray head is forced into the gear mesh. Beating air into the oil tends to oxidize the oil more rapidly. It is, therefore, important that the lubricating oils are of high quality and high grade. They must not be corrosive to gears or sleeve bearings; they must be neutral in reaction, have good defoaming properties, have a high demulsibility factor, and have a viscosity index not less than 60. Coolers are provided if necessary to maintain the oil temperature.

There is usually a strainer provided with the gear drive unit that prevents the introduction of foreign material such as metallic chips, grit, and dirt from entering the bearings. To minimize the escape of oil vapor from the gear case and still provide a means of maintaining a low pressure inside the gear case, a large breather is typically included with these drives. Oil vapor, which is developed by heat in the case, in attempting to pass out of the case into the atmosphere must follow a tortuous path through the breather. In so doing, it comes in contact with the large surface of the wire mesh screen in the breather and is cooled and condensed.

Drain-back holes can be provided in the breather in cases where excessive oil condensation is developed to permit the oil to return to the gear case. These holes are not provided where the expected condensation is normal. These breathers are clearly marked for a specific oil level and need to be periodically checked to ensure that the proper oil

level is maintained. If too high an oil level is allowed in the breathers, the escaping vapors are forced to operate against this additional pressure. The vapors follow the path of least resistance, which might be along the shaft, thereby causing oil leakage from the oil seal. Figure 2-13 shows a cutaway view of a common type of gear drive. Figure 2-14 shows an installed Falk speed reducer.

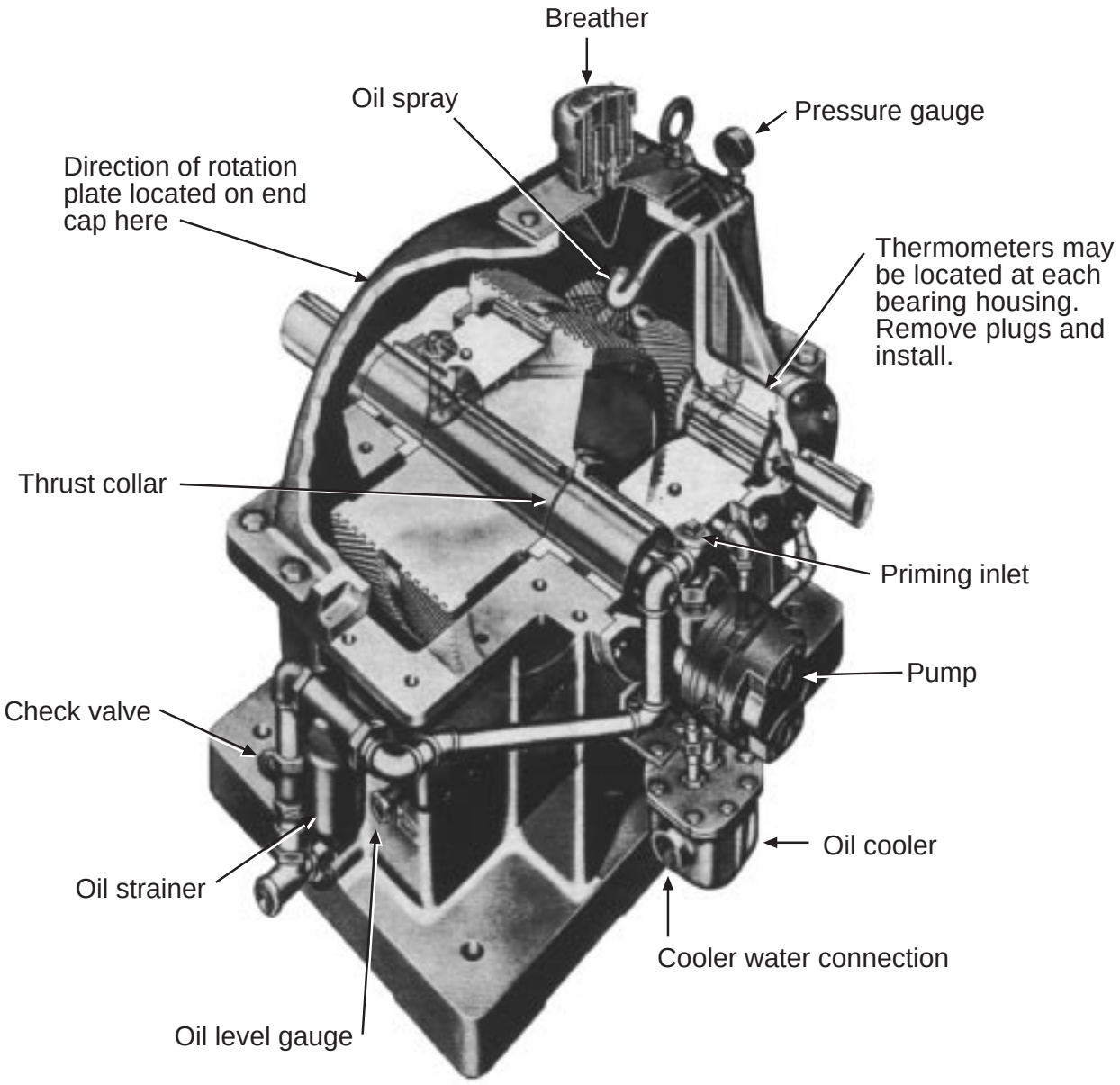


Figure 2-13
Cutaway View of High-Speed Gear Drive

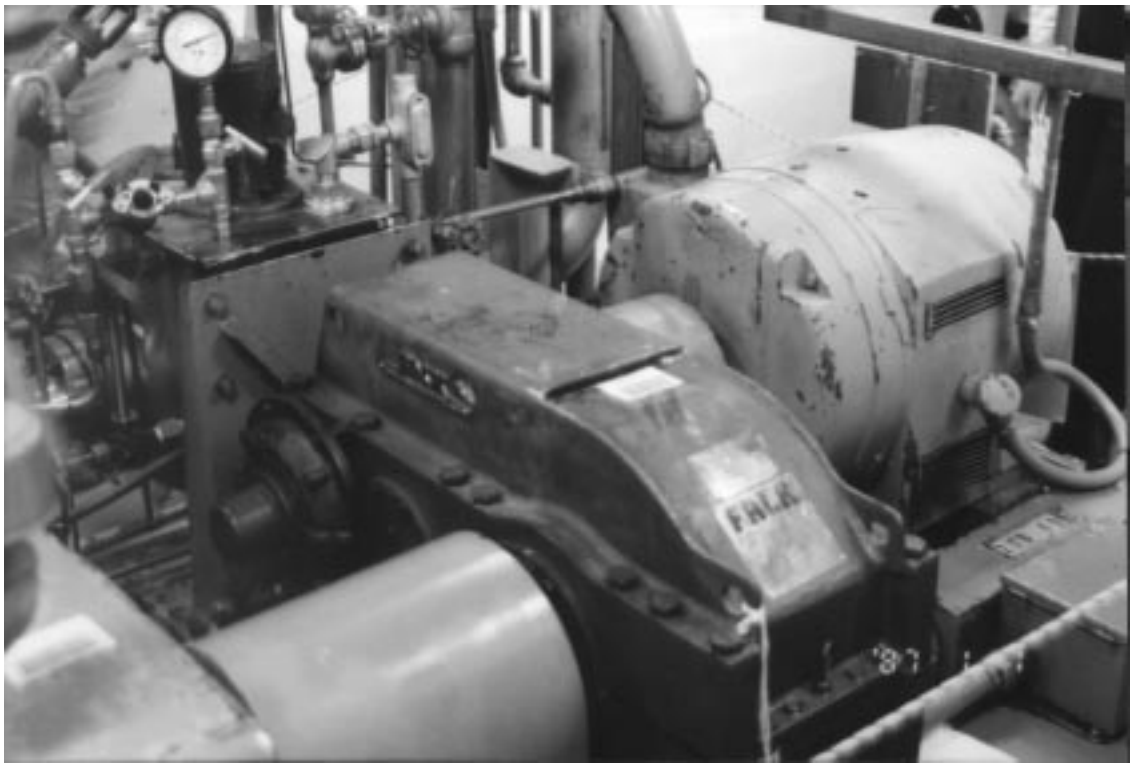


Figure 2-14
Falk Speed Reducer

2.1.5 Diffuser

Pumps designed with stationary vanes are called diffuser pumps. Figure 2-15 shows a diffuser-type centrifugal pump. Diffuser pumps use stationary guide vanes around the impeller to gradually expand the passages for flow to convert velocity energy to pressure head. The main function of diffuser vanes is to lower flow velocity and raise static pressure with as small a loss as possible. The diffuser is usually downstream of the impeller and can be regarded as a flow guidance device fixed to the pump casing. Diffusers have the shape of an axial, semi-axial, radial, or onion-shaped rim of blades. The downstream diffuser deflects the liquid at the exit from the impeller into a swirl-free flow as far as possible and with minimum loss. During this process, absolute velocity in a single-flow line decreases while the static pressure increases. The number of guide vanes is different from the number of impeller vanes to prevent interference vibrations; neither number should be a common multiple of the other.

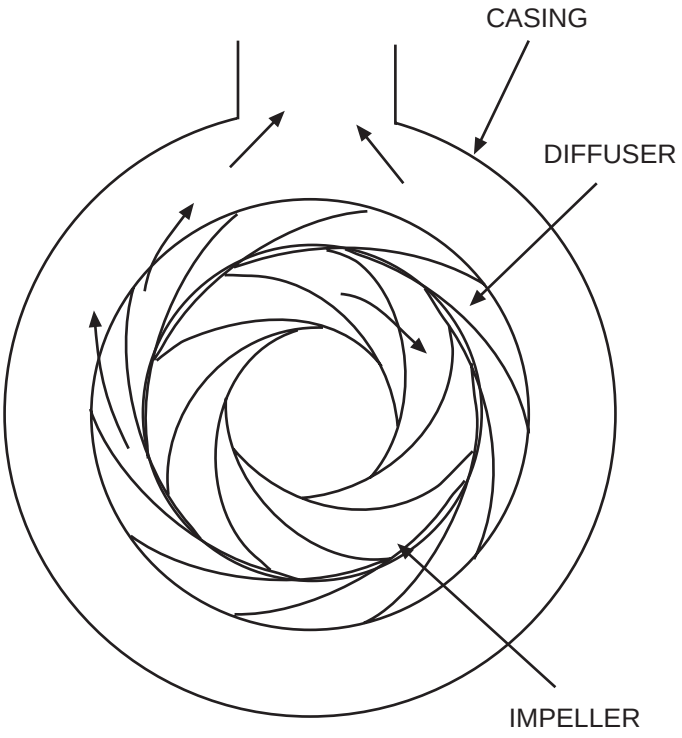


Figure 2-15
Diffuser-Type Centrifugal Pump

2.1.6 Bearings

Sleeve bearings can be made from ferrous and non-ferrous alloys as well as non-metallic materials. Because only the shaft rotates, sleeve bearings are quiet and not subject to endurance fatigue. The following criteria must be considered when selecting a sleeve bearing:

- Speed
- Load
- Temperature
- Operating environment
- Life requirement of the bearing

In addition, special operating conditions such as vibration shock and oscillating loads must also be analyzed.

Babbitted sleeve bearings can provide reasonable and dependable service under adverse chemical or moisture conditions with minimal maintenance. Their soft composition restricts them to light to moderate loads at reasonable temperatures. A babbitt sleeve can sustain a maximum allowable temperature of 150°F. Commercial babbitt material is composed of antimony, copper, tin, and lead. The base content can be either tin or lead and can range from 65–85% of the total material weight. Babbitted sleeve bearings require relatively little maintenance, and most repairs can be made quickly on the job site, usually by rescraping the existing sleeve or pouring a replacement sleeve. They also have the advantage of minimizing the danger of scoring or damaging the support shaft or rotor.

Bronze sleeves are suitable for applications where load conditions and operating temperatures exceed the limitations of babbitt. Bronze sleeves provide greater strength than babbitt and can operate at temperatures up to approximately 300°F. Because bronze is considerably harder than babbitt, it can score or damage the shaft when the wrong grade of lubricant or too little lubricant is used.

A typical journal bearing is shown in Figure 2-16.

The Kingsbury thrust bearing is used to compensate for axial thrust. Kingsbury-type thrust bearings consist of a collar, which is mounted on the shaft, that revolves between one or more sets of babbitt-faced segmental shoes. The backs of these shoes rest against round hardened steel pivots that permit the shoes to assume a tilt and change their angle with respect to the shaft collar.

Bearings in which the thrust is always exerted in the same direction are equipped with shoes on one side only. Bearings used in applications where thrust is exerted in both directions have shoes on both sides of the collar.

Rotation of the shaft collar drags a film of oil into the space between the shoes and the collar, and as the oil forms, the shoes adjust themselves to the angle most efficient for the load conditions and the oil viscosity. Figure 2-17 shows a Kingsbury thrust bearing.

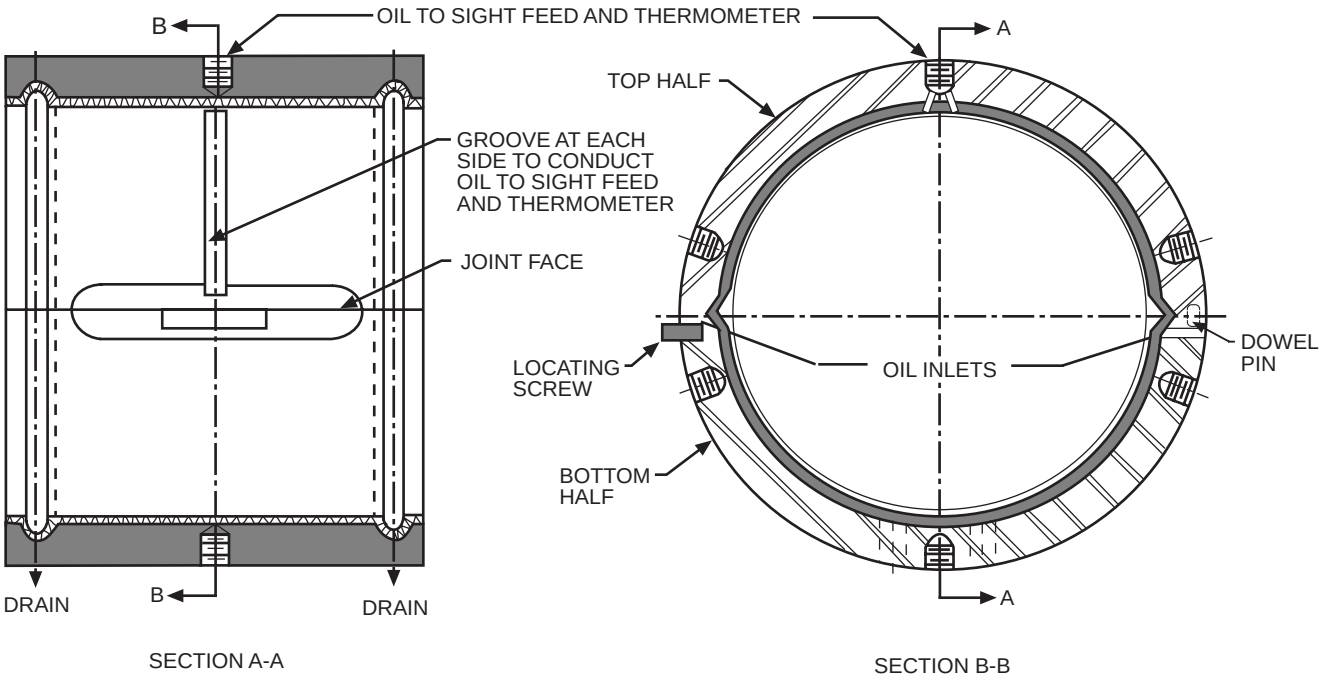


Figure 2-16
Typical Journal Bearing

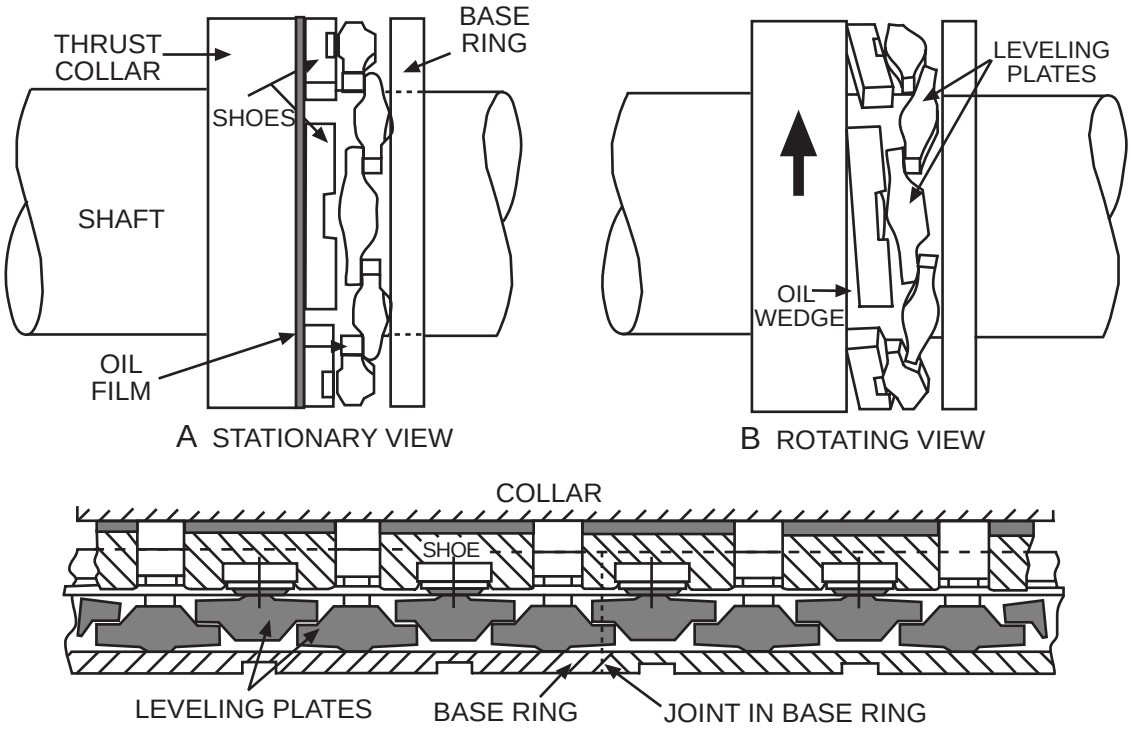


Figure 2-17
Kingsbury Thrust Bearing

2.1.7 Radial and Axial Forces

Hydraulic excitation forces and pressure pulsations created by excessive flow deceleration at partial load can result in failure in a variety of pump components. These forces and pulsations are the result of excessive flow recirculation in the impeller inlet, diffuser, or volute sections of the pump. Typically, some degree of recirculation is present in centrifugal pumps below a specific flow rate and is a minor concern. However, excessive recirculation can be extremely damaging and destructive. For every type of pump, there exists a range of optimum recirculating flow; operating a pump within this range minimizes the risk of pump damage and/or failure. Unfortunately, there is no established method to predict exactly the onset of damaging recirculation.

In multi-stage, high-pressure pumps, radial and axial balance is a must. Unbalanced radial forces react in a plane at right angles to the centerline of the shaft and cause excessive vibration, unnecessary wear, and noise. Unbalanced axial forces react in a plane parallel to the centerline of the shaft and must be controlled to prevent damage to bearings and internal parts of the pump.

There are two types of radial forces that must be compensated for: dynamic and hydraulic. Dynamic unbalance is caused by variations in metal sections and the resultant unequal distribution of weight in castings for impellers. Because the effects of dynamic unbalance increase as the square of the speed, impellers are balanced in special dynamic balancing machines. Hydraulic unbalance is caused by unequal pressure in the liquid around the periphery of the impellers. This creates a radial thrust which, if not balanced, causes serious shaft deflection.

Perfect hydraulic radial balance is obtained with diffusers fitted with equally spaced multiple vanes. Diffusers of this type equalize the pressure at all points about the periphery of the impellers and eliminate radial thrust and the resultant shaft deflection. The axial forces to be balanced are caused by pressure differentials between stages. The proportions of the balancing device are fixed to obtain the desired axial balance. Some models use a balance drum, a balance disk, or some combination of the two to balance the impeller thrust loads.

In models that use a balance drum, the balancing chamber at the back of the last stage impeller is separated from the pump interior by a drum mounted on the shaft. The drum is separated by a small radial clearance from the stationary portion of the balancing device, called the balancing drum head, which is fixed to the pump casing. The balancing chamber is connected either to the pump suction or to the vessel from which the pump takes its suction.

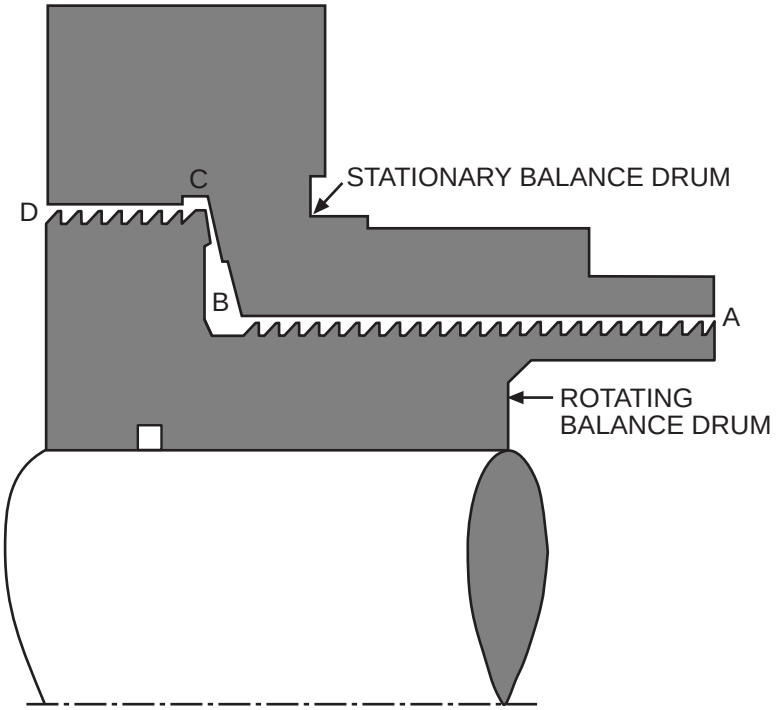


Figure 2-18
Balance Drum Arrangement

In Figure 2-18, water at essentially discharge pressure enters the clearance between the rotating and stationary drums at point A. The water follows a path through the first fixed orifice (A to B), through the variable orifice (B to C), and then through the second fixed orifice (C to D). Chamber D is normally connected by a pipe to the suction source. If a condition of increased impeller thrust toward the suction (A) occurs, the rotor tends to move toward the suction, closing the variable orifice between the drum faces (B to C). By thus reducing the balance flow, the pressure drop between A and B decreases. The resulting greater pressure at B creates an increased thrust in the outboard direction, thereby providing self-compensation for the increased impeller thrust. A similar type of self-compensating balance occurs should the impeller thrust toward the suction decrease. The rotor is free to move axially and, hence, permits the variable gap at the drum faces to match the requirements for hydraulic balance.

The drum diameter can be selected to balance the thrust completely or to balance 90–95% of the thrust, depending on whether a slight thrust load in a specific direction on the thrust bearing is desired.

For models utilizing a balance disk, the rotating disk is separated from the balance disk head by a small axial clearance. The leakage through this clearance flows into the balancing chamber and from there either to the pump suction or the suction vessel. The back of the balance disk is subjected to the balancing chamber backpressure, whereas the disk face experiences a range of pressures. These vary from discharge pressure at its smallest diameter to backpressure at its periphery.

The inner and outer disk diameters are chosen so that the difference between the total force acting on the disk face and that acting on its back balance the impeller axial thrust. If the axial thrust exceeds the thrust acting on the disk during operation, the disk is moved toward the disk head, reducing the clearance. The amount of leakage through this clearance is reduced so that the friction losses in the leakage return line are also reduced, lowering the backpressure in the balancing chamber. This automatically increases the pressure difference acting on the disk and moves it away from the disk head, increasing the clearance. Now the pressure builds up in the balancing chamber, and the disk is again moved toward the disk head until an equilibrium is reached.

To ensure proper balance disk operation, the change in backpressure must be of an appreciable magnitude. This is usually accomplished by introducing a restricting orifice in the leakage return line. A typical balance disk arrangement is shown in Figure 2-19.

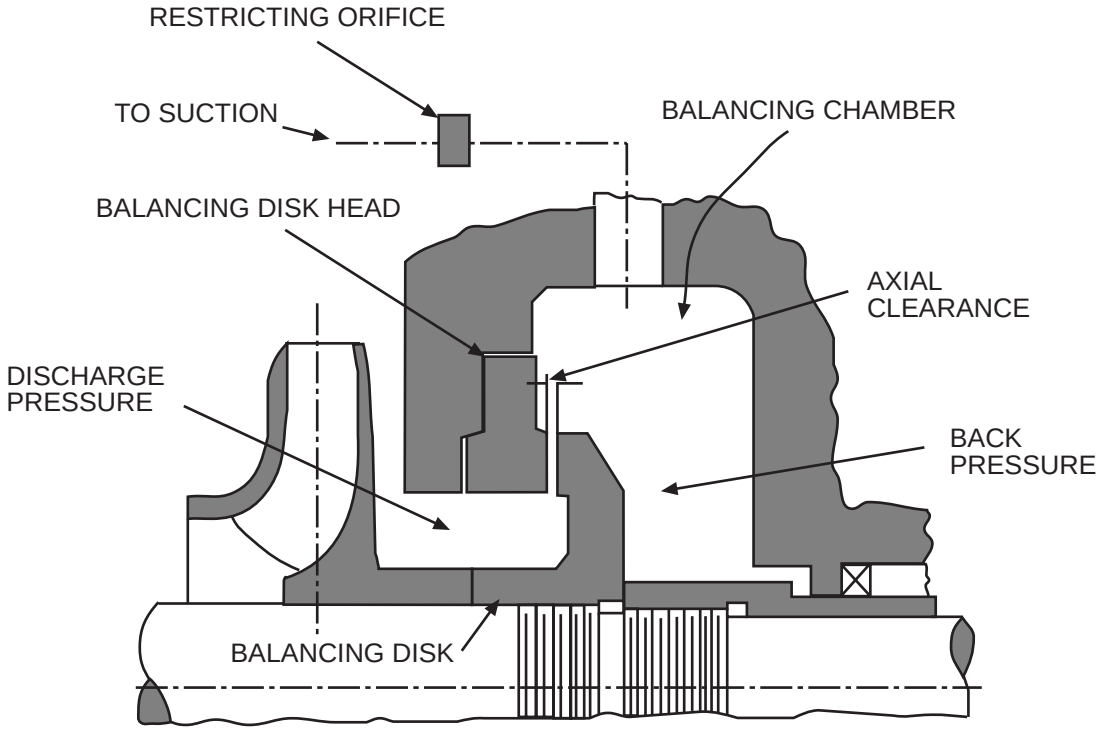


Figure 2-19
Balance Disk Arrangement

2.1.8 Performance and Efficiency

Pump performance is made up of three basic parameters: capacity, head, and speed. Usually, the head-capacity component of pump performance is plotted for a fixed speed. Performance curves also show brake-horsepower (Bhp) requirements and efficiencies. Pumps are selected to operate as close to their best efficiency point (BEP) as possible. Pump performance curves depict (for a given pump speed) the relationship between required capacity and:

- Total head developed by the pump
- Efficiency of the pump
- Horsepower required
- Net positive suction head (NPSH) required

The actual work performed by the pump is the product of the weight of the fluid pumped in a given time period and the head developed while pumping. Referred to as water horsepower (whp), this work can be calculated by:

$$whp = QH[s.g.] \div 3960 \qquad \text{(Eq. 2-8)}$$

where: Q = capacity (in gallons / minute)
H = head (in feet)
s.g. = specific gravity of the fluid

The power required of the pump driver (Bhp) is the water horsepower divided by the pump efficiency η, or:

$$Bhp = QH[s.g.] \div 3960[\eta] \qquad \text{(Eq. 2-9)}$$

For the pump depicted in Figure 2-5, let Q = 300 gpm of water (*s.g.* = 1); then, head = 5000 ft, and by Equation 2-8, *whp* = 379. At a flow of 300 gpm, pump efficiency is 66% (Figure 2-5); and, therefore, by Equation 2-9, the motor must provide 575 Bhp.

For the pump depicted, selected for 4850 rpm with a 150 gpm flow rate at 5800 feet of head, the horsepower required (hp) at the pumping point is 460 Bhp. The NPSH required is approximately 10 feet.

It is a characteristic of centrifugal pumps that as discharge pressure (head) increases, flow and horsepower requirements decrease. Conversely, maximum motor loading and maximum flow occur at minimum head. For example, as shown in Figure 2-20, if the flow were reduced to 75 gpm, head would increase to about 6000 feet. The horsepower requirement would then decrease to about 380 hp.

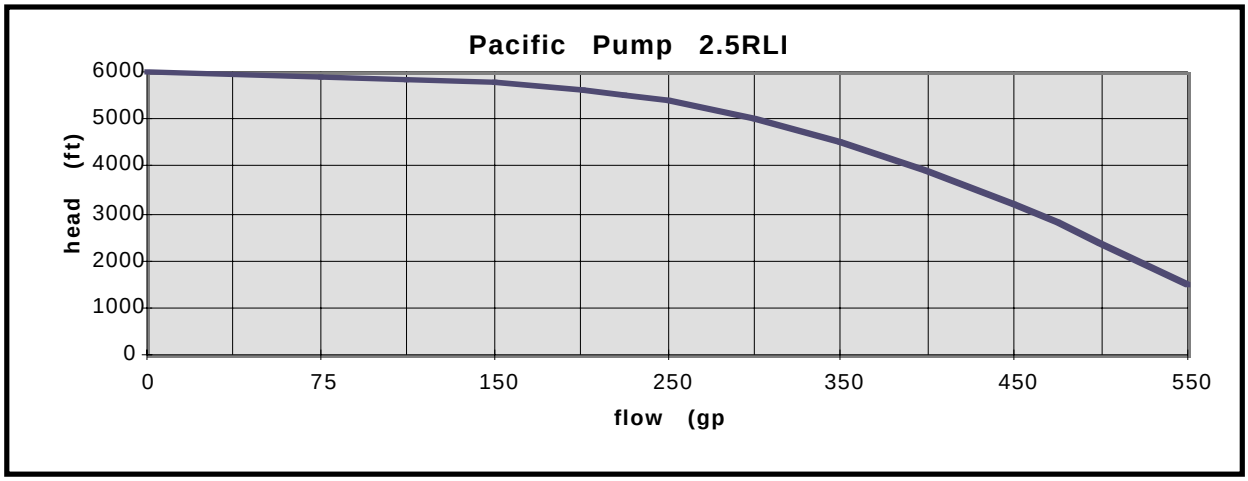


Figure 2-20
Centrifugal Pump Head vs. Flow Curve

2.1.9 Centrifugal Charging Pump Installations: Split Case Designs

The Pacific model 2.5RLIJ centrifugal pump is one of several centrifugal pump designs in use for charging applications. Another type of construction of centrifugal pumps is the double suction, first-stage, axially split case construction. Centrifugal charging pumps manufactured by Byron Jackson include the DVMX and CBMX pump designs. The nine-stage DVMX pump operates at 4600 rpm, and a cross-sectional view is shown in Figure 2-21. This pump design, unlike Pacific's barrel construction, has an axially split case and does not use a balance disk or balance drum for pressure breakdown. The axial hydraulic thrust is balanced by mounting the impellers in an "opposed" configuration. The first stage is a double suction impeller with the remaining eight stages balanced.

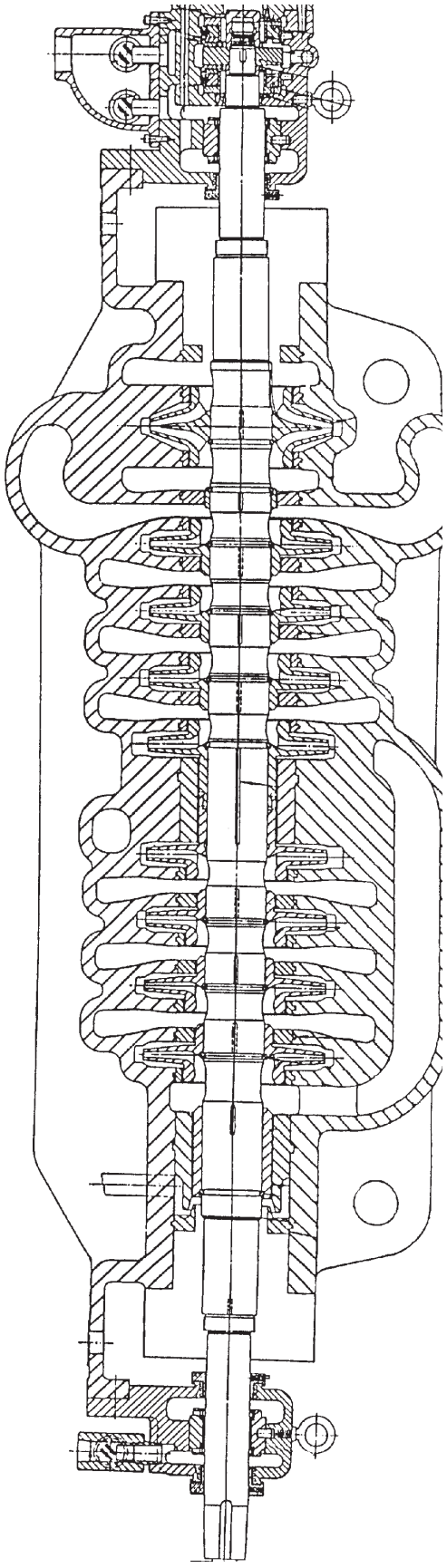


Figure 2-21
Byron Jackson Type-DVMX Axially Split Case

This construction permits the element to be installed without disassembly after the balancing operation. Radial sleeve and pivot shoe thrust bearings ensure long and reliable operation with minimum maintenance requirements. Figure 2-22 shows typical internal construction and assembly of a DVMX-type pump.

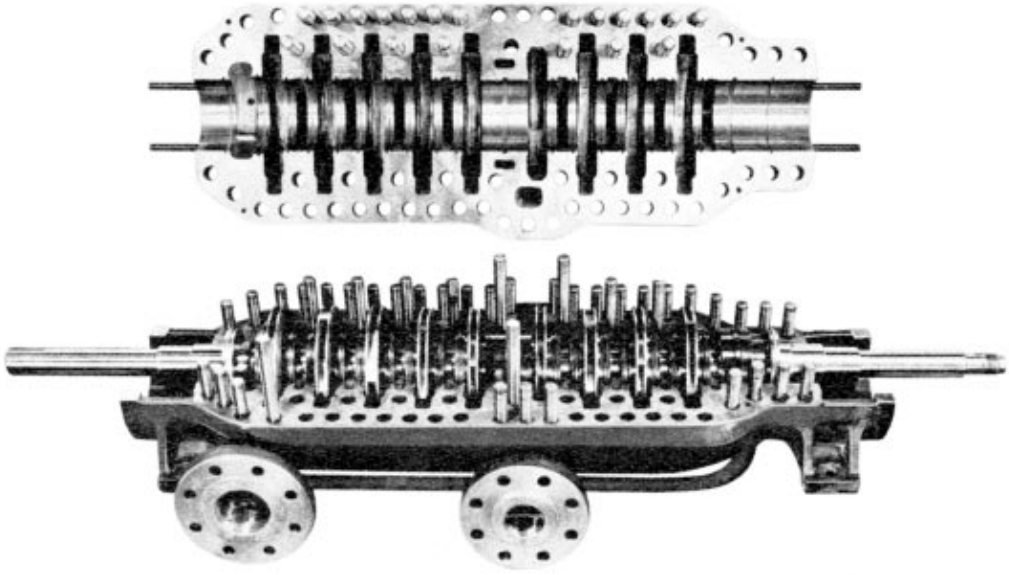


Figure 2-22
Byron Jackson Type DVMX Charging Pump

Higher speed centrifugal charging pumps have also been installed in international plants. The CBMX model operates at 9000 rpm and consists of six stages with overall dimensions smaller than the nine-stage DVMX. Several features are unique to the CBMX. First, the pressure casing is constructed from a block of forged material. Second, each impeller has a separate volute case. The volute cases are split such that the bottom halves can be placed into the bottom half case forging, allowing the balanced rotating element to be installed as an assembly. The top halves of the volutes are next installed on each stage. Figure 2-23 is a sectional of the type CBMX, and Figure 2-24 shows the rotating assembly and the split volute cases.

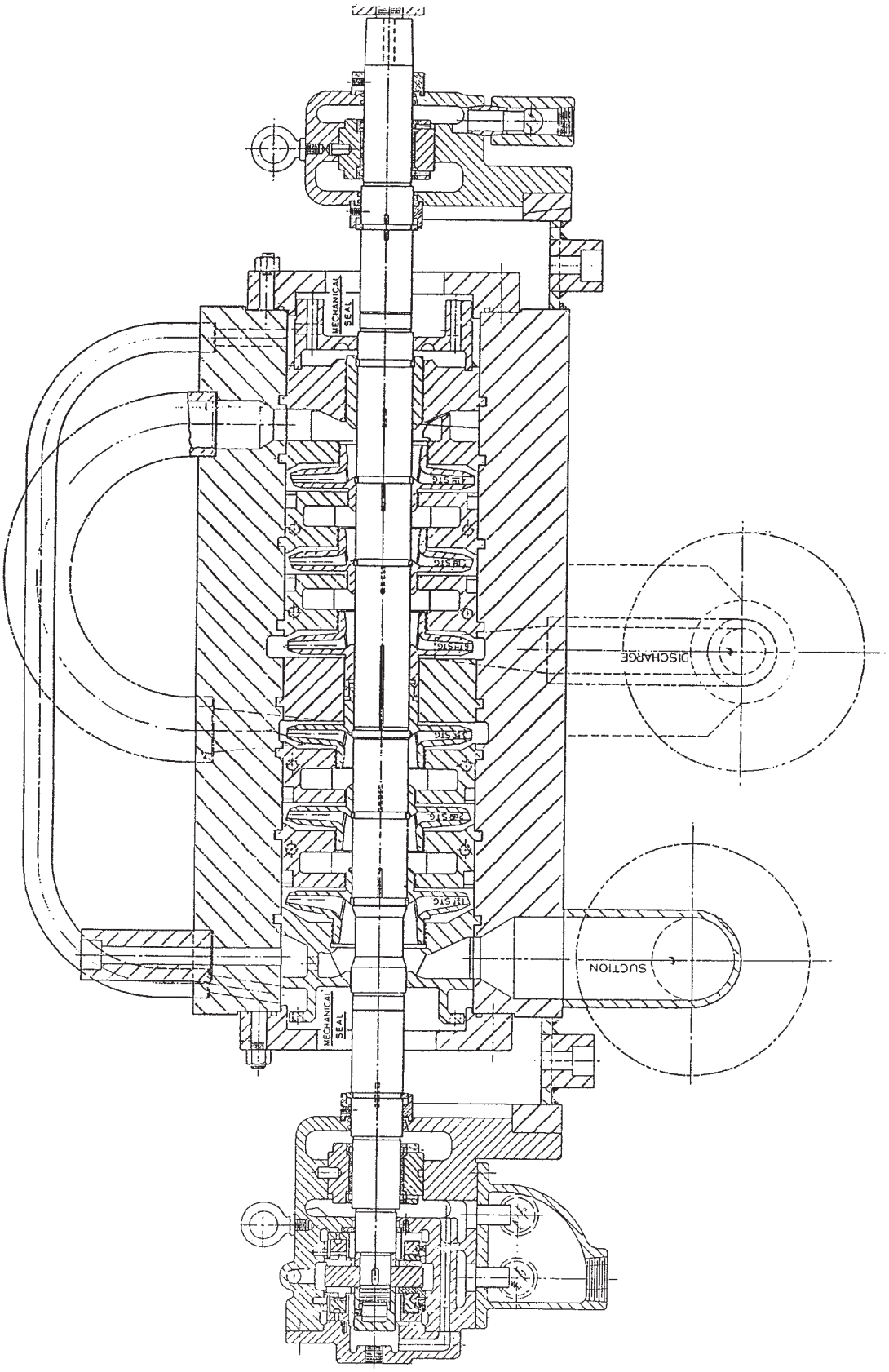


Figure 2-23
Byron Jackson Type CBMX Charging Pump

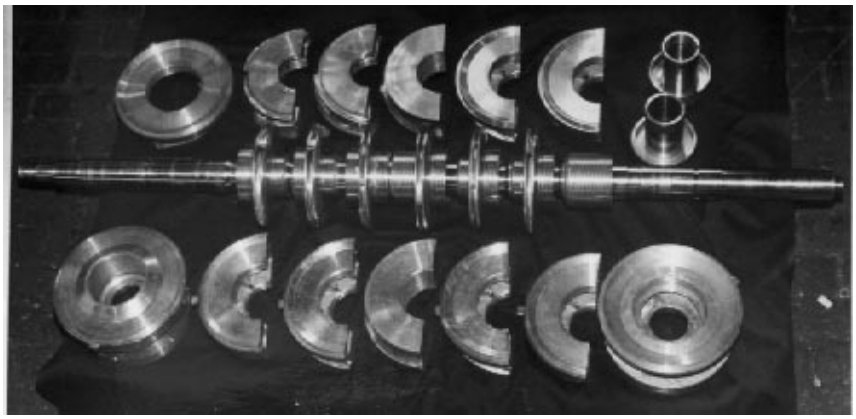


Figure 2-24
Byron Jackson Type CBMX Charging Pump Assembled Rotor
and Stationary Internal Parts

Pumps having the DVMX pump design include ANO Unit 1, Rancho Seco, Surry Units 1 and 2, and Tihangi. The CBMX model has a single installation in Agra, Brazil. Both the DVMX and CBMX use John Crane-supplied mechanical seals, which are self-cooling and require no cooling water.

2.2 Positive Displacement Pump Design and Construction

Positive displacement pumps are designed to pump fluids by moving a trapped volume of fluid against system pressure. Each discharge stroke displaces a definite amount of liquid, regardless of the resistance against which the pump is operating. In the charging pump application, a horizontal triplex (three plunger) or quintuplex (five plunger) pump is used. Some applications utilize a variable-speed design that uses a variable-speed transmission unit between the motor and pump to control output flow rates. The plungers are connected directly to the crossheads, with stuffing boxes connected directly to the fluid cylinders.

Positive displacement pumps in general have good volumetric efficiency (typical values range from 92–96%), with overall mechanical efficiencies in the 85–90% range. They can produce high pressures, produce flow that is directly proportional to speed, and prime themselves, even at very low inlet pressures. Piston-type positive displacement pumps have the highest volumetric efficiency because the leakage path between the piston and cylinder can be closely controlled. A typical horizontal triplex positive displacement pump is shown in Figure 2-25.

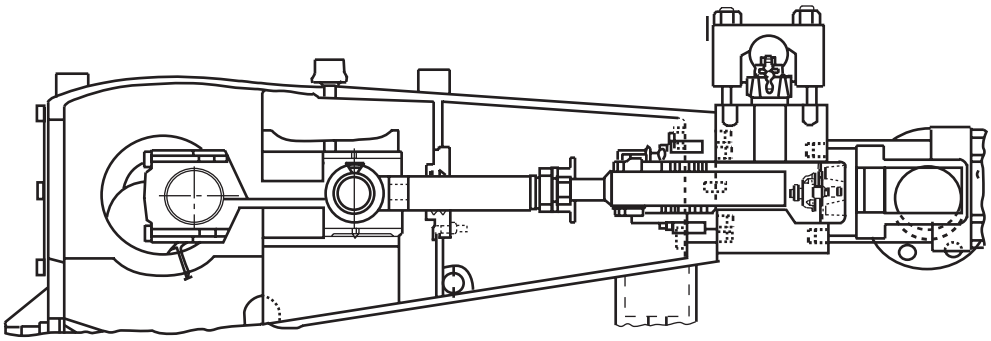


Figure 2-25
Typical Horizontal Triplex Positive Displacement Pump

These types of pumps are comprised of seven key components:

- Fluid cylinder
- Plungers
- Suction and discharge valves
- Packing
- Power end
- Stabilizers
- Desurgers

Plungers fit snugly in the cylinder and are moved back and forth by the crankshaft. The plungers are cooled by the packing cooling system and pumped fluid. Plunger and crosshead assemblies differ from manufacturer to manufacturer.

The design of the typical positive displacement (PD) pump, as shown in Figure 2-25, incorporates a series of intersecting bores. These bores are required to accommodate the suction and discharge flow and the movement of the plunger. The plunger acts as a piston to move the fluid through suction and discharge poppet-type check valves.

Sealing of the fluid in the pump is by large caps on the end bore and top bore, and by packing around the plunger. In all cases, some type of water flow for packing cooling is provided by an external source, either pumped or gravity-fed to the packing, and injected between the primary and secondary packing sets.

A reciprocating pump is, by the very nature of its operation, a cyclic fatigue machine. With each revolution of the pump, each cylinder cycles between suction and discharge pressure, creating one stress cycle on that cylinder. These cyclic stresses range from approximately 40–2300 psi, 3–4 times a second.

Cyclic stress cracks can initiate and propagate in regions where the stress is most severe. Reciprocating pump designs have an inherent weakness in the cylinder block at the intersection between the piston bore and the valve port bores. Most cylinder failures occur with a crack propagating from this location. The expected end of life for a cylinder block is 20,000 hours. Liquid penetrant inspections can be performed when valves are replaced (approximately every 3000–4000 hours of operation) to enable early detection of defects.

Sections 2.2.1 and 2.2.2 describe the major differences between the Gaulin and Union pumps, which comprise the majority of positive displacement pumps in the industry.

2.2.1 Gaulin Positive Displacement Pump

2.2.1.1 Fluid Cylinder

The Gaulin cylinder is a triplex design with an integral stuffing box. Suction and discharge galleys are provided with the suction at the bottom and discharge at the top of the pump. The suction valve is smaller in design and is installed through the top of the pump into the suction valve bore. The discharge valve is larger and is installed after the suction valve is installed. The stuffing box is provided with at least a primary and secondary packing arrangement. The seal water is injected between the primary and secondary packing sets. Some Gaulin designs incorporate a tertiary set that provides for packing leak-off collection between the second and third packing sets. The secondary packing leakage is collected in the plunger well and drained off to a plant drain. Pressure boundary integrity on the plunger and valve bores is provided by stainless steel caps, secured with two studs and nuts on each.

New style blocks made of 17-4 PH high-strength stainless steel have resolved the early fatigue failure problem experienced by a number of the old style blocks (316 stainless steel). In addition to the material change, the new blocks are shot peened for added strength and have larger radii on all corners. Some new blocks also have larger seal water ports that provide more cooling for the packing.

A comparison between the old style blocks and the new style 17-4 PH blocks used in one plant is shown in Table 2-2 below.

Table 2-2
Comparison of Existing Block and Replacement Block Material Properties

Material Properties at 250°F	Existing Block	Replacement Block
Material	Austenitic stainless steel A-351CF-8M (cast) or A-182F316 (forged)	17-4 PH SA-564-630 (Forged)
Yield Strength	24.55 KSI	104.10 KSI
Allowable Stress Intensity	20.00 KSI	45.00 KSI
Ultimate Tensile Strength	69.30 KSI	135.00 KSI
Thermal Expansion Coefficient	8.8 x 10E-6 m/m-°F	5.9 x 10E-6 m/m-°F
Modulus of Elasticity	27.3 x 10^6 psi	27.3 x 10^6 psi
Knuckle Radius of Critical Stress Locations	Existing block	Replacement block
Intersection of Plunger and Valve Bores	Sharp corner	1/4"r
Plunger Bore Entrance	Sharp corner	1/4"r, shot peened
Change in Fit-up	Machined bolt through flanges at suction/discharge	Machined flange faces integral to block

If it is assumed that block strength is directly proportional to the allowable stress, then the replacement block is more than twice as strong as the existing block. The higher tensile strength of the replacement block is indicative of a higher endurance limit. The lower coefficient of thermal expansion means less thermally induced stress at equal operating temperatures.

The machined radii at the intersection of the plunger and valve bores and the plunger bore entrance reduce the peak stress levels as well as the fatigue usage factor. The change in the suction and discharge flange configuration results in a heavier, thicker block. The additional stress on the block mounting studs, due to the increased weight, was determined to be insignificant compared to the preload stress.

2.2.1.2 Plungers

The plunger adapter is connected to the plunger via a threaded engagement. Alignment of the plunger is controlled by the crosshead assembly in the power end and a bearing in the packing. Recent plunger changeouts at some plants have used plungers coated with Colmonoy 72 over a 300 series base material that is subsequently titanium nitrided, resulting in a gold-anodized-type appearance to the plunger. This material treatment results in an extraordinarily hard plunger surface that has resulted in fewer changeouts of both plungers and packing. Experience has shown that these plungers can continue to be used, even if the gold finish has worn away, as long as the plunger still meets surface finish and critical dimension criteria.

API 674 calls for a minimum plunger hardness of Rockwell C 35. Colmonoy coatings typically range from the upper 50s to the low 60s on the Rc scale. Titanium-nitrided surfaces are generally considered “off the scale” but are sometimes quoted as approximately Rc 90. Packing suppliers have long stated that packing life is enhanced by using the hardest possible plungers. Common plunger coatings found in this application are Colmonoy 6, Colmonoy 72, or ceramic coatings like chrome oxide. The original plungers have a Colmonoy 6 or a Colmonoy 72 surface coating.

Surface finish, if not specified locally, should be 16 RMS or better for the uncoated or Colmonoy-style coatings and 20 RMS or better for the ceramic-style coatings. The slight roughness of the ceramic-style coatings is due to the porous nature of ceramics. While the ceramic coatings are brittle by nature and difficult to handle, the porous nature is thought to enhance packing life by carrying trace amounts of cooling water to each part of the packing set.

2.2.1.3 Valves

The typical valves consist of two springs and a valve disk. Disk alignment is controlled by a center bearing or guiding surface. The valve seals on both the inner and outer diameters. Nominal opening pressure is 5 psid. The valves are maintained in the bore by interference fit. Installation is accomplished by a shrink fit into the bore. The valves should not bottom-out on the machined lip in the valve bore.

2.2.1.4 Packing

Packing is comprised of at least two sets, primary and secondary. Some Gaulin pumps have a third set of packing. Section 2.2.6 addresses packing in detail.

2.2.1.5 Power End

The power end converts the rotary motion of the speed reducer or motor to a reciprocating motion via an eccentric shaft, connecting rods, crosshead and plunger adapters. It utilizes two main bearings.

The connecting rod subassemblies consist of:

- Two halves of the connecting rod
- A set of two steel-backed babbitted bearing halves
- Connecting rod bolting
- Bearing ball bolt
- Locking set screws

The connecting rods are assembled to the crossheads by inserting the bearing ball bolt into the crosshead and running up the crosshead adjusting nut. The spanner nut is locked in place using two set screws. Lubrication is provided with a pressure-feed system from an eccentric shaft-driven externally mounted gear-type pump. A externally mounted adjustable relief valve is provided to maintain system pressure. It also has an externally mounted filter. The crossheads, connecting rod bearings, main bearings, and crosshead bearing ball bolt are all pressure fed.

2.2.2 Union Positive Displacement Pump

2.2.2.1 Fluid Cylinder

Union cylinders used are both triple and quintuplex designs with removable and interchangeable stuffing boxes. Suction and discharge galleys are provided with the suction on the bottom and the discharge on the top. The suction and discharge valves are identical and interchangeable. The stuffing box is provided with both primary and secondary packing sets. Seal water is circulated between the primary and secondary packing sets. The secondary packing leakage is collected in the plunger well and drained off to the plant drain. Pressure boundary integrity is maintained using stainless steel cylinder head and discharge valve cover extensions retained with strongbacks and four studs and nuts.

Most installations use a 304 stainless steel forged block, but these blocks have exhibited susceptibility to fatigue cracking. They now available in 17-4 PH and are currently installed in new plants overseas.

A comparison between the old style blocks and the new style 17-4 PH blocks used in one plant is shown in Table 2-2.

If it is assumed that block strength is directly proportional to the allowable stress, then the replacement block is more than twice as strong as the existing block. The higher tensile strength of the replacement block is indicative of a higher endurance limit. The lower coefficient of thermal expansion means less thermally induced stress at equal operating temperatures.

The machined radii at the intersection of the plunger and valve bores and the plunger bore entrance reduce the peak stress levels as well as the fatigue usage factor. The change in the suction and discharge flange configuration results in a heavier, thicker block. The additional stress on the block mounting studs, due to the increased weight, was determined to be insignificant compared to the preload stress.

2.2.2.2 Plungers

The reciprocating charging pumps use a plunger (see Figure 2-26) with a grooved end that accepts two halves of a split retaining ring. The split ring is trapped in the plunger groove by a special threaded retainer that is threaded onto the male end of the cross-head stub. Plunger and stub alignment is ensured by a hardened pin that is inserted into the ends of the crosshead stub and the plunger.

Plungers and crossheads must be handled carefully during maintenance to prevent damage. Great care is taken during the manufacturing of the parts to ensure that the ends of the plungers and crosshead stubs are square and free of burrs or other discontinuities that could affect plunger alignment. Any damage that affects the plunger and crosshead stub alignment could result in failure of either component due to high side loading generated during the compression stroke.



Figure 2-26
Cylinder and Plunger

2.2.2.3 Power End

Drivers vary depending on the style of pump. Many of the quintuplex pumps are driven by a Westinghouse 200 hp, 1800 rpm, 460V, three-phase, 60 Hz motor that is connected to a variable-speed fluid drive that transmits torque to the pump input shaft. These pumps have additional internal reduction herringbone gearing that results in an appropriate pump speed range. Power for triplex pumps varies from 100–125 input horsepower. The Gaulin constant speed pumps use a separate speed reducer. Union

pumps use an internal herringbone gear and fluid drive in Westinghouse plants. Combustion Engineering plants have a motor connected directly to the pump with internal speed reduction gearing.

2.2.2.4 Connecting Rods

The connecting rod subassemblies consist of:

- The two halves of the connecting rod
- A set of two steel or bronze-backed babbitted bearing halves
- Shims
- Connecting rod bolting
- Wrist pin
- Locking set screws

The connecting rods are assembled to the crossheads (see Figure 2-27) by inserting the wrist pin through the crosshead and connecting rod sleeve bearing. The wrist pin is locked in place using back-to-back half-dog locking set screws.

The connecting rod bearing halves (see Figure 2-28) are anti-rotated using extended shims that are placed between the connecting rod and cap. Because of the relatively slow speed that is typical of reciprocating pumps, standard calculations for bearing clearances do not apply. Bearing clearances are established through careful consultation with bearing manufacturers.



Figure 2-27
Crosshead Assembly: Cylinder Body and Stub Extension



Figure 2-28
Main Bearing Cap

2.2.2.5 Driveshaft

The driveshaft used in Union pumps, QX-300, is a one-piece forged and eccentric ground crankshaft. Gaulin pumps use a multiple-piece crankshaft that consists of a straight shaft with eccentric cams that are pinned at specific clock angles to drive the crosshead assemblies in their reciprocating motion. The drive mechanism includes the eccentric shaft, connecting rods, and crossheads. The drive components are lubricated by oil under pressure.

Power supplied by the electric motor is transferred to the pump end by a parallel-shaft, double-reduction, gear-driven speed reducer, or through a variable-speed fluid drive transmission.

A typical speed reducer uses roller bearings and a “no leak” magnum seal. The inner cage of the seal has close fitting clearances between the shaft and the bearing race, which prevent oil from leaving the reducer. The oil drainback ensures an effective seal by returning any oil that gets beyond the bush seal back into the unit. Because there are no moving parts, this throttling bush seal arrangement lasts as long as the reducer, if properly maintained.

An outer cover assembly, when filled with grease, prevents the introduction of foreign particles by trapping them before they enter the reducer. Periodically introducing fresh grease through the fitting in the cover purges contaminants.

Continuous splash lubrication for the speed reducer is provided through the use of a large oil reservoir. Oil is carried to the working faces of the gear teeth through the oil troughs in the reservoir and then is passed to the bearings.

Positive displacement plunger-type pumps are designed with a plunger load upper limit. The plunger load calculation is based on the system pressure acting on the plunger cross-sectional area. The combination of loads generated by a three- or five-plunger pump is reacted by the powered bearing system.

2.2.3 Suction and Discharge Valves

The cylinder and plunger combination must operate with a set of valves to let water in on the intake stroke and let it out on the pressure stroke. These pumps are single-acting, meaning that each cylinder has only one set of suction and discharge valves. The valves can be either a flat-plate double-ported type or a wing-guided single-ported style.

Valve dynamic effects can cause large deviations in the pressures acting on the working parts of a pump. The typical valve consists of a spring and a valve plate that seals against a seat. An overpressure occurs once per cycle, producing dynamic stresses on pump components. Because the pressure required to open the valve is controlled by the spring forces and the differential pressure across it, the valve design can influence the magnitude of overpressure. The forces acting on a closed valve that can result in over-pressure are described by the following equations and are visually represented in Figure 2-29.

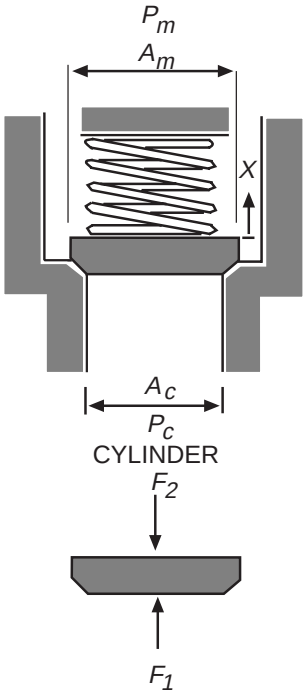


Figure 2-29
Forces Acting on a Closed Valve

Spring force: $F_s = F_{preload}$

Pressure forces: $F_1 = P_c A_c$ (in cylinder) $F_2 = P_m A_m + F_{preload}$ (in manifold)

The valve will open when:

$F_1 > F_2$ or

$P_c A_c > P_m A_m + F_{preload}$

$P_c > P_m (A_m / A_c) + F_{preload} / A_c$

where: A_c = the area of the valve in contact with the cylinder pressure

A_m = the area of the valve in contact with the manifold pressure

For the valve to open, the cylinder pressure must be greater than the manifold pressure by the ratio of the areas A_m / A_c plus an additional factor to overcome the spring force. The area ratio A_m / A_c is typically 1.1–1.5 to yield a seating area that is sufficient to control valve impact stresses.

In addition to the spring and the mass and differential pressure effects, there is a "stiction" effect that opposes the separation of the two lubricated flat surfaces. This stiction force is influenced by the initial fluid film thickness, the viscosity of the fluid, and the geometry of the surfaces. This stiction results in "overpressure" spikes on the discharge valves and "underpressure" spikes on the suction valves. Large overpressure spikes can cause various problems, such as working barrel failure, crosshead guide failure, bearing damage, crankshaft/connecting rod failure, and reduced valve life. Studies have shown that stiction and the differential pressure required to operate the valves increase with time.

Another phenomenon related to valve performance is valve lag. The time required for a valve (suction or discharge) to return fully from a fully open position to its seat is dependent on the valve mass and the return spring properties. Therefore, as the speed of a pump increases, the fixed finite time required for valve closure to occur results in a greater relative valve lag in relation to crank rotation. The lag of the discharge valve closing can actually cause backflow through the discharge valve. Likewise, a lag in the suction valve can allow backflow through the valve. Figure 2-30 shows a typical suction and discharge valve cross section for one cylinder.

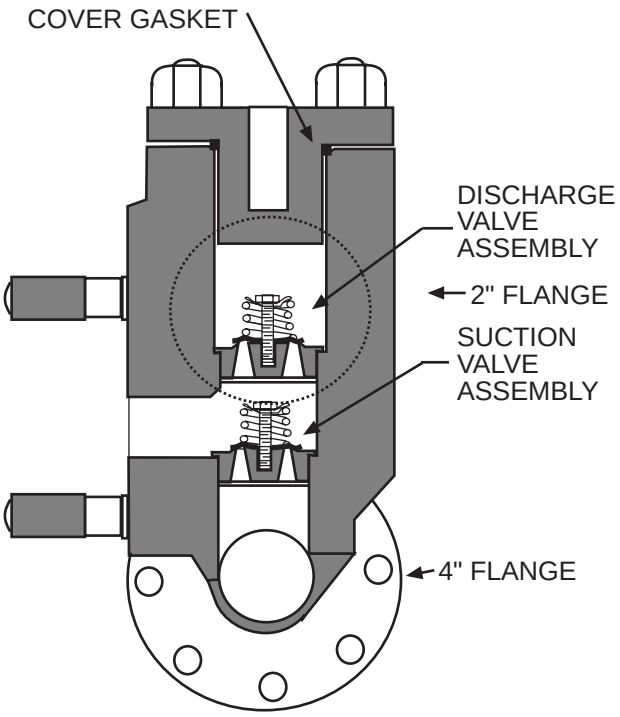


Figure 2-30
Typical Valve Cross Section

Valve replacement is indicated when signs of excessive internal leakage are identified. With flat plate valves, fluid cylinder and valve pressure/flow dynamics tend to push the plates off-center, causing wear of the plate ID against the valve guide. Over a long period of time, this wear prevents the valve from providing a leak-proof seal when closed. As a result, pump volumetric efficiency decreases. Some sites have taken measurements of a pump's flow rate versus rpm after a valve replacement to establish a "like new" baseline for volumetric efficiency. Additional measurements of volumetric efficiency are taken over the months that follow to compare to the baseline. When a decrease in volumetric efficiency is noted, the pump is scheduled for inspection and possible valve replacement.

Ultrasonic detection is one method to determine valve leakage. Additionally, it can be used to determine bladder leakage on the suction and discharge dampeners, cavitation in the pump, and bearing noises, as well as to pinpoint many other leaking components. All operating equipment and most leakage problems produce a broad range of sound. The high-frequency ultrasonic components of these sounds are extremely short wave in nature, and a short wave signal tends to be fairly directional. It is, therefore, easy to isolate these signals from background noises and to detect their exact location.

Airborne ultrasonic transducers are relatively simple to use. They consist of a basic hand-held unit with headphones, a meter, a sensitivity adjustment, and usually interchangeable modules that are used in either the scanning mode or contact mode. The

interchangeable modules allow the user to adjust for different types of inspection problems. The scanning mode is used to detect ultrasounds that travel in the atmosphere, such as a pressure leak, while the contact mode is used to detect ultrasounds generated within a casing, such as in a bearing, pump, or valve.

The reason ultrasound is so versatile is that it detects the sound of a leak. When a fluid leaks, it moves from the high-pressure side through the leak site to the low-pressure side, where it expands rapidly and produces a turbulent flow. This turbulence has strong ultrasonic components. The intensity of the ultrasonic signal falls off rapidly from the source, allowing the exact spot of a leak to be located.

Some instruments include a rubber focusing probe that narrows the area of reception so that a small emission can be pinpointed. The rubber focusing probe is also an excellent tool for confirming the location of a leak. This is done by pressing the probe against the surface of the suspected area to determine if the sound of the leak remains constant. If it decreases in volume, the leak is elsewhere.

System valves are usually checked for leakage with the contact probe on the downstream side. This is accomplished by first touching the upstream side and adjusting the sensitivity to about 50% of scale. The downstream side is then touched and the sound intensity is compared. If the signal is lower than the upstream, the valve is considered not leaking. If it is louder than the upstream side and is accompanied by a typical rushing sound, it is considered to be leaking. Pump suction and discharge valves can be checked for abnormal noises such as leak by, chattering, and valve slamming. Additionally, it is possible to determine which valve is leaking by listening with a standard mechanic's stethoscope. While not sophisticated, a distinct difference can be heard between tightly closing valves and leak-by or bypass flows. The leaking valve does not have a definite and distinct closing sound.

2.2.4 Stabilizer and Desurger

Pulsation is the acceleration and deceleration of uncontrolled energy units. These units of energy are transmitted through liquid flowing in a system of pipes. Pulsation in these systems is most often created by the start/stop or reversing action of a reciprocating type pump. Observed as vibration, pulsation can be measured by using a simple pressure gauge or using a transducer and oscilloscope for greater accuracy. Pulsation is generally expressed as a sinusoidal curve or wave of peaks and valleys of pressure spikes.

A related phenomenon, surge (water hammer) is created by rapid changes in a liquid's velocity. Velocity changes can be caused by a rapid or abrupt change in direction, valves or elbows in pipe systems, changes in pipe diameter, or differences in elevation. Startup and shutdown of the pump itself creates an instantaneous change in liquid velocity. The single biggest contributor to surging is the quick-closing valve. When a valve closes rapidly, fluid flow comes to an abrupt stop and reverses direction. The pressure spike created can be as great as five times the working pressure in the system.

This rapid change in velocity creates a compression wave that travels the length of the pipe run and continues to oscillate until friction dissipates or a system component fails. For all practical purposes, liquids are not compressible. Liquid neither absorbs nor cushions the pulse or surge but rather transmits the energy.

Virtually every commonly used pump creates pulsation. However, centrifugal designs produce a very high-frequency (rapid) but low-amplitude (narrow pressure band) pulse. The flow of fluid is a near steady state with minimal peaks and valleys of pressure spikes. Centrifugal pumps usually create surge (water hammer) only when starting up or shutting down. Reciprocating positive displacement pumps move liquid by trapping and then expelling discrete units, or slugs, of liquids. These slugs of uncontrolled energy generate the pulsation. In reciprocating pumps, the pumped fluid actually comes to a complete stop every time the plunger changes direction, which can occur up to several times per second. This start/stop action creates harmful pressure spikes in the form of high-amplitude, low-frequency pulses.

Unfortunately, eliminating pump-induced pulsation does not necessarily remove the potential for piping system damage, because a change in velocity can also create destructive pressure spikes. Quick-closing valves are certainly a major culprit, but less obvious situations such as direction changes in the system, changes in pipe diameter, and lengthy vertical rises can also contribute to surge. To a large extent, surge does not depend on pump type. Surge is most damaging at higher flow rates with large pipe diameters or long pipe runs. The larger the volume of liquid involved, the greater the force exerted on the system.

To take up irregularities and induce a uniform flow in suction and discharge lines, pulsation dampeners are used (see Figure 2-31). The volume of the dampener should be three to four times the displacement of one plunger stroke. A simple dampener is usually sufficient on the inlet of the suction side of pumps, but at higher delivery pressures, the gas charge is soon lost by dissolving into the fluid pumped. Dampeners with some form of diaphragm or bladder to hold the charge are used in most applications to prevent this loss. Some facilities use suction and discharge vessels without bladders or gas blankets that act as high-pass filters.



Figure 2-31
Discharge Dampener

Because gas is compressible and liquid is not, Boyle's law of $P_1V_1 = P_2V_2$ is the principle involved. That is, volume is inversely proportional to pressure. Air or gas under pressure is introduced into the top section of a sealed pressure vessel. Liquid is allowed to enter into the lower half, but some form of separation (usually a bladder, bellows, or diaphragm) separates the liquid from the gas. The pressure of the gas is set at a predetermined level based on calculated or observed fluid pressure in the pipeline (usually 80–90% system pressure). When pulsation creates pressure spikes, the trapped gas in the dampener absorbs the spikes. As pumped fluid flows, it takes the path of least resistance and enters the dampener. As fluid fills the dampener, the bladder compresses the gas to equal maximum fluid pressure. At the point of pump shift, the fluid pressure

drops below the compressed gas pressure. With fluid pressure now less than gas pressure, the bladder is forced down, discharging any accumulated fluid back into the pipeline and filling the void created during pump shift. If the dampener is located in the highest peak amplitude of the fluid, considering piping configuration and nominal fluid flow, the result is a continuous, vibration-free flow of fluid. A typical suction stabilizer arrangement is shown in Figure 2-32, while Figure 2-33 shows the principle of operation for a common discharge dampener.

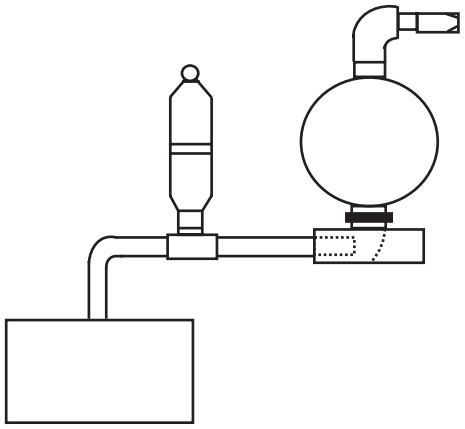


Figure 2-32
Suction Stabilizer Arrangement

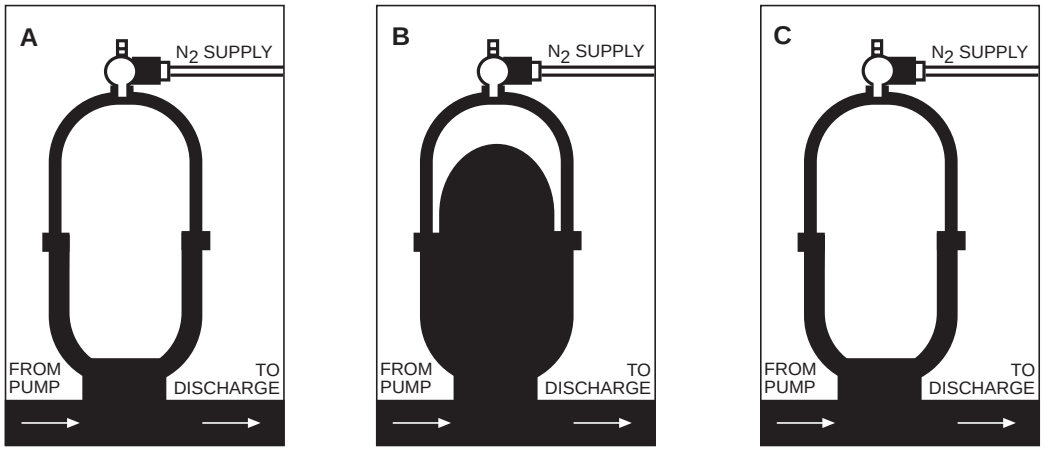


Figure 2-33
Typical Discharge Dampener Operation

Sizing of dampeners can be estimated using the calculation that follows.

Information necessary:

- Volume per stroke of pump (in cubic inches). One gallon equals 231 cubic inches. Volume is determined by the following: 0.7854 x (bore diameter)^2 x (stroke length).

- Type of pump: simplex, duplex, or triplex.
- Maximum operating pressure.
- Minimum operating pressure.
- Average operating pressure (minimum and maximum divided by 2).

Capacity (c) equals:

$$\frac{(V) \times (K) \times (P_{av} / P_{min})^{Y}}{1 - (P_{av} / P_{max})^{Y}}$$

where: V = volume per stroke in cubic inches
K = number from pump type below
P_{av} = average discharge pressure
P_{min} = minimum discharge pressure
P_{max} = maximum discharge pressure
Y = 1 if the charge is compressed air, 0.714 if the gas charge is nitrogen

K factor for various pump types

Simplex single-acting	0.60
Simplex double-acting	0.25
Duplex single-acting	0.25
Duplex double-acting	0.15
Triplex single-acting	0.13
Triplex double-acting	0.06

Pulsation dampeners are available in capacities from less than a cubic inch to well over 100 gallons and pressures greater than 30,000 psi.

2.2.5 Fluid Drive

The fluid drive input shaft, impeller, and casings rotate together at motor speed. The output shaft and runner rotate together at output speed. There is no mechanical connection between the input and output members. Power is smoothly transmitted from the impeller to the runner by a vortex of oil. The impeller pumps oil into the runner where the kinetic energy of the oil is absorbed and transmitted to the output shaft. The output speed is adjusted by regulating the amount of oil in the working circuit.

The circulating pump sends oil from the housing reservoir to the external oil cooler. Oil from the cooler is returned to the inside of the fluid drive housing and enters the working circuit through the back of the impeller. Oil entering the working circuit is acted upon by centrifugal force and takes the form of an annular ring in the working circuit and scoop tube chamber. The scoop tube is positioned by the external speed control to set the level of oil in the scoop tube chamber. This determines the amount of oil in the working circuit and, therefore, the output speed of the fluid drive. Smooth, stepless control over a wide range of speeds is available.

The input and output rotor system is supported with ball or roller antifriction bearings. The output inboard bearing is located between the impeller and the runner. The output inboard bearing is of the ball or spherical roller type. The bearings use a force-feed oil lubrication system.

Speed change is accomplished by varying the quantity of oil within the rotors by a scoop tube mounted internally within the fluid drive. Movement of the scoop tube occurs by means of an external control rod extending outside the housing from the output end bell. Full stroke of the scoop tube is maintained by external, adjustable stops. When connected to an external actuator, the actuator stops must always be set to contact before the mechanical stops on the fluid drive.

The positive displacement oil pump is either integral to or chain driven from the input shaft. If chain driven, the chain is spray lubricated. The pump picks up oil from the sump and delivers it to an oil cooler. From the cooler, the oil is sent to both the fluid drive power circuit and the bearings. After passing through the circuit, the oil is returned to the sump by the scoop tube. The standard water-to-oil coolers are shell and tube construction, while the alternate air-to-oil coolers are cooled by a direct-drive fan.

2.2.6 Packing

Shaft packing forms a watertight seal between the cylinder / stuffing box and the plungers to prevent leakage from the system to the environment. The packing is made up of a primary set and a secondary set of packing with cooling water injected or convected between the two sets. The primary and secondary sets are usually of similar make-up and consist of a low-pressure-side adapter ring (usually bronze or austenetic material), multiple rings of packing, a high-pressure-side adapter ring (usually bronze or austenetic material), and a packing preload spring (stainless steel). See Figures 2-34 through 2-36. Some units also use a tertiary packing. Packing leak-off is collected between the secondary and tertiary sets. The two types of packing generally used are the molded V-ring (sometimes called chevron style) and the braided composition varieties.



Figure 2-34
Stuffing Box Packing Showing Packing Rings,
Male and Female Adapters, and Preload Springs

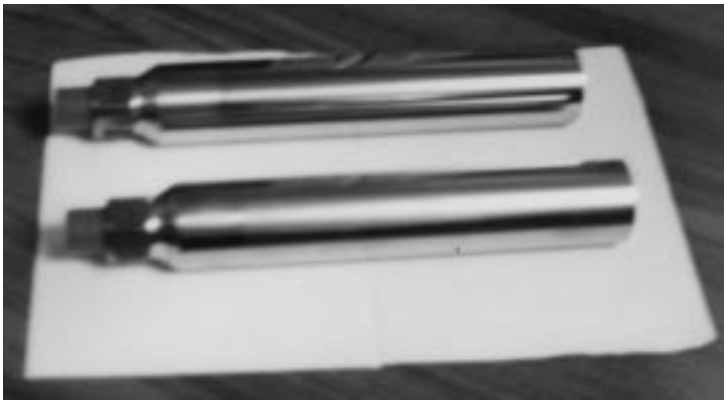


Figure 2-35
Plungers

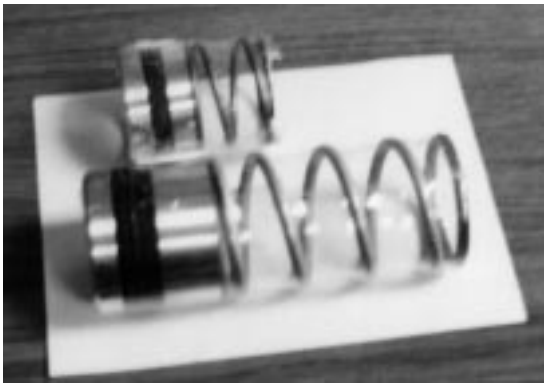


Figure 2-36
Packing Rings and Preload Springs

2.2.6.1 Molded V-Ring Packing

Unlike O-rings, the V-ring seal provides a seal in only one direction. V-rings are always installed with the open end of the V facing the pressure. Male and female adapters are used as the high- and low-pressure-side adapters, respectively. The packing sets are preloaded by primary and secondary stainless steel springs. The springs are always placed on the high-pressure side of the packing and act to keep the packing loaded under low-pressure or no-pressure conditions (such as during the suction intake stroke). Most current designs require no adjustment, because the packing sets are preloaded by springs. For pumps with gland nut configurations, the gland nut should be fully seated and torqued to a level recommended by the pump manufacturer. Some older designs allowed the secondary packing preload to be adjusted by changing the torque on the gland nut. This packing arrangement was susceptible to damage caused by overtorquing the gland nut.

V-ring packing can be split for installation; however, it is then more prone to leakage than solid packing. Splitting the packing is not recommended on new installationtñbut should be done only as an emergency fix. Male and female adapters are used in conjunction with V-rings for reinforcement. Stacked V-rings have great theoretical sealing ability at low to high pressures but are usually bulky and very susceptible to poor assembly techniques. A smaller, less bulky design is manufactured by Garlock that is interchangeable with originally supplied packing. An advantage of V-ring packing is that the rings can be cut and wrapped around the plunger without disassembling the cylinder. Friction can be reduced by specifying that one or more of the rings should be of special material with good lubricity. If the stack is designed wrong, the pressure load falls to the first seal, and the remaining seals see very little pressure and might even run dry. Stacked V-rings sometimes fail because the first ring in the stack gets most of the lubricant and the remaining rings in the stack run dry. A V-ring packing arrangement is shown in Figure 2-37.

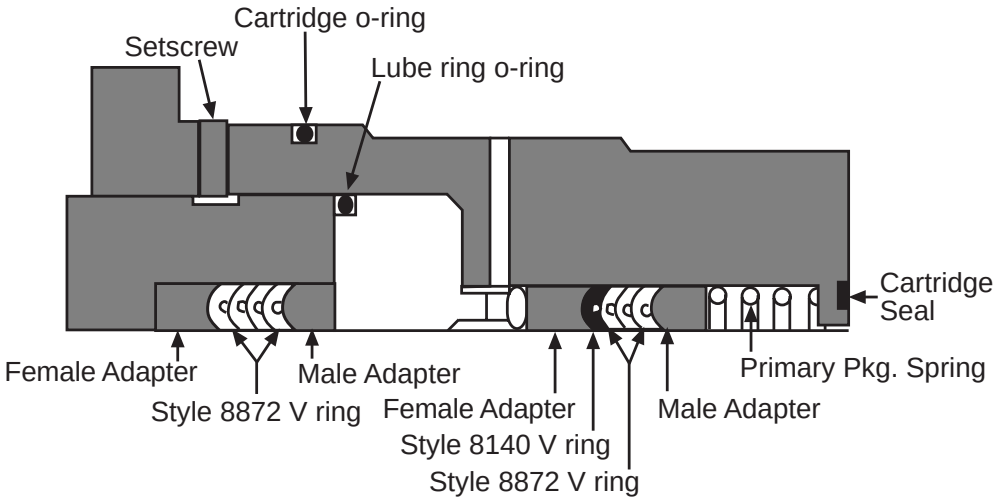


Figure 2-37
V-Ring Packing Installation

2.2.6.2 Braided Composition Packing

The braided composition type of packing is typically an aramid fiber. Kevlar is the most commonly used of this type of packing, and it is usually treated or impregnated with TFE (tetrafluoroethylene), which acts to lubricate the packing and reduce the packing friction. A standard braided composition packing arrangement (Kevlar-aramid) is shown in Figure 2-38.

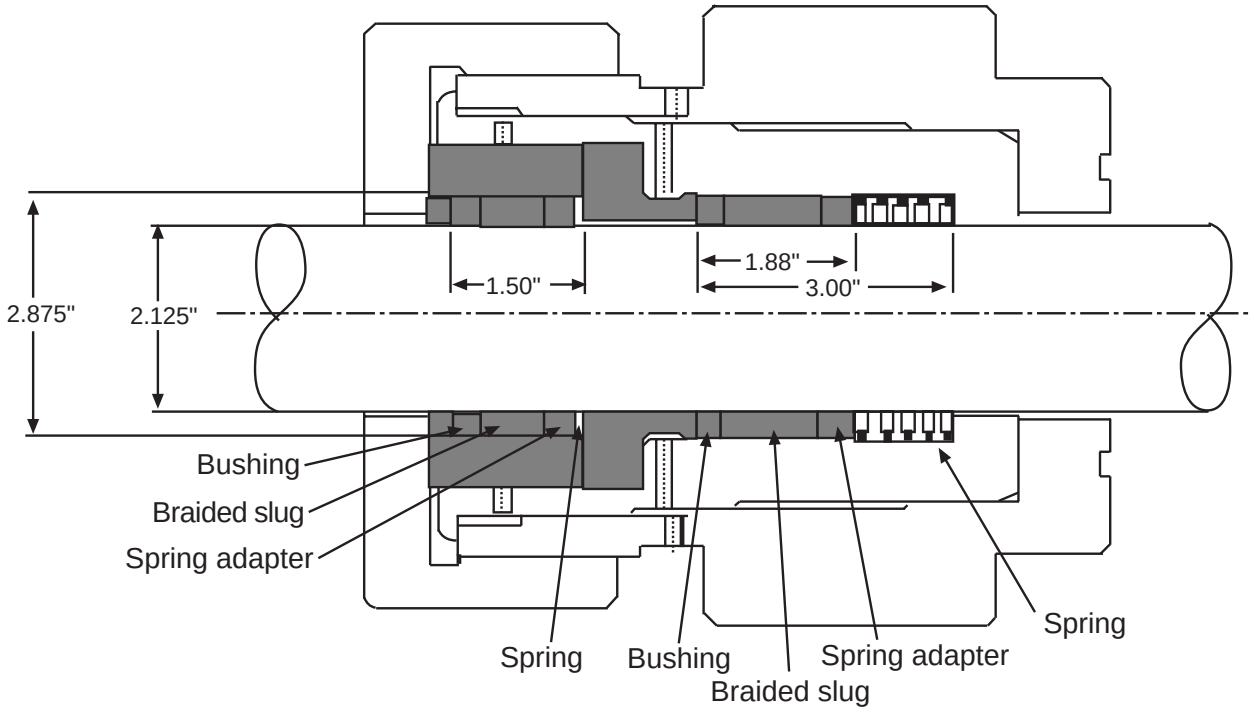


Figure 2-38
Typical Braided Composition Packing Arrangement

The number of rings for a set of packing varies from three to five. If the stuffing box is deeper than required for the five rings, a bushing of carbon or non-galling stainless steel should be used to shorten the box depth. Live loading is used in some applications to maintain a positive seating force. Disk spring washers are commonly used on packing box studs. As the packing set loses volume, springs maintain a minimum level of load on the packing and maintain the seal. Many packing arrangements are equipped with a seal water system that provides a positive pressure to eliminate leakage to the atmosphere and to help reduce packing and stuffing box temperatures. A basic packing seal water system is shown in Figure 2-39.

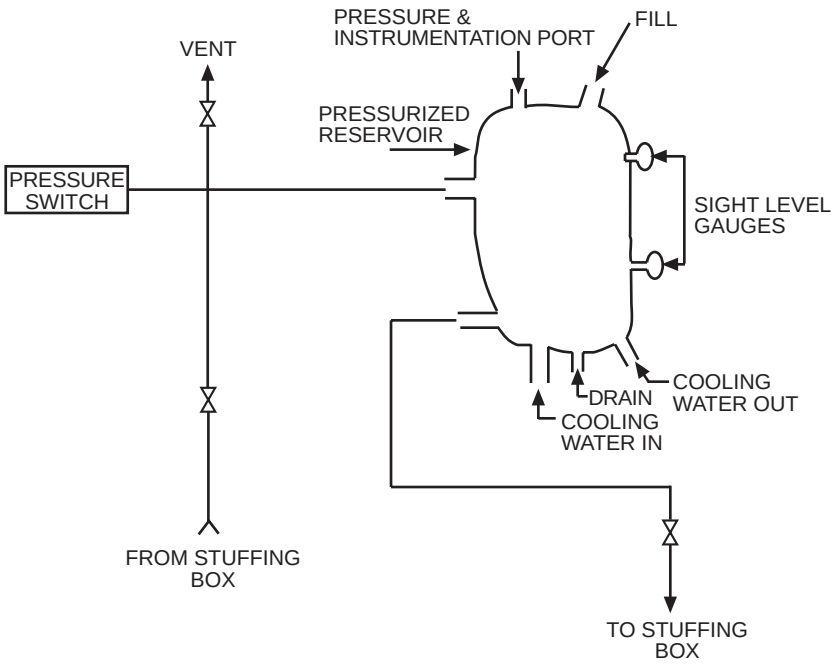


Figure 2-39
Simplified Packing Cooling System

2.2.7 Piping Considerations

Inadequate pulsation control and poor valve performance, along with faulty mechanical piping design, are the primary causes of excessive vibration in positive displacement pump systems. Pressure pulsations acting on unbalanced areas such as elbows and closed valves generate dynamic shaking forces that can cause high vibration levels in spite of adequate mechanical piping supports.

When a piping system is exposed to fluctuating pressure disturbances or pulsations, it frequently responds by vibrating. The magnitude and nature of the piping system vibration is dependent upon the frequency and energy content of the excitation. Low-to-moderate level periodic excitation, such as the pressure pulsations from positive displacement pumps, does not ordinarily excite significant levels of response in the piping system, as long as the excitation frequencies are different from the natural vibrating frequencies of the pipe. If the pulsation frequency coincides with the natural frequencies of the piping system, however, resonant vibration can occur. Random excitation associated with cavitation, bubble collapse, and rapid pressure reductions also leads to resonant vibration. This type of vibration is known as *self-excited vibration*. Resonant response, whether due to the effects of fixed frequency or random excitation, can lead to unacceptable piping system damage.

2.2.8 Performance and Efficiency

Performance and efficiency in positive displacement pumps are difficult to measure due to the pulsating flow delivered. This pulsating flow creates a problem when trying to select a steady-state value to use in calculations of power and efficiency. Pump efficiency depends on too many variables to allow easy calculation. Typical variables are internal backflow (slip), cavitation, and friction (dry and viscous). However, pumps of similar design tend to follow the same pattern of losses, and these can be estimated with "averaged" data. Volumetric efficiency—an important part of overall efficiency—is the ratio of delivered flow to ideal flow. Delivered flow is calculated using the following formula:

$$Q_d = E_v DN$$

where Q_d = delivered flow

E_v = volumetric efficiency

D = displacement

N = speed

Volumetric efficiency depends on the shape and clearance of rotating and stationary elements, the suction characteristics of the pump, the viscosity of the liquid, and the speed and pressure of the pump. If clearances are small and if there is good dynamic sealing between the rotor and stator, internal backflow is small and volumetric efficiency high. The viscosity of the liquid has a pronounced effect on volumetric and overall efficiency and also on friction load and cavitation. The effects are opposing. High viscosity helps reduce slip (improving volumetric efficiency) but increases friction load and hinders suction. Low viscosity improves suction and decreases friction load but increases slip (lowers volumetric efficiency). Slip is a function of pressure, not speed.

2.3 Bearing Design and Construction

Bearings are load-carrying elements. Journal bearings are cylindrical and are used when the shaft load is essentially perpendicular to the axis of the shaft. Thrust bearings are ring-shaped and are used when the shaft load is in the direction of the axis of the shaft. Characteristic of all plain bearings is the sliding movement of the shaft (or thrust collar) surface with respect to the bearing surface. To reduce the friction that results from this sliding movement, the bearings are provided with a continuous film of oil between the sliding surfaces.

Most bearings are made of metals or alloys. Typical bearing alloys that have excellent antiseizure characteristics and can be used at high sliding speeds are the tin-base and lead-base babbitts, both relatively soft. To improve their load-carrying capacity, these alloys must be supported by a bearing shell of stronger material. Bearing shells are usually made of steel, cast iron, or bronze.

Ultrasonic inspection and monitoring of bearings is a reliable method for detecting incipient bearing failure. The ultrasonic warning appears prior to a rise in temperature or an increase in driving torque. If a vibration program already exists for bearing analysis, an ultrasonic bearing monitoring program can be of assistance.

2.3.1 Lubrication Theory

Lubrication reduces friction between moving parts by substituting fluid friction for solid friction. By reducing friction and thereby reducing the amount of energy that is dissipated as heat, lubrication reduces the amount of energy required to perform mechanical actions and also reduces the amount of energy that is dissipated as heat.

The friction that exists between a body at rest and the surface upon which it rests is called *static friction*. The friction that exists between moving bodies (or between one moving body and a stationary surface) is called *kinetic friction*. Static friction, which must be overcome to put any body in motion, is greater than kinetic friction, which must be overcome to keep the body in motion.

There are three types of kinetic friction:

- Sliding friction. Sliding friction exists when the surface of one solid body is moved across the surface of another solid body.
- Rolling friction. Rolling friction exists when a curved body such as a cylinder or sphere rolls upon a flat or curved surface.
- Fluid friction. Fluid friction is the resistance to motion exhibited by a fluid.

Fluid friction exists because of the cohesion between particles of the fluid and the adhesion of fluid particles to the object or medium that is tending to move the fluid. Cohesion is the molecular attraction between particles that tends to hold a substance or body together; adhesion is the molecular attraction between particles that tends to cause unlike surfaces to stick together. From the point of view of lubrication, adhesion is the property of a lubricant which causes it to stick (or adhere) to the parts being lubricated; cohesion is the property which holds the lubricant together and enables it to resist breakdown under pressure. Cohesion and adhesion are possessed by different materials in widely varying degrees. In general, solid bodies are highly cohesive but only slightly adhesive. Most fluids are quite highly adhesive but only slightly cohesive.

Fluid lubrication is based on the actual separation of surfaces so that no metal-to-metal contact occurs. As long as the lubricant film remains unbroken, sliding friction and rolling friction are replaced by fluid friction. An accepted theory of lubrication is based on the Langmuir theory of the action of fluid films of oil between two surfaces, one or both of which are in motion.

Theoretically, there are three or more layers or films of oil existing between two lubricated bearing surfaces. Two of the films are boundary films, one of which clings to the surface of the rotating journal and one of which clings to the stationary lining of the bearing. Between these two boundary films are one or more fluid films.

When the rotating journal is set in motion, as shown in Figure 2-40, the relationship of the journal to the bearing lining is such that a wedge of oil is formed. The oil films begin to slide between the two boundary films, thus continuously preventing contact between the two surfaces.

As the journal slowly starts to turn and the speed increases, oil adhering to the surfaces of the journal is carried into the film, increasing the film thickness and tending to lift the journal. Changes in temperature, with consequent changes in oil viscosity, cause changes in the film thickness and in the position of the journal. If conditions are correct, the two surfaces are effectively separated, except for a possible momentary contact at the time the motion is started.

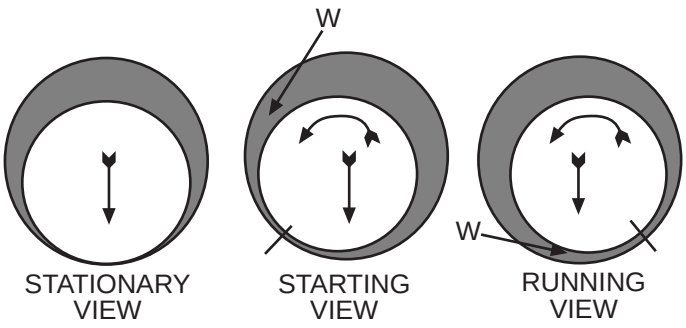


Figure 2-40
Oil Film Lubrication

A number of factors determine the efficiency of oil film lubrication, including such things as:

- Pressure
- Temperature
- Viscosity
- Speed
- Alignment

- Condition of the bearing surfaces
- Running clearances between the bearing surfaces
- Starting torque
- Nature and purity of the lubricant

A lubricant must be able to stick to the bearing surfaces and support the load at operating speeds. More adhesiveness is required to make a lubricant adhere to bearing surfaces at high speeds than at low speeds. At low speeds, greater cohesiveness is required to keep the lubricant from being squeezed out from between the bearing surfaces. Large clearances between bearing surfaces require high viscosity and cohesiveness in the lubricant to ensure maintenance of the lubricating oil film. The larger the clearance, the greater must be the resistance of the lubricant to being pounded out, with consequent destruction of the lubricating oil film. High unit load on a bearing requires high viscosity of the lubricant. A lubricant subjected to high loading must be sufficiently cohesive to hold together and maintain the oil film.

2.3.2 Gear Reducer Bearings

The parallel-shaft, double-reduction, gear-driven speed reducer uses either tapered or spherical roller bearings to compensate for radial loads and axial thrust loads. Tapered roller bearings combine the advantages of ball and straight roller bearings because they can take either radial or thrust loads or any combination of the two. Additionally, they have a high load-carrying capacity. The tapered roller bearing is designed so that all elements in the roller surface and the raceways intersect at a common point on the bearing axis.

Roller bearings generally are made of carburized material with case hardness Rockwell C 58–63 and core hardness Rockwell C 25–40. These materials are limited to operating temperatures of less than 350°F, but special low-carbon steels that can be carburized operate satisfactorily at temperatures as high as about 600°F.

Rolling element bearings require significantly less oil flow than journal bearings. As a result, they exhibit less power consumption and heat generation. They also have lower starting torque requirements, particularly at low temperatures where lubricant viscosity is high. The life of a bearing is expressed as the number of revolutions, or the number of hours at a given speed, that a bearing will complete before failing by subsurface fatigue. Figure 2-41 shows a tapered roller bearing.

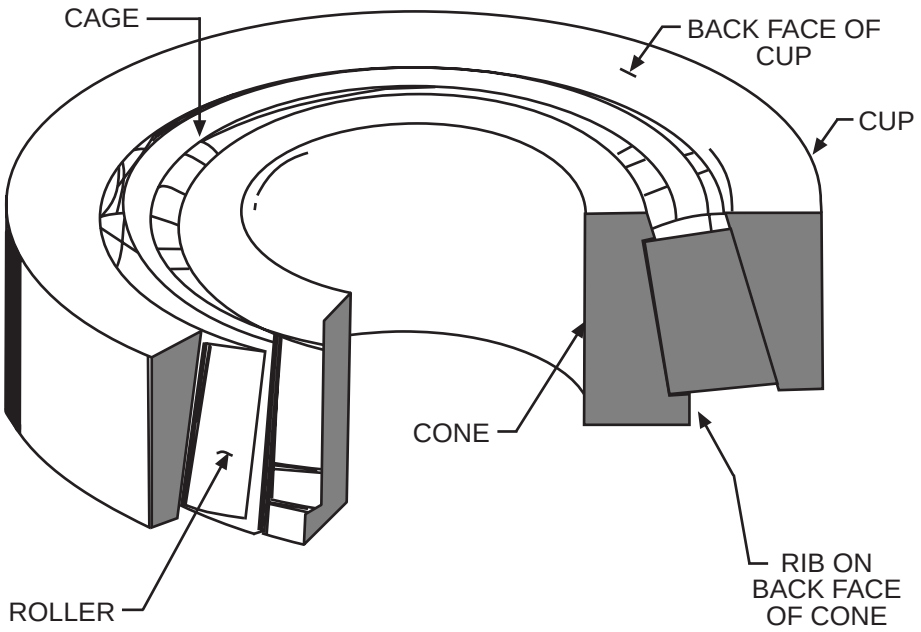


Figure 2-41
Tapered Roller Bearing

2.3.3 Crankcase Bearings

The bearings used in the crankcase consist of sleeve bearings to compensate for radial loads. Wear develops gradually and is easy to detect before it causes serious damage. These bearings perform well under severe service or conditions such as high moisture or extremely high temperatures.

Analyzing the direction and magnitude of the load helps to determine the correct bearing material, as well as the support housing design, housing material, and location of lubricant entry. Because each bearing material has different advantages and disadvantages, sleeve bearings are manufactured from a variety of materials to suit widely different applications.

The typical shaft journal bearings used in the charging pump are babbitt-lined, fully bronze-sleeved bearings.

2.4 Pump Drive Design and Construction

The typical shaft in a Union pump is a heavy, one-piece, forged and eccentric ground crankshaft. The shaft runs in two heavy duty sets of tapered roller bearings that are force lubricated by the crankshaft-driven internal oil pump. A special “ball-type” thrust bearing is mounted on the end of the shaft to absorb the loads generated by the pump internal reduction gearing.

Gaulin pumps use a heavy, one-piece, high-strength alloy casting on which are installed ductile iron cams. The shaft runs in two babbitt-lined, fully bronzed bearings located in bearing retainers. A fiber thrust washer is mounted on each end of the shaft between the outside cam and the shaft bearings, making the total end play between 1/32–1/16". The shaft is gun-drilled, meaning that oil is pumped through the center of the shaft and out to each cam, so that the connecting rod bearings ride on a cushion of oil under pressure. There are plugs installed in both ends of the shaft.

The gear drive is a parallel-shaft, double-reduction-type speed reducer. Gearbox shaft seals are usually either the more common elastomeric lip seals (shown in Figure 2-42) or noncontacting labyrinth seals. By elastomeric, it is meant that the sealing lip is made of a synthetic rubber compound.

Oil sealing is accomplished through an interference fit between the flexible sealing lip and the shaft. Because the elastomer can lose tension and elasticity during operation, a garter spring is usually incorporated on the lip to ensure pressure at the sealing interface. Typically, a dirt excluder lip on the air side is used to shield the oil sealing lip from contaminants.

The most commonly used lip seal materials in gearbox applications are the nitriles, sometimes known as Buna N. The standard nitrile compound is compatible with most mineral oils and can operate at temperatures up to about 225°F. Nitrile seals are relatively inexpensive and readily available. The material is not suited for temperatures above 250°F because it tends to harden.

For high-temperature applications, fluoroelastomers are recommended. These materials, sometimes known by the trade name Viton, can be used for temperatures up to 400°F. They have outstanding resistance to a wide variety of fluids, including synthetic lubricants.

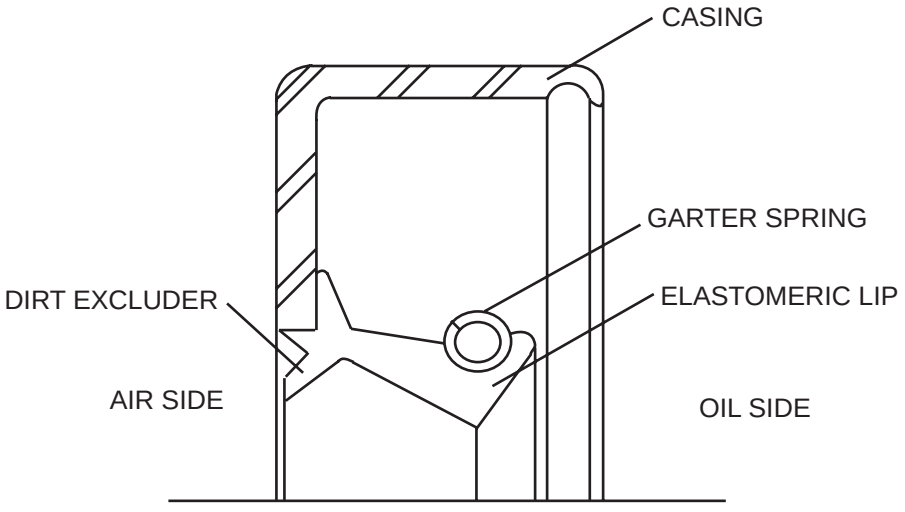
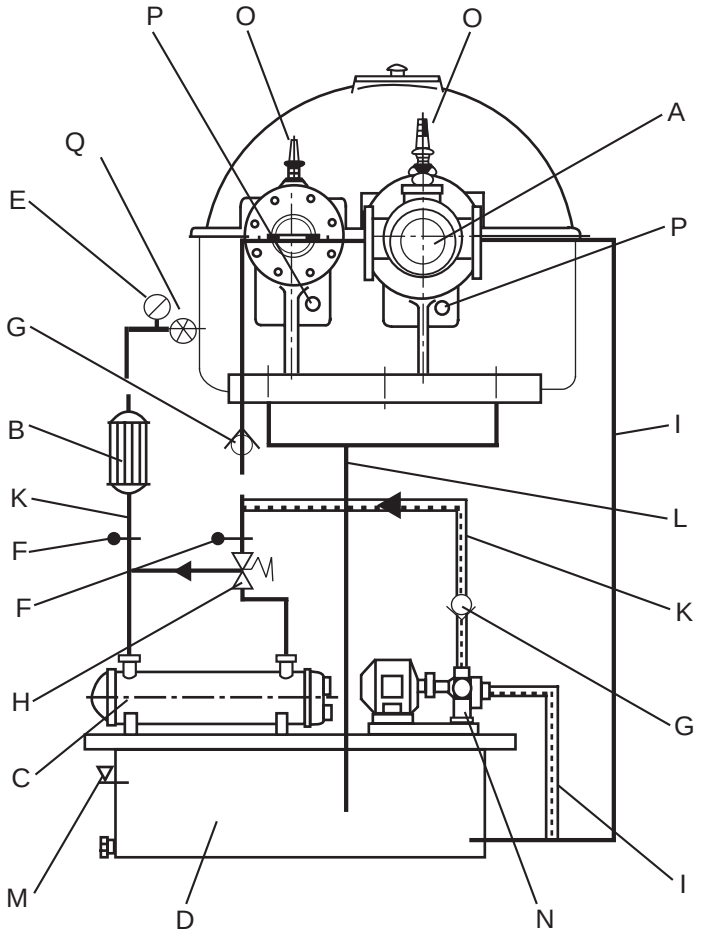


Figure 2-42
Typical Lip Seal

The selection of a gear oil depends on several factors, including the unit's operating speed and load, ambient temperatures, and the lubricants available at the site. The most important parameter in selecting a lubricant is the viscosity. High-speed units require less viscous oil than gears operating at low speed. At high speed, an acceptable oil film is generated at the tooth contact even with a low-viscosity oil. Also, the churning that occurs at high speed is less severe with a low-viscosity oil, resulting in lower power losses. At lower operating speeds, a thinner oil film is generated, and more viscous oils are required to separate the contacting surfaces. In general, the lowest viscosity oil sufficient to form an adequate film at all operating conditions should be chosen.

There are two types of lubrication systems in use: splash and forced feed. In the splash system, the unit is filled with oil to a predetermined level and operated as a sealed system with no external connections. In the forced-feed system, shown in Figure 2-43, oil is introduced into the unit through jets under pressure. Scavenge oil is pumped through a cooler and a filter prior to reentering the gearbox. The splash system, shown in Figure 2-44, is far simpler and less expensive than the forced-feed design, but is applicable only to low-speed units. As speeds increase, the heat generated in the gearbox becomes excessive, and an external system is required to cool the lubricant.



A – Oil Pump
B – Oil Filter
C – Oil Cooler
D – Oil Tank
E – Pressure Gauge
F – Thermometer
G – Check Valve
H – Overflow Valve
I – Suction Pipe
K – Pressure Pipe
L – Return Flow Pipe
M – Oil Level Gauge
N – Primer and Spare Pump Kit
O – Temperature and Flow Gauge
P – Connection for Remote Thermometer
Q – Pressure Regulator

Figure 2-43
Typical Forced-Feed Lubrication System

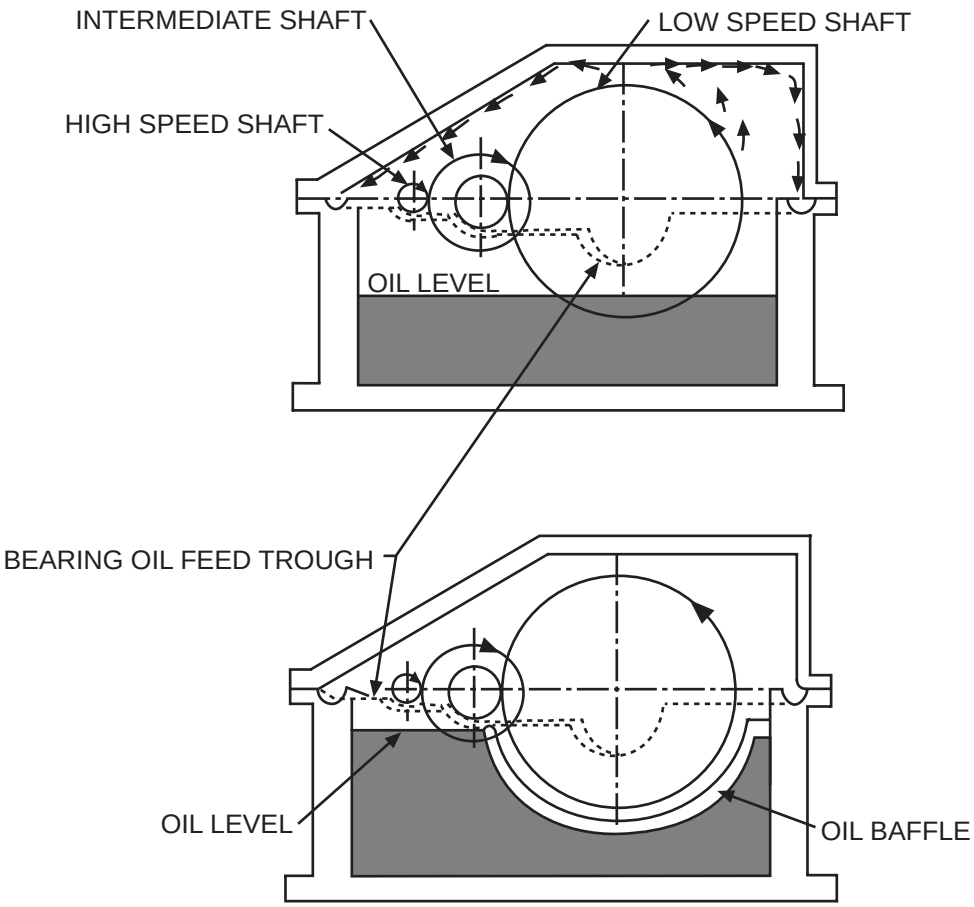


Figure 2-44
Typical Splash-Type Lubrication System

In the forced-feed system, the shaft-driven oil pump takes a suction on the sump and discharges through a cooler, filter, and usually a pressure relief valve located at the inlet to the gearbox. An auxiliary pump is usually included to prime the system prior to startup or to be used as a backup in case of failure of the main pump. A bypass is normally located at the cooler to ensure that the oil is not cooled too much. A typical maximum temperature for oil into the gearbox is 130°F. A typical oil temperature rise across the gearbox is 30°F. Usually, gearbox feed pressures are on the order of 20–100 psi.

2.5 Coupling Design and Construction

Flexible couplings are designed to accommodate various types of load configurations. The term *flexible* does not mean that the coupling gives complete freedom of movement to coupled shafts with no transmissible force. Flexible couplings actually give limited freedom of movement with transmitted force. These forces are either lateral (at right angles to the shafts) or axial in nature, or some combination of the two. Lateral forces

can produce a bending moment on the shafts and a radial load on the support bearings. Axial force can produce undesirable thrust loads if not considered in the original design.

On centrifugal pumps, flexible couplings are provided for connecting the pump to the speed increaser and the speed increaser to the motor. The coupling between the pump and speed increaser is a spoolpiece-type coupling. It is dynamically balanced and matchmarked. The coupling between the speed increaser and the motor is a non-spoolpiece-type coupling. Its end play can be limited by a spacer button on the speed increaser shaft, if provided.

Installing a spoolpiece-type coupling next to the pump permits the pump coupling half to be removed and installed on the pump shaft without disturbing the position of either the pump, driver, fluid drive, or speed increaser.

It is necessary to limit coupling end play in applications where electric motors are fitted with sleeve bearings. By limiting end play in the coupling, the thrust bearing on the pump speed increaser holds the rotor of the motor so that the thrust collars on the motor shaft do not touch the motor bearing faces. This prevents overheating or damage to the motor bearings. Figure 2-45 shows a typical coupling used on centrifugal pumps.



Figure 2-45
Typical Flexible Coupling

On positive displacement pumps, gear couplings or shim-pack type couplings are used between the motor and the gearbox and between the gearbox and the crankcase. See Figure 2-46.

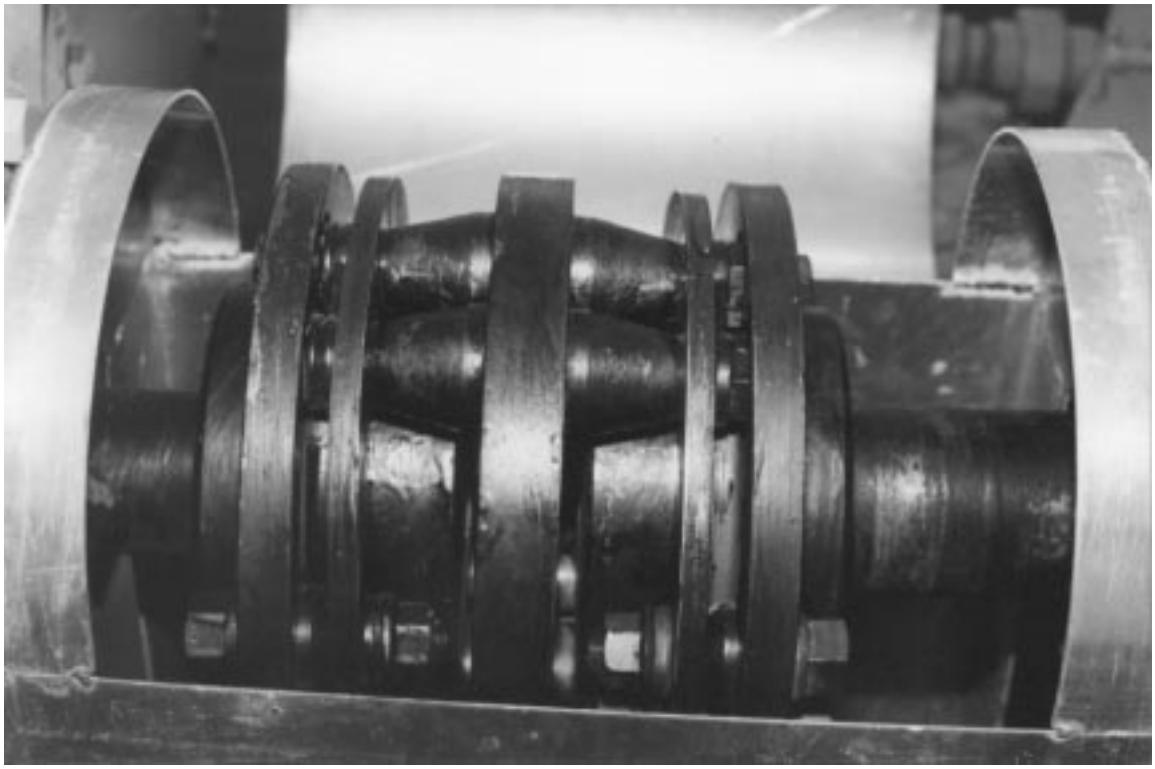


Figure 2-46
Thomas / Rexnord DBZ Size 351 Flexible Coupling between Gyrol Fluid Drive Low Speed Shaft and Pump Input Shaft

2.5.1 Motor/Gearbox and Gearbox/Crankcase Couplings

The motor-to-gearbox and gearbox-to-crankcase couplings used in this application are large gear-type couplings. The crowned gear teeth are triple crowned with a 20° full-depth tooth form to deliver greater capacity under misalignment. Since perfect shaft alignment is ideal but impractical (due to foundation settling, temperature changes, normal bearing wear, and vibration), all flexible couplings must provide some degree of angular motion between hub and sleeve.

Barrel-shaped hub teeth with three curved surfaces—tip, flank, and root—distribute the load over a wide area at the strongest part of the teeth. They also permit free axial movement, provide close tooth contact under all conditions, give a positive film of lubricant on teeth in contact areas, and eliminate corner loading. Hubs are machined from carbon or alloy heat-treated steel to close tolerances. Flanged sleeves are machined from carbon or alloy heat-treated steel.

Larger couplings typically use seal rings at the sleeve center and end plate joints. A typical large gear coupling is shown in Figure 2-47.

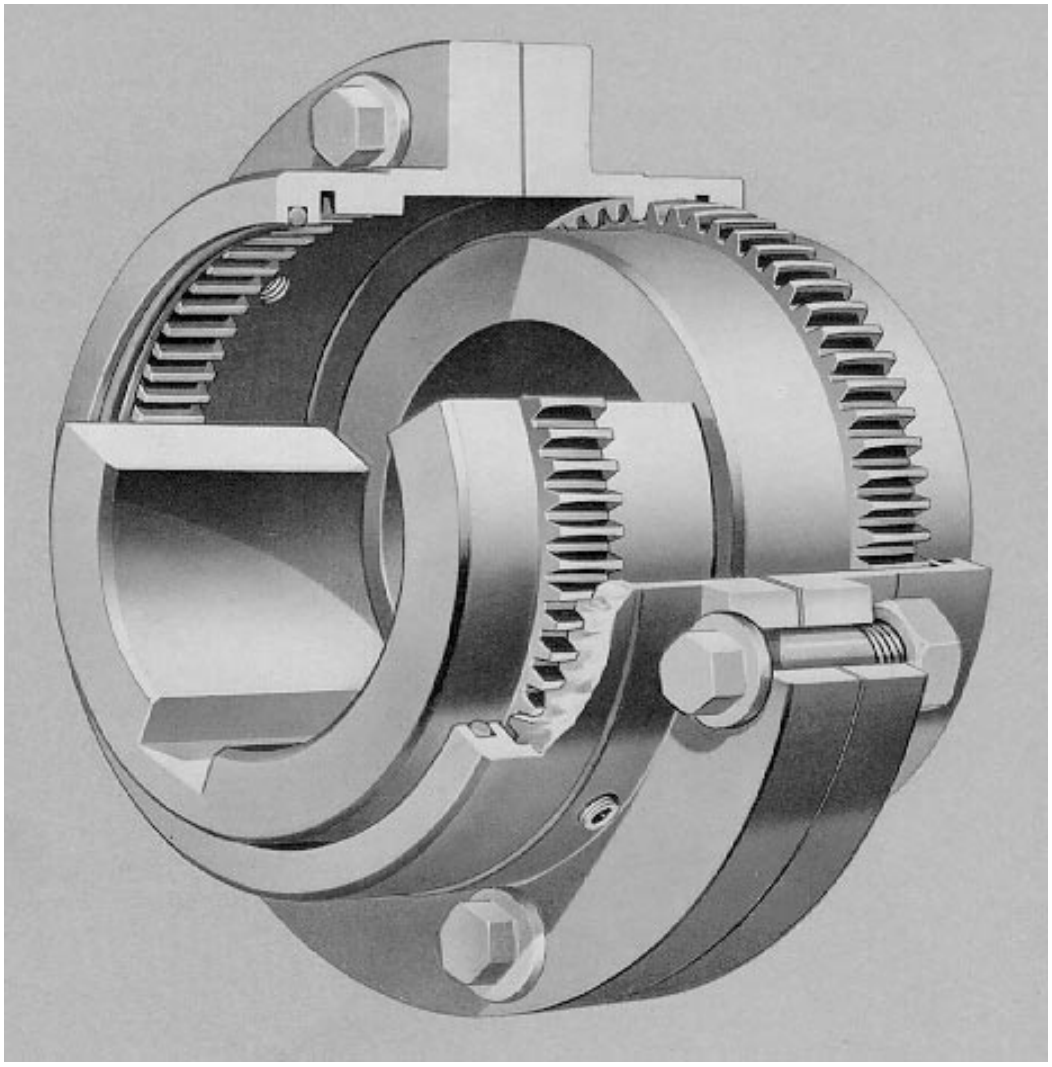


Figure 2-47
Large Gear Coupling

2.5.2 Alignment Techniques

Misalignment can be defined as “any condition in which the driver and driven shaft centerlines are not coplanar vertically and horizontally at the machine’s normal operating temperature.” It is one of the most common sources of excessive, disabling vibration in rotating machinery. Misalignment also makes pumps vibrate, causes leaks in seals, makes bearings hot, and generates wear in couplings. Machinery feet that are not individually coplanar with their bases when unbolted are the most common condition that underlies misalignment of rotating machines. This noncoplanar condition is caused by soft foot and pipe stress.

Soft foot is a condition that occurs due to non-parallel surfaces of the unit's foundation to the feet of the machine. It is caused by either a poorly constructed base or manufacturing of the frame with out-of-specification tolerances. The industry standard allows up to 0.003“ out-of-flatness between motor feet; but for accurate alignment, flatness should be within 0.001”. This can be achieved through pre-installation checks and machining the feet of the unit prior to installation. The use of step and / or tapered shims can compensate, but this adds difficulty to the alignment process.

Soft foot is a measurement of the flexing of the motor / pump frame at the foot while bringing the base of the foot parallel with the base. It causes distortion of the frame, which can change the relationship between shaft and shaft components. This condition is usually prevalent in aluminum frame motors. It makes a precision alignment very difficult, sometimes unachievable. Its largely diagonal base stresses are transmitted through the bearing housing to the shaft, which consequently “wobbles” when turned by hand during alignment measurement and vibrates at motor speed while operating.

Pipe stress is an attachment problem. It transmits stresses through the case and distorts foot / base planarity and thus degrades shaft rotation just as soft foot does. Frequently, misalignment has both offset and angular components. Offset is the distance between shaft centerlines at a given point. Closely related to this is parallel offset, the distance between parallel centerlines. Angularity is the angle between two centerlines.

The three basic alignment techniques currently used are rim and face, reverse dial, and laser alignment. Reverse dial and laser alignment are the more accurate methods because they are calculation based.

A more detailed discussion of reverse dial and laser alignment practices and the basic procedures is contained in Appendix D.

3

PREDICTIVE AND PREVENTIVE MAINTENANCE

Reliability-centered maintenance (RCM) principles establish maintenance tasks that are directed at optimizing and documenting component and equipment reliability. Demands for safety and availability result in a need to ensure that all efforts aimed at increasing reliability are cost effective. The risks associated with different maintenance approaches are difficult to assess. Reliance on equipment vendors for establishing maintenance programs is less than optimal. Vendor recommendations do not usually incorporate actual plant service or downtime cost considerations and almost never recognize applicable predictive maintenance tasks. The incorporation of RCM principles to define the maintenance tasks required to maximize reliability ensures that each maintenance task is applicable and cost effective. An applicable task is defined as a task that either detects, prevents, or mitigates the functional failure consequences; while an effective task is one that eliminates or reduces the probability of failure occurrence. The RCM process has been applied by the aerospace industry, power utilities, and the military for many years.

Selected maintenance tasks can be either predictive or time-directed and are usually in one of the following forms:

- Routine preventive maintenance
- Scheduled overhaul of an item at or before a specified age limit
- Scheduled replacement of an item (or one of its parts) at or before some specified age limit
- Scheduled inspection of a component or system to find actual or potential failures

3.1 Charging Pump Maintenance Plans

Maintenance plans resulting from RCM analysis of the chemical and volume control system (CVCS) charging pumps are presented in Table 3-1 for centrifugal pumps and Table 3-2 for positive displacement pumps. These tables reflect a general consensus of RCM-based recommendations for CVCS pump maintenance requirements and frequency. The skills/crafts identified on the tables are generic in nature. Equipment condition reflects the pump status (running or shutdown) required for performing the maintenance action. The maintenance type is one of three possible categories: CM = condition-based monitoring; PM = preventive maintenance (time-directed mainte-

nance); or PL = repair, rebuild, or replace. RCM addresses only condition-based monitoring and time-directed maintenance. The PL tasks have been added to the tables as a reference to the maintenance procedures contained in Section 4 of this manual. The estimated time shown in the tables represents an average total time to perform a task, not person-hours.

Table 3-3 describes the maintenance plan developed by Electricité de France (EdF) for the KSB centrifugal pumps used in the CVCS. The French plan resulted from the application of RCM principles in a framework EdF calls OMF (Optimisation de la Maintenance pas la Fiabilite, or Optimization of Maintenance for Reliability). A 1990 OMF analysis of the CVCS by EdF resulted in the replacement of numerous charging pump inspections and overhauls with surveillance (predictive or condition-based monitoring) tasks. In addition, the pump overhaul/open and inspect task was extended to about ten years (25,000 operating hours).

The tables are provided so that each plant can compare its current maintenance plan to the RCM-derived plans, and thus assess the potential for savings that can be achieved by comparing current maintenance activities with those based on RCM techniques.

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Table 3-2
Charging Pump Equipment Maintenance Plan (Positive Displacement Pumps)

CHARGING PUMP
EQUIPMENT MAINTENANCE PLAN

EQUIPMENT NO: P/D Pump DESCRIPTION: Positive Displacement

FUNCTION: Provides borated water to primary system.

COMPONENTS MODEL MANUFACTURER

Coupling
Gear Box
Motor
Pump
Stabilizer/dampener
Valves

FAILURE CONSEQUENCE:
Safety_**X**__ Envir___ Oper___
Cost__ None__

P&ID NO:

Item No.	Maintenance Requirement	Skill/ Craft	Equip. Cond.	Maint. Type	Frequency	Procedure	Est. Time (hrs)	Remarks
1	Conduct operational inspection.	Oper	Running	CM	Daily	P/D Pump-D1	0.1	
2	Tighten all bolts, nuts, and fittings.	Mech	Shutdown	PM	Monthly	P/D Pump-M1	1.0	
3	Conduct vibration analysis.	Mech	Running	CM	Monthly	P/D Pump-M2	1.0	
4	Conduct motor current analysis.	Mech	Running	CM	Monthly	P/D Pump-M3	0.5	
5	Collect oil samples for analysis.	Oper	Shutdown	CM	Quarterly	P/D Pump-Q1	0.5	
6	Conduct surveillance testing.	Syst	Running	CM	Quarterly	P/D Pump-Q2	4.0	
7	Lubricate motor and coupling.	Mech	Shutdown	PM	Semiannual	P/D Pump-S2	1.0	
8	Drain and replace oil and oil filter.	Mech	Shutdown	PL	Situational	P/D Pump-R1	4.0	Required by oil analysis
9	Change gear reducer oil.	Mech	Shutdown	PL	Situational	P/D Pump-R2	2.0	Required by oil analysis
10	Perform alignment.	Mech	Shutdown	PL	Situational	P/D Pump-R3	4.0	Required by vibration analysis
11	Replace packing.	Mech	Shutdown	PL	Situational	P/D Pump-R4	12.0	As required
12	Replace suction/discharge valves.	Mech	Shutdown	PL	Situational	P/D Pump-R5	8.0	As required
13	Rebuild gear reducer.	Mech	Shutdown	PL	Situational	P/D Pump-R6	16.0	As required
14	Repair suction stabilizer.	Mech	Shutdown	PL	Situational	P/D Pump-R7	8.0	As required
15	Repair discharge dampener.	Mech	Shutdown	PL	Situational	P/D Pump-R8	8.0	As required
16	Replace connecting rod.	Mech	Shutdown	PL	Situational	P/D Pump-R9	16.0	As required
17	Replace plunger/adapter.	Mech	Shutdown	PL	Situational	P/D Pump-R10	16.0	As required
18	Adjust cross-head ball and socket.	Mech	Shutdown	PL	Situational	P/D Pump-R11	8.0	As required

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3.2 Predictive Maintenance

Parameters that monitor centrifugal pump failure modes are known as conditions to be monitored. These are the conditions that provide trendable data that can then be compared to baseline data and tolerances (performance criteria) to predict when a failure might occur. In such cases, pump maintenance can be planned and scheduled, thereby minimizing costs and repair time.

Conditions to be monitored can be determined from predominant failure modes that require monitoring due to frequency of occurrence and the failure effects on safety, environment, operations, or cost.

Ideal conditions to be monitored provide clear and unambiguous results that are either directly obtained or can be derived from measurable parameters. These parameters can be monitored periodically to provide component condition data that show degradation.

The following steps present a systematic method for selecting conditions to be monitored:

1. Determine all predominant failure modes for each critical component.
2. Determine failure causes.
3. Determine monitoring activities:
    a. Measurable parameters
    b. Derived indicators
    c. Adequacy of instrumentation
4. Determine applicability and practicality of monitoring:
    a. Cost
    b. Mode of operation during monitoring
    c. Availability of instrumentation
5. Develop monitoring rationale for each failure mode.

Because not all failure modes can be accurately monitored, many of the components will be removed from the list of conditions to be monitored. Monitoring is best employed on components that experience gradual degradation of measurable parameters throughout the life of the component.

Component performance criteria form the cornerstone of condition-based monitoring or predictive maintenance. Therefore, the criteria selected require rigorous evaluation. The aim of predictive maintenance is to determine the condition of components by a comparison of baseline data and actual data to detect incipient failures through trend analysis. For each condition to be monitored, a set of initial and minimum performance

criteria should be established. Initial performance criteria can be obtained from specification sheets, baseline operational (installation test) data, or (in lieu of the specifications or baseline data) during the time of first testing.

Performance criteria define acceptance limits for component performance in support of overall pump performance. Component performance criteria are determined as limits for the conditions to be monitored and can be defined by three limit classifications:

- Initial - performance specifications to which the component was designed
- Operational - minimum component performance below which the operational reliability or function of the pump would be impaired
- Safety - minimum component performance below which continued degradation would jeopardize personnel or equipment safety

Where differences exist, the most limiting of the operational or safety criteria should be applied as the minimum performance criteria.

3.2.1 Data Collection Methods

Data collection consistency is the key element in reliable trend analysis. Well-defined data collection procedures that allow repeatability of measurements should be used by qualified personnel when taking data that is to be used for the early diagnosis of incipient failures.

One of the most cost-effective and certainly the least intrusive data collection method is the daily operational inspection performed by the operators. When provided with inspection procedures and criteria, the pump room operators can collect reliable (if the systems are properly instrumented) and trendable data upon which maintenance decisions can be made. Accurate recording of actual operating hours is necessary and can be achieved by using the control room logs or the installation of hour run meters. For example, the operator could record:

- Fluid temperature and density (boron concentration)
- Pump flow
- Pump suction and discharge pressures and temperatures
- Motor current
- Motor temperature

These data elements could be compared to predetermined criteria and then compared to the pump data collected during performance testing. This data collection is in addition to the daily (or more frequent) inspections that focus on fluid leaks, lubricant levels, noises, and vibrations.

3.2.2 Data Analysis

The key to performing analysis leading to reliable diagnois is being able to correlate the degradation mechanisms with the symptoms. In addition to traditional trending methods, recent innovations in acoustic analysis, vibration pattern recognition, expert systems, and neural networks are being investigated for applicability in the industry.

A review of available data sources was conducted using the Nuclear Plant Reliability Data System (NPRDS), Operating Experience Code (OPEC), survey responses, and plant meetings. For the years 1990–1994, there were over 120 reported charging pump failures. Of these failures, over 80% involved positive displacement pumps.

Figure 3-1 shows overall fewer reported events for centrifugal pumps, and a significant decrease in the reported failures for the positive displacement pumps. In 1994 there were five reported failures for centrifugal pumps as compared to over 35 in 1991 (three years prior). Positive displacement charging pump failures showed a significant reduction of failures from 150 in 1990 to less than 20 in 1994.

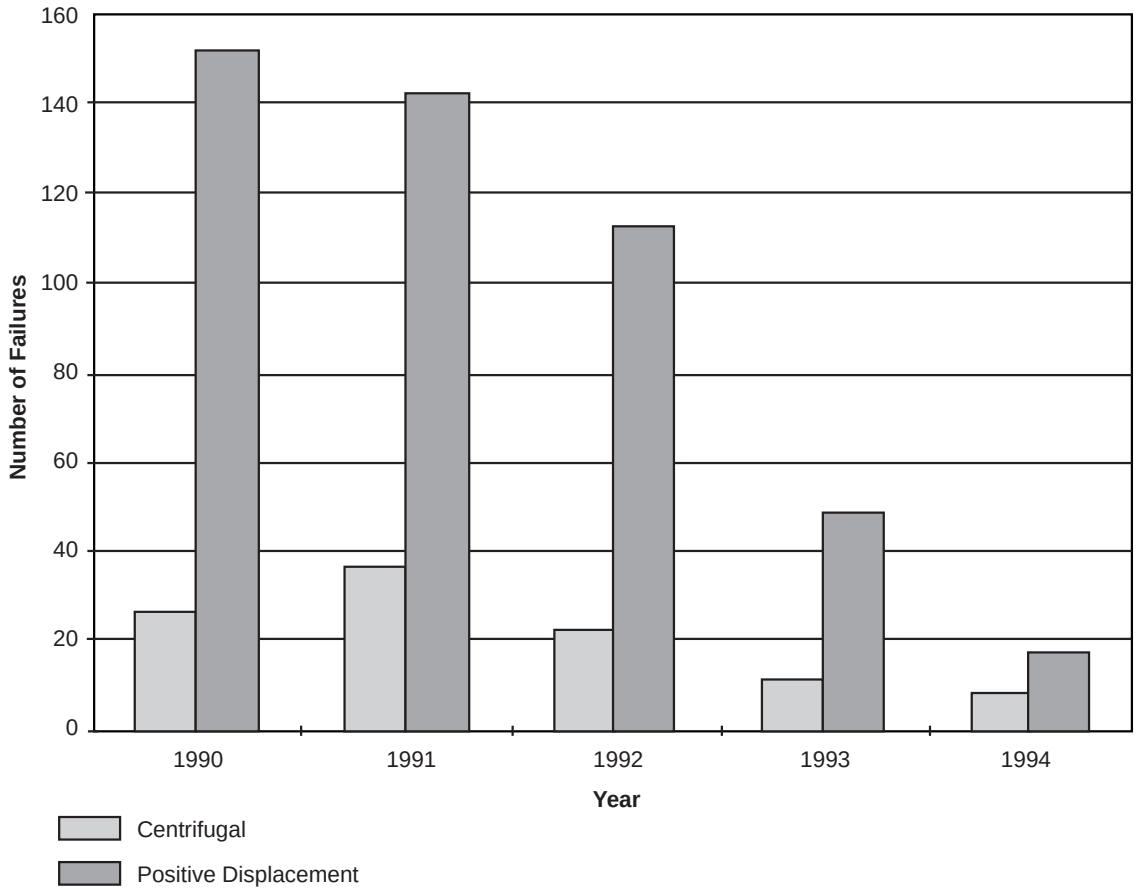


Figure 3-1
Charging Pump Failures (NPRSD)

Yearly failure rates for both pump types continue to show yearly improvement in reduced failures. This dramatic improvement trend for the positive displacement pumps was first considered to be a result, in part, of the reduced "running" of the positive displacement pumps. This improvement trend was confirmed by several plants operating only positive displacement pumps. The failure data presented here grouped the pump failures by pump type only. Further data reduction would be necessary to draw conclusions or make a comparative analysis between pump manufacturer, plant design, and operating time.

The six most significant areas for improving positive displacement performance are: packing, plunger, valves, cylinder block, pulsation dampener, and suction stabilizer. Figure 3-2 shows a summary of component failure causes, with the majority of failures (62%) related to packing.

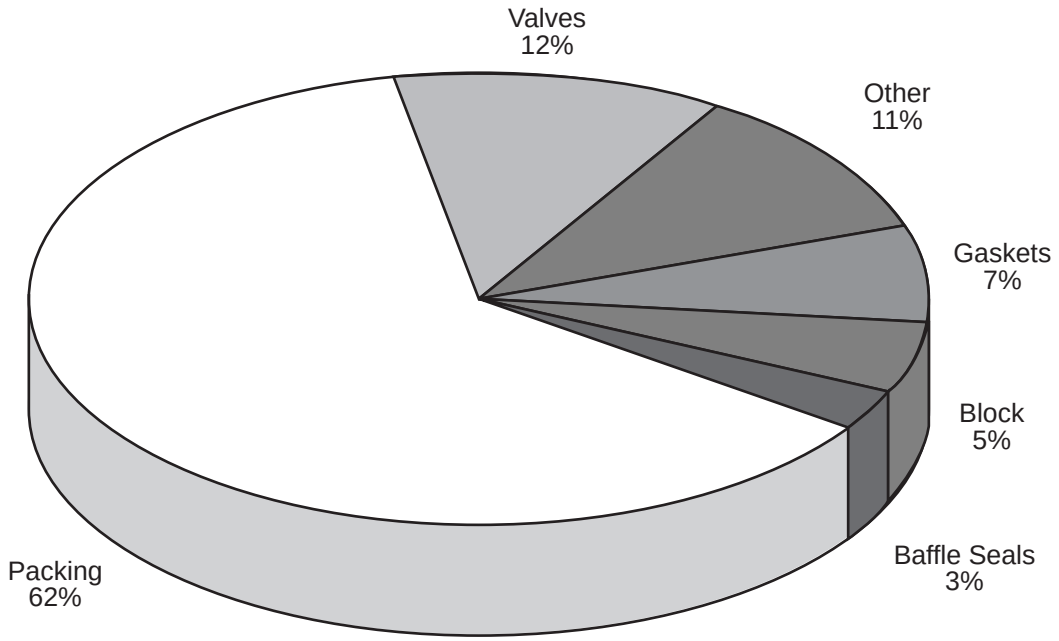


Figure 3-2
Positive Displacement Pump Component Failures

Figure 3-3 shows a breakout of packing failure causes. Fifty-eight percent of failures are attributed to normal packing wear. Other factors contributing to packing failures are abnormal wear, improper maintenance, manufacturing, and design.

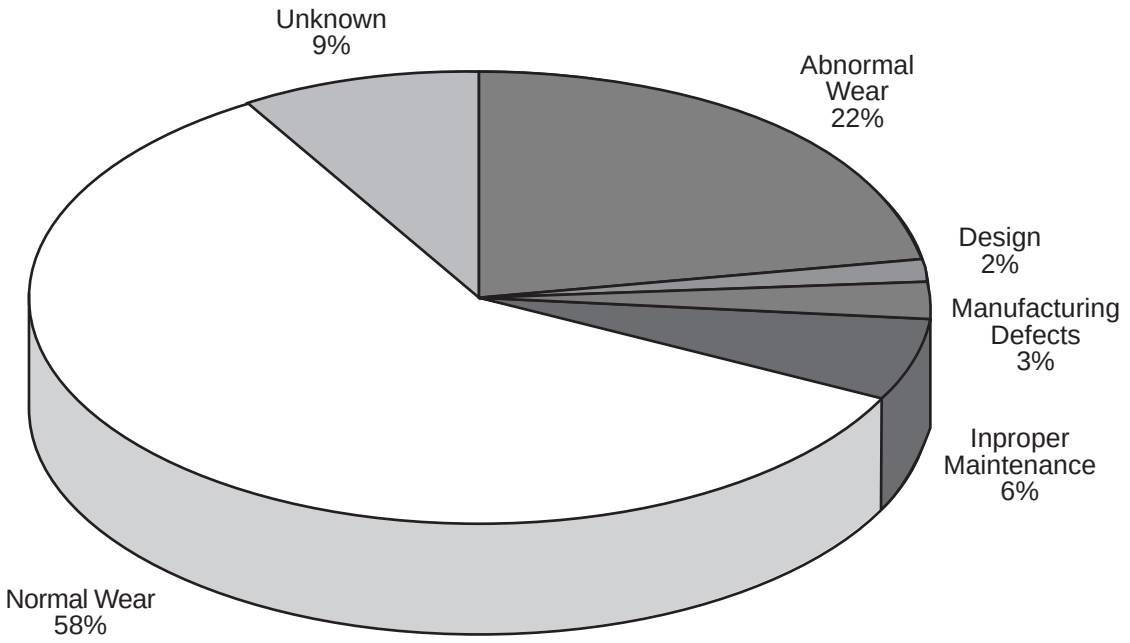


Figure 3-3
Positive Displacement Pump Packing Failure Causes

To summarize component discussions for positive displacement charging pumps:

- Low flow rate is one of the major reasons for the non-utilization of positive displacement charging pumps at nuclear stations that have installed both centrifugal and positive displacement charging pumps. Non-utilization of positive displacement pumps is not entirely based on reliability concerns.
- The decreasing trend in reported pump failures is consistent with utility opinion and experience.
- In the early 1990s, maintenance costs at some plants exceeded $125,000 annually, mostly attributed to packing changeouts.
- The failure data for positive displacement pumps may be skewed. The data do not consider standby versus normal operation.
- Plants are reporting packing failures differently. Plants are changing their definition of failure. Instead of packing failure being considered as failure, it may now be considered end-of-life or normal wear. This has been influenced by the increased life of the packing. For example, a high repack rate of 50–55 per year has been reduced to one repack in seven months.
- Operational considerations for improving performance includes running seal water in advance of starting, and maintaining pressure margin in suction (30–35 psi) to prevent hydrogen from coming out of solution.

The leading component failures for centrifugal charging pumps are in Figure 3-4: mechanical seals (40%), lubrication system including oil cooler and oil line leaks, rotating element, bearings, and pump casing. Figure 3-5 provides a year-to-year breakout of the major component failures. Issues with mechanical seals were highest in 1990 and 1991. Mechanical seals are the single largest identified component failure.

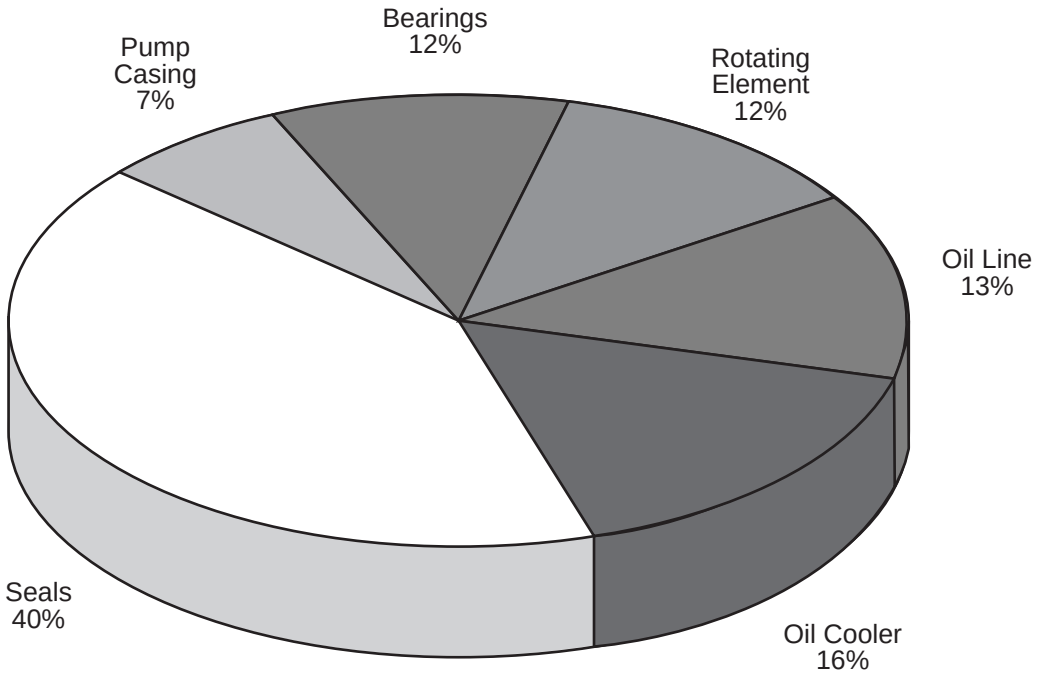


Figure 3-4
Centrifugal Pump Component Failures by Year

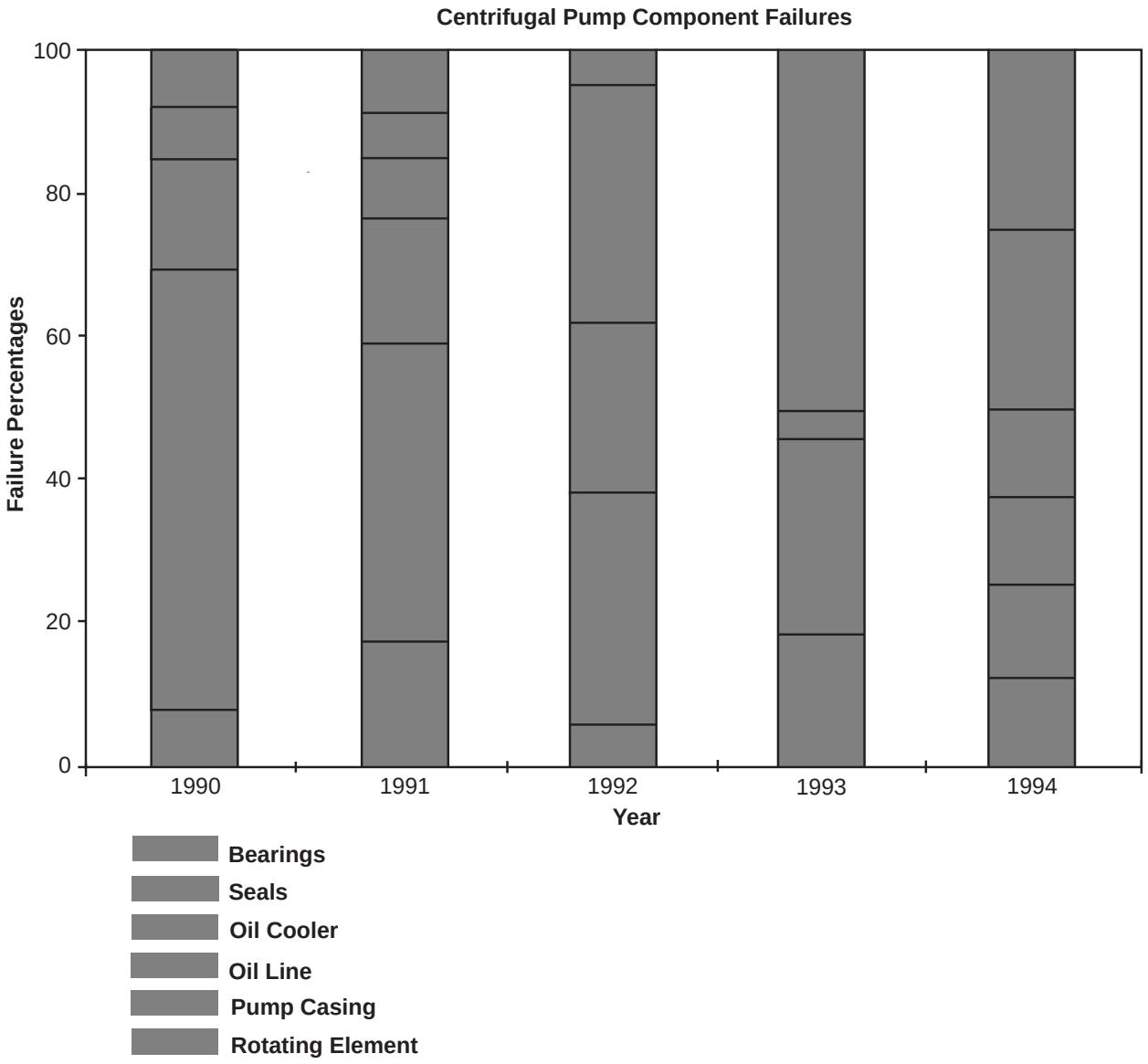


Figure 3-5
Centrifugal Pump Component Failures

Figure 3-6 shows the failure causes associated with mechanical seals. Normal seal wear and improper maintenance are reported as the two leading causes for failure.

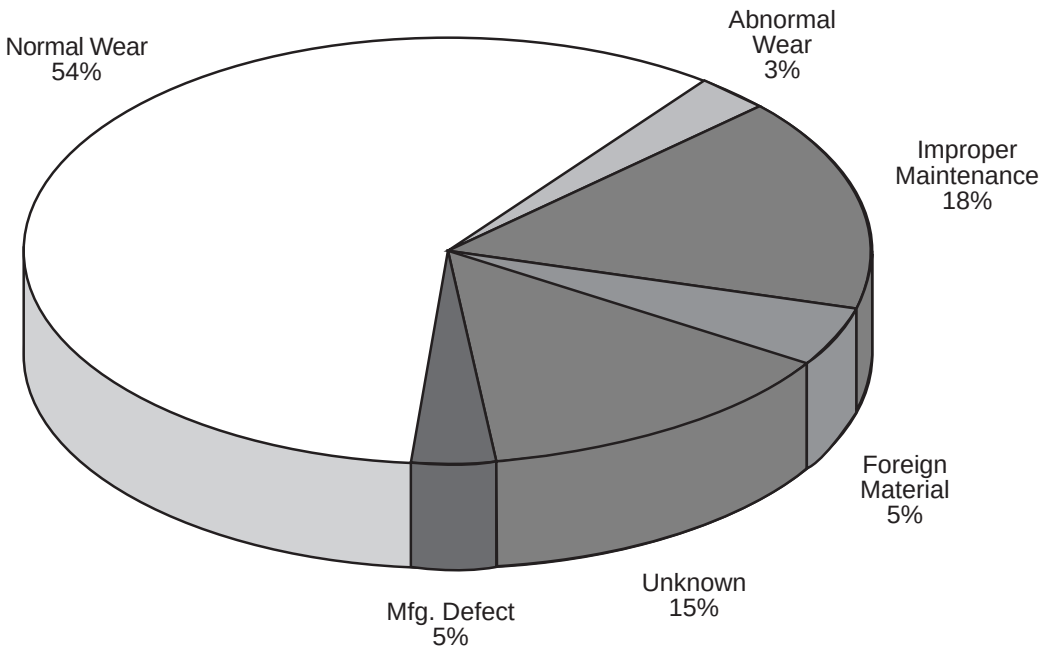


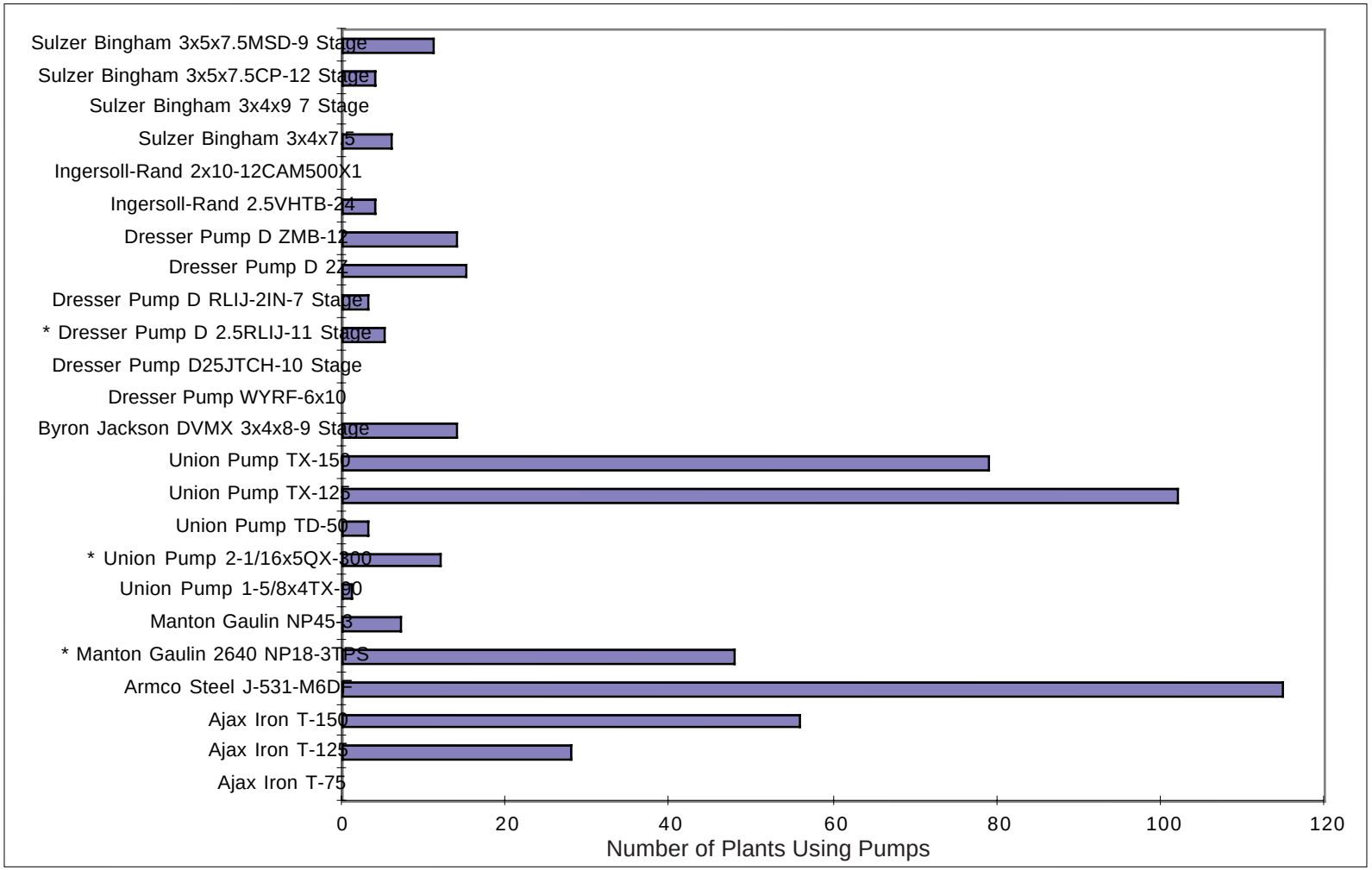
Figure 3-6
Mechanical Seal Failure Causes

To summarize component discussions for centrifugal charging pumps:

- The definition of acceptable leakage rate for mechanical seals varies from plant to plant. A leakage rate of 1 drop / sec is sufficient criterion to change out an in-service seal.
- Changes in either the balance line flow or the discharge flow control valve should be monitored as an indication of pump internal wear or degradation.
- Pump casing inspections and case cladding repair has been identified as a concern.

3.2.3 Failure Frequency

Reduction of the failure data can provide failure rate information. Figure 3-7 lists the charging pumps in use within the industry and the associated failure rates for the major components based on data obtained from the Nuclear Plant Reliability Data System (NPRDS). Although pump failures are reported, the data are of little value in determining the expected life of the charging pumps due to insufficient failure cause reporting. It does show, however, that the most predominant pump in the industry (Dresser) experienced the fewest average number of failures per year of plant operation (less than 1.0 pump failure per year per plant).



*Industry prevalent designs

Figure 3-7
Average Number of Charging Pump Failures Per Plant

3.2.4 Failure Modes

Reduction of the failure data can provide failure mode information if the definitions of component and equipment functions are universally applied in the data reporting system. This is not usually the case; therefore, the data must be verified and expanded to include input from experienced plant maintenance technicians and personnel. The following sections provide the definitions of functional failure modes that can be applied in the RCM process to determine maintenance tasks for charging pumps.

3.2.4.1 Mechanical Seals

Shaft seal failure is indicated when the seal fails to perform its designed function, which is to provide a seal between the pump shaft and casing, thus preventing fluid leakage from the high-pressure side to the low-pressure side or the atmosphere. Zero leakage is desired; however, leak rates of up to 60 drops per minute can be tolerated without significantly affecting seal performance. Above 60 drops per minute, frequent observations are recommended to determine if the condition is degrading.

Table 3-4
Mechanical Seals Failure Modes

Mechanical Seals		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Damaged or worn faces	Leaks, loss of fluid	CM: Inspection PM: Cleaning PL: Replace seal
Damaged O-rings or bellows	Leaks, loss of fluid	CM: Inspection PM: Cleaning PL: Replace seal

3.2.4.2 Oil Lines and Coolers

Oil line and cooler failure is indicated when the lines and cooler fail to perform their designed function, which is to contain and provide lubricant to the bearings at a temperature below 130°F with no leaks.

Table 3-5
Oil Lines and Oil Coolers Failure Modes

Oil Lines		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Loose fittings	Leaks, loss of lubrication	CM: Inspection PM: Tighten fittings
Fatigue crack	Leaks, loss of lubrication	CM: Inspection PM: Cleaning
Plugged line	Loss of sufficient lubrication	CM: Trend flow rate CM: Trend temperature
Oil Coolers		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Loose fittings	Leaks, loss of fluids	CM: Inspection PM: Tighten fittings
Fatigue crack	Leaks, loss of fluids	CM: Inspection PM: Cleaning
	Mixing of fluids	CM: Sample fluids
Fouled heat exchanger	Loss of sufficient cooling	CM: Trend flow rate CM: Trend temperature

3.2.4.3 Bearings

Bearing failure can occur for a variety of reasons. The major factors that, either by themselves or in combination, can lead to premature failure in service include:

- Incorrect fitting
- Excessive preloading during installation
- Insufficient or unsuitable lubrication
- Overloading
- Impact loading
- Vibration
- Excessive operating or environmental temperature
- Contamination by abrasive matter
- Ingress of harmful liquids
- Stray electrical currents

These factors can lead to wear conditions such as:

- Flaking
- Pitting

- Cracks or fractures
- Rotational creep
- Smearing
- Softening
- Indentation
- Fluting
- Corrosion

Table 3-6
Bearings Failure Modes

Bearings		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Wear	Excessive vibration	CM: Vibration analysis CM: Motor current analysis PM: Lubricate
	Misalignment	CM: Vibration analysis PL: Realign

3.2.4.4 Pump Casings

Casing failure is indicated when the casing fails to perform its designed function, which is to contain the fluid and provide stability to pump internal elements.

Table 3-7
Pump Casing Failure Modes

Casing		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Loose sections	Leaks	CM: Inspection PM: Tighten bolts
	Shaft misalignment	CM: Vibration analysis

3.2.4.5 Impellers

Impeller failure is indicated when the impeller fails to perform its designed function, which is to cause the fluid to flow at a rate of 150 gallons per minute at a pressure of 2800 psi (Pacific Dresser).

Table 3-8
Impeller Failure Modes

Impellers		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Out of balance	Wobble	CM: Vibration analysis
Fouled	Low/no flow	CM: Performance test
Worn	Low flow	CM: Performance test

3.2.4.6 Couplings

Drive coupling failure is indicated when the coupling fails to perform its designed function, which is to connect the pump shaft to the drive motor shaft, transmitting torque and axial thrust while accounting for misalignment.

Table 3-9
Coupling Failure Modes

Couplings		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Misaligned Worn Cracked Shims	Wobble	CM: Vibration analysis

3.2.4.7 Shafts

Shaft failure is indicated when the shaft fails to perform its designed function, which is to support and align the pump impeller within the casing, transmitting torque rotational force from the coupled driver to the impeller.

Table 3-10
Shaft Failure Modes

Shafts		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Warped	Wobble	CM: Vibration analysis
Cracked, Fractured	No flow	CM: Performance test

3.2.4.8 Fluid Cylinders

Fluid cylinder failure occurs when the fluid cylinder fails to contain and direct the flow of liquid to the proper location, resulting in leakage from the fluid cylinder. Typically, this occurs at the areas of the block that have been bored out for the suction and discharge check valves, starting with a small crack and propagating into larger cracks that allow block leakage.

Table 3-11
Fluid Cylinder Failure Modes

Fluid Cylinder		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Cracking	Leaks High seal water pressure	CM: Inspection CM: Seal water system pressure monitoring

3.2.4.9 Fluid Cylinder Valves

Valve failure occurs when the valves fail to perform their designed function, which is to control the flow of liquid into and out of the fluid cylinder. Failure can result in damage to other pump internal components due to pressure buildup in the pump or the inability of the pump to deliver sufficient pressure if the valves (particularly the suction valve) are stuck in the open position.

Table 3-12
Fluid Cylinder Valve Failure Modes

Fluid Cylinder Valves		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Sticking	Loss of pressure/flow	CM: Performance test Acoustic monitoring
Broken	Loss of pressure/flow	CM: Performance test Acoustic monitoring
Excessive wear	Leaking or reduction in flow/capacity	CM: Performance test Acoustic monitoring

3.2.4.10 Packing

Packing failure occurs when the packing fails to perform its designed function, which is to prevent water leakage along the shaft to the atmosphere and, in some cases, prevent air leakage into the pump. The normal life span of the packing should be in the range of

5,000–10,000 hours. The most common modes of packing failure are wear caused by an unsatisfactory plunger finish, inadequate seal water supply, or using an inappropriate packing.

Table 3-13
Packing Failure Modes

Packing		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Worn	Leaks (external)	CM: Inspection
Blown cylinder packing	Reduced capacity	CM: Seal water system pressure transient high

3.2.4.11 Plungers

Plunger failure occurs when the plunger fails to perform its designed function, which is to interact with the cylinder bores, packing, and valves to supply sufficient pressure to the liquid being pumped to meet the system head requirements. The most common mode of plunger failure is unsatisfactory surface finish, which degrades the packing and allows leakage out of the system.

Table 3-14
Plunger Failure Modes

Plunger		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Worn	Leaks	CM: Inspection

3.2.4.12 Fluid Cylinder End Cap Gaskets

Fluid cylinder end cap gaskets are considered failed when they fail to maintain a leak-free seal between the fluid cylinder and the atmosphere.

Table 3-15
Fluid Cylinder End Cap Gasket Failure Modes

Fluid Cylinder End Cap Gaskets		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Deformed	Leaks	CM: Inspection

3.2.4.13 Plunger Packing Seal Lubricant/Cooling

The plunger packing seal lubricant/cooling system is considered failed when it fails to perform its designed function, which is to lubricate and cool the packing, which prevents overheating and failure of the packing.

Table 3-16
Plunger Packing Seal Lubricant/Cooling Failure Modes

Plunger Packing Seal Lubricant/Cooling		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Loose fittings	Leaks, loss of lubrication Overheated packing	CM: Inspection PM: Tighten fittings
Plugged line	Loss of sufficient lubrication	CM: Trend flow rate CM: Trend temperature

3.2.4.14 Gears

Gear failure occurs when the gears fail to perform their designed function, which is to develop mechanical advantage and transfer torque from the drive shaft to the crankshaft.

Gear sets are limited to four types of failure:

- Wear (abrasive and corrosive)
- Scoring
- Pitting
- Tooth breakage

High viscosity oil and additives are effective in preventing gear wear.

Abrasive wear is usually caused by foreign material in the lubricant. It can occur with any type of gear, under any operating conditions. As shown in Figure 3-8, the introduction of large abrasive materials results in the destruction of the involute profile, with pitting around the pitch line.

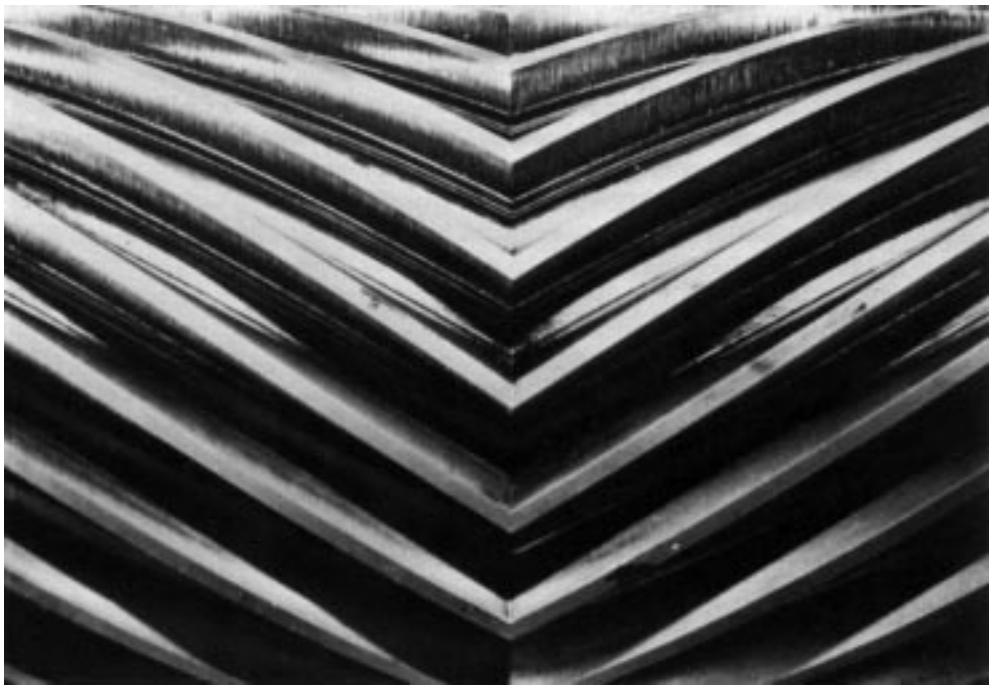


Figure 3-8
Abrasive Gear Wear
From *CRC Handbook of Lubrication (Theory and Practice of Tribology), Volume II, Theory and Design*

The choice of lubricant has little to do with minimizing abrasive wear. Cleaning the system and replacing the oil is the only effective treatment.

Corrosive wear is caused by contaminants, such as water, in the lubricant. In addition to the wear of the meshing surfaces, corrosive wear is usually found on other gear parts as well. If wear on other parts is not evident, chemical analysis of the oil might be necessary to distinguish between corrosive and abrasive wear.

Scoring of gears is caused by inadequate load-carrying ability of the lubricant. It results in a rough tooth surface with evidence of having been welded to its mating surface by the frictional heat and then torn loose as the gears rotate. Figure 3-9 shows a typical scoring pattern, excessively worn on the gear teeth and roots, with the pitch line intact.

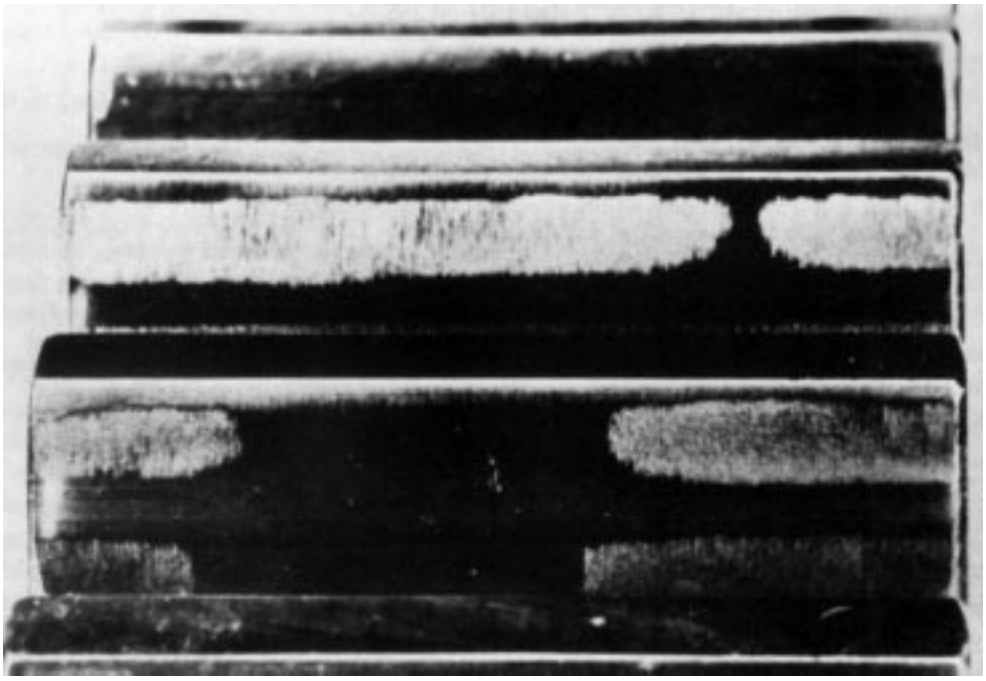


Figure 3-9
Scoring
From *CRC Handbook of Lubrication (Theory and Practice of Tribology), Volume II, Theory and Design*

Table 3-17
Gears Failure Modes

Gears

Failure Modes	Effects	Applicable Maintenance Task
Wear	Excessive vibration	CM: Vibration analysis CM: Motor current analysis PM: Lubricate
	Misalignment	CM: Vibration analysis PL: Rebuild
Broken teeth	Excessive vibration	CM: Vibration analysis CM: Motor current analysis PL: Rebuild

3.2.4.15 Stabilizer and Dampener Bladders

Stabilizer and dampener bladder failure occurs when the bladders fail to perform their designed function, which is to maintain sufficient nitrogen pressure to compensate for surges in system pressure. This is accompanied by increased vibration levels, especially when the discharge bladder fails. Failure of the suction bladder most likely results in cavitation in the pump due to the reduced net positive suction head available.

Table 3-18
Stabilizer and Dampener Bladder Failure Modes

Stabilizer and Dampener Bladder		
Failure Modes	**Effects**	**Applicable Maintenance Task**
Rupture	Gas entrainment	CM: Inspection
	Cavitation	CM: Vibration analysis
Pressure leak	Increase surges	CM: Vibration analysis
		CM: Acoustic monitoring

3.3 Preventive Maintenance

Preventive maintenance includes basic activities to sustain or prolong an existing equipment condition. Tasks such as cleaning, preserving, and lubricating are preventive maintenance tasks. They are designed to prevent or minimize equipment degradation and, therefore, prolong useful life. Complete overhauls, scheduled with fixed frequency, were once considered preventive in nature. They are not preventive and, in fact, usually increase overall failure rates by introducing break-in mortality into an otherwise stable system.

3.4 Failure Case History, Gaulin Pump

A slight leak was observed on the end cap of the number 3 cylinder. The suspected cause was a leaking end cap gasket. Maintenance was performed such that the gasket was replaced (nylon gasket material). No unusual indications were noted on the cap or the block sealing surfaces. The cap was torqued and the pump returned to service. During performance of the post-maintenance test, the leak was observed to return, and the test failed. The pump front cap was removed again, and the block was closely inspected in the gasket seating area. A small crack was observed in the gasket seating area. The pump cylinder was unpacked, and dye penetrant tested. A crack was found in the cylinder that had propagated from the suction valve bore / plunger bore intersection. The fluid cylinder was replaced with a new 17-4 PH block with increased radii and shot-peened bores. It was repacked and returned to service.

The cause of failure was attributed to high cyclic fatigue. It is unknown how many hours of service the fluid cylinder had been in service.

3.5 EdF Charging Pumps

3.5.1 Charging Pump Design and Construction

Electricité de France (EdF) operates two different types of PWR units[1]: 900 MW reactors (34 units) and 1,300 MW reactors (20 units). The 900 MW reactors have three pumps; the 1,300 MW have only two. During normal operation, only one pump at each reactor is running. All the pumps are centrifugal, and the supplier is Guinard (Guinard Pumps has been owned by KSB since 1992). Figure 3-10 shows a typical 900 MW unit charging pump system.

3.5.1.1 900 MW Unit Pumps

One hundred and two charging pumps are in operation in the 900 MW units. These charging pumps can be operated in two ways:

- Nonstop service for the seal water injection at the reactor coolant pumps and also the primary circuit and chemical and volume control system feeding.
- Safety service in accidental conditions to feed the high-pressure safety injection system. The pumps dedicated to this operation are running only for tests; their operating hours are low, but they perform frequent start-ups.

The pumps are of horizontal axis and are composed of:

- An electric motor with the following characteristics:
  - — Voltage 6,600 V
  - — Power 710 kW
  - — Speed 1,500 rpm
- A gearbox, lubricated with an oil pressure system. The output-to-input speed ratio increase is 3.026. The first 30 units were supplied with Citroen gearboxes and the last four units with Comelor gearboxes.
- A 12-stage pump, with a thrust bearing lubricated by the gearbox oil pressure system and two hydrostatic bearings lubricated by the pumped water. The shaft tightness is provided with mechanical seals.
- An auxiliary electrical pump providing the pressure in the gearbox and pump oil system during start-up. In operation, the oil pressure system is fed by a gear direct driven pump.

Figure 3-11 shows a longitudinal view of a 900 MW unit pump.

[1] This information, compiled by the Central Maintenance Department of EdF for all of their operating units, is current through the end of 1994.

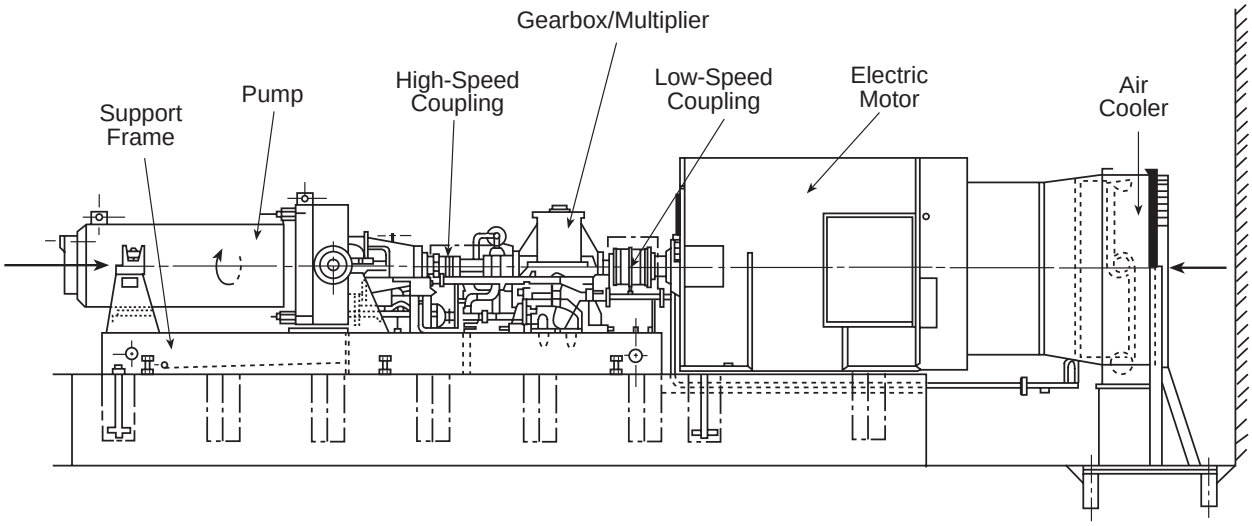


Figure 3-10
Typical 900 MW Unit Charging Pump System

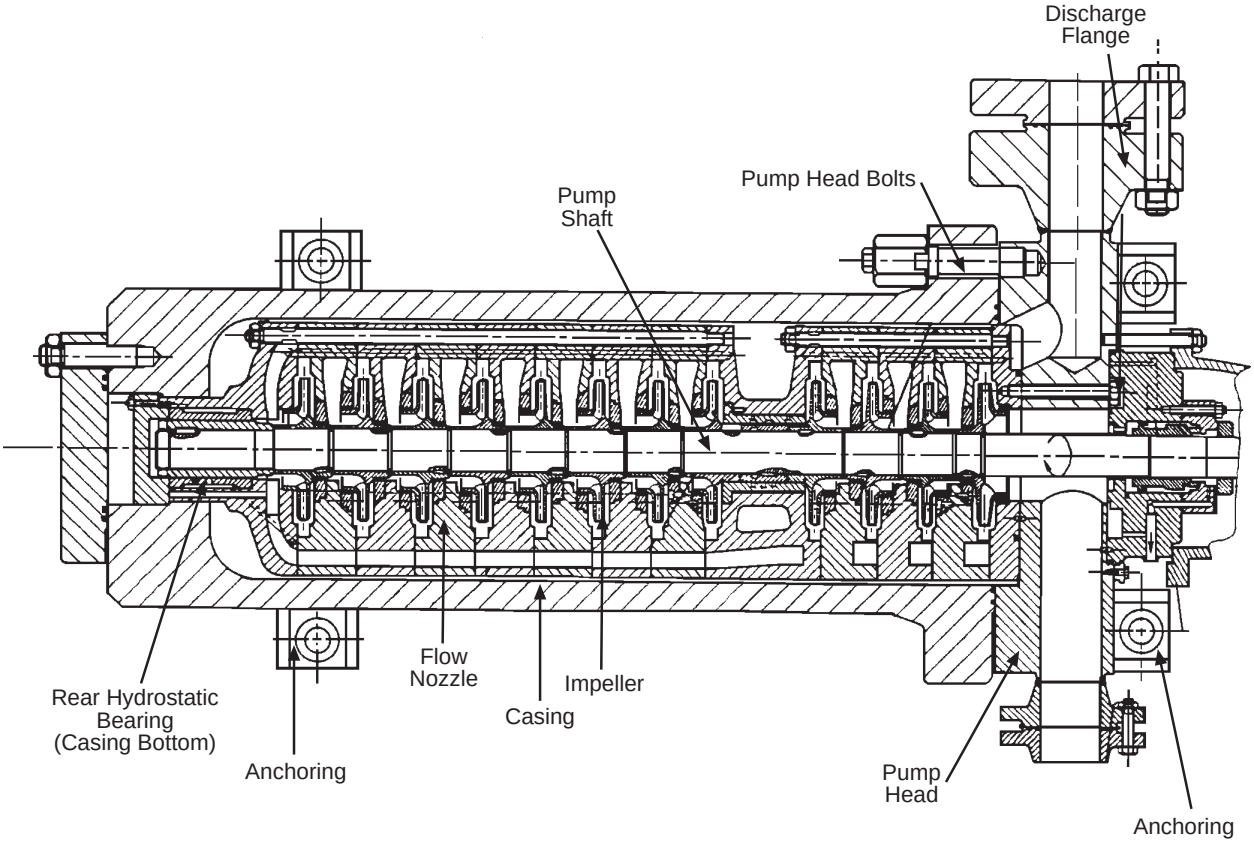


Figure 3-11
Longitudinal View of a 900 MW Unit Charging Pump

The main characteristics of the pumps operating in 900 MW units are listed in Table 3-19.

Table 3-19
Main Characteristics of 900 MW Unit Charging Pumps

Characteristic	Unit	Charging Operation	Safety Operation
Fluid		Borated water	Borated or containment sump water
Density	t/m^3 $lb./ft.^3$	1 to 1.04 62.43 to 64.93	1 to 1.04 62.43 to 64.93
Flow rate	m^3/h gpm	34 149.7	148.2 652.6
Head	mCF ft.	1,767 (+35, -7) 5,798 (+115, -23)	730 (+20, -20) 2,395 (+66, -66)
NPSH	mCF ft.		8 26
Suction pressure	bar psi	5.3 max. 76.9 max.	10 max. 145 max.
Discharge pressure	bar psi	189.1 max./180.5 min. 2,743 max./2,618 min.	86.5 max./73.6 min. 1,255 max./1,067 min.
Suction temperature	°C °F	55 max./46 min. 131 max./115 min.	130 max./7 min. 266 max./45 min.
Shutoff head	mCF ft.	1,830 (+30, -0) 6,004 (+98, -0)	

Due to early failure problems, the following modifications have been implemented on the pumps:

- Gearbox
  - — New machining of the multiplying linkage to improve the quality of the contacting surface of the gear teeth
  - — Modification of the high-speed slinger because of inadequate tightness
  - — Adding exhaust holes on the bearing heads and replacement of the vent pipes (oil leaks due to the motor ventilation)
- Couplings
  - — Replacement of the grease ports with plugs

  - — Adding a ring with two O-ring gaskets inside the high-speed coupling (mitigation of the oil leaks)
  - — Replacement of the original high-speed gear coupling with a flexible metal disk coupling
- Oil circuit
  - — Replacement of the rubber flange gaskets with Supranite Superoil gaskets
  - — Injection of Vaprotec oil in the oil tank to prevent corrosion
  - — Modification of the calibration of the pressure switches and replacement of the Tombac bellows with Perburnan diaphragms
  - — Replacement of the 40 µm filters with 100 µm filters, after operating a while
  - — Removal of the filter bypass
- Pump
  - — Installation of impeller wear rings with labyrinth seals (improvement of protection against the particles that are in suspension in water)
  - — Modification of the water supply of the hydrostatic bearing located at the casing bottom (supplied with the discharge of the sixth impeller and return to the fifth impeller suction with a branching at the rear of a pressure reduction ring)

3.5.1.2 1,300 MW Unit Pumps

Forty charging pumps, type HMBS 11LL V3-3-12-45, are in operation in the 1,300 MW units. These pumps are run for the seal water injection at the reactor coolant pumps and also for the primary circuit and chemical and volume control system feeding. Figure 3-12 shows a longitudinal view of a 1300 MW unit pump.

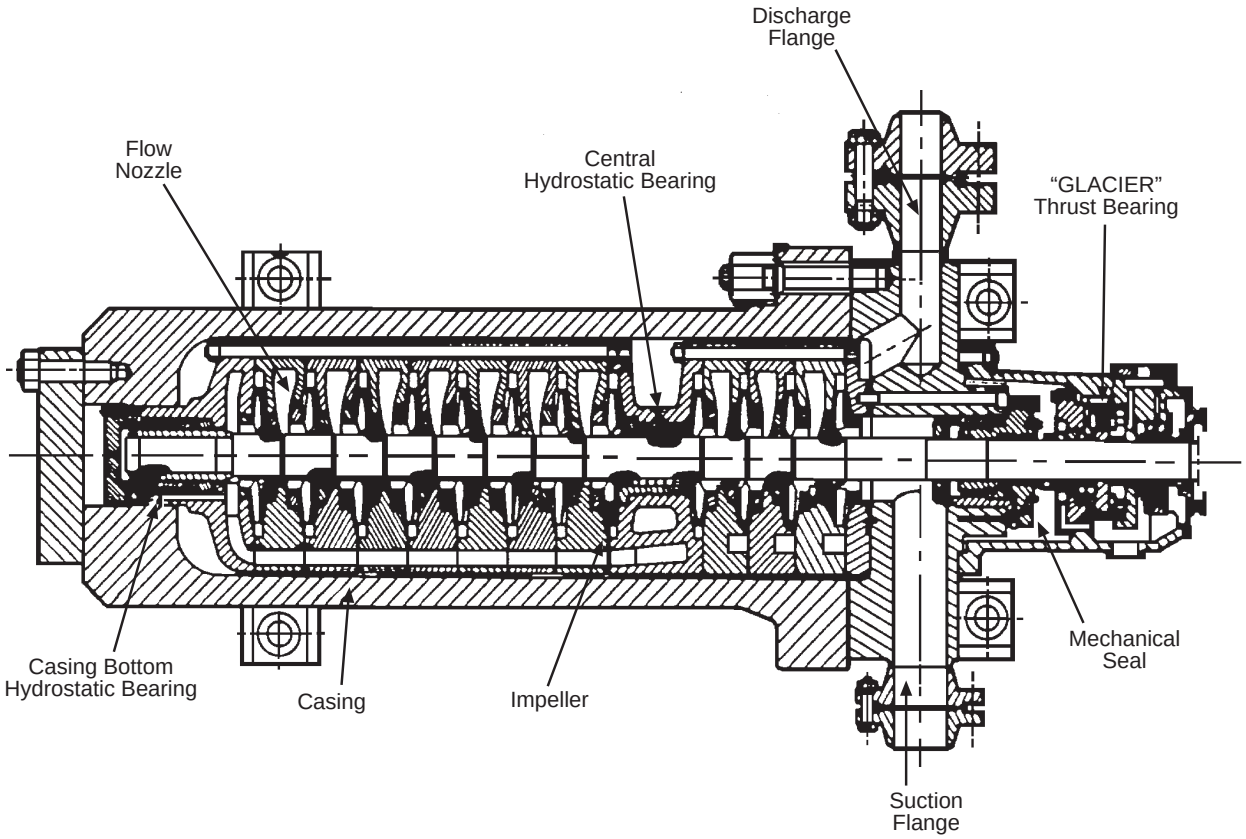


Figure 3-12
Longitudinal View of a 1300 MW Unit Charging Pump

The main characteristics of the pumps operating in the 1,300 MW units are listed in Table 3-20.

Table 3-20
Main Characteristics of 1,300 MW Unit Charging Pumps

Characteristic	Unit	Minimum Flow	Rated Flow	Maximum Flow
Fluid		Borated water	Borated or containment sump water	
Density	kg/m^3 $lb./ft.^3$	1,000 to 1,020 62.43 to 63.68	1,000 to 1,020 62.43 to 63.68	1,000 to 1,020 62.43 to 63.68
Flow rate	m^3/h gpm	8 35.2	44 194	100 440
Head	mCF ft.	1,950 6,398	1,770 5,807	900 2,953
NPSH	mCF ft.	< 6 < 19.7		
Suction pressure	bar psi	5 72.5	5 72.5	15.5 225
Suction temperature	°C °F	110 max./> 7 min. 230 max./> 45 min.	40 max./7 min. 104 max./45 min.	40 max./20 min. 104 max./68 min.
Shutoff head	mCF ft.	1,850 minimum and 2,000 maximum 6,070 minimum and 6,562 maximum		

3.5.2 Reported Problems and Failures

3.5.2.1 900 MW Units

The reported problems and failures are described in relation to the involved component.

3.5.2.1.1 Gearbox/Multiplier

This component usually fails from teeth flaking and pitting, which leads to the replacement of the toothed wheel and the pinion. The root cause of the failure is the action of the pinion on the toothed wheel in cases of major misalignment, as often observed. This misalignment is due to a lack of stiffness of the support frame.

3.5.2.1.2 Low- and High-Speed Couplings

Flaking of several teeth was noted on the coupling sleeves located at the low-and high-speed coupling connection, which led to the replacement of the coupling.

3.5.2.1.2.1 Low-Speed Couplings

The low-speed couplings supplied by Citroen need grease added frequently. This addition of grease is done after the installation of a grease port, which is then removed and replaced with a plug to prevent leaks.

The following remedial actions have been implemented to avoid grease issues and reduce vibration levels:

- Inspecting the couplings and replacing the grease every year. After cleaning, using a grease with a low dispersing capacity.
- Improving the tightness of the high-speed coupling to avoid the grease degradation and the associated oil products.
- Checking the alignment after disassembling the spacer and adjusting the fixed points (centering pin and expansion key).
- Balancing the coupling plates.
- Checking the tightness of the pump anchor bolts (30m daN / 2,552 inch lb).

The low-speed couplings supplied by Comelor are oil-lubricated, rather than grease-lubricated.

3.5.2.1.2.2 High-Speed Couplings

The grease leakage occurring at the high-speed couplings has been solved by changing the original design to a metal disk coupling.

3.5.2.1.3 Main Pump

3.5.2.1.3.1 Mechanical Seal/Package Failure

A water leak occurred on a pump due to a mechanical seal failure. The root cause of the leak was the loosening of six screws. This problem was solved by locking the screws with Loctite.

3.5.2.1.3.2 Rear Hydrostatic Bearing (Located at the Casing Bottom)

Abnormal wear has been seen after only a few thousand hours of operation. Improvements have been made by the manufacturer to the design of the water supply system of this hydrostatic bearing, the condition of which is the single factor that determines the pump inspection interval. These modifications consist of:

- Supplying water with a tap that originates from the sixth stage and returns to the fifth stage suction. This supply results in an increased bearing stiffness. Moreover, the tap occurs in a location where the particles are not likely to build up. This decreases the risk of fretting for the bearing when starting the pump.
- Machining of labyrinths in the wear rings.
- Modifying the shape of the bearing support location, in order to decrease the occurrence of particle build-up. It is of major importance when a pump starts that its supply water is clean, the air in the pipes has been evacuated, and no contamination occurred during the maintenance of the reactor chemical control circuit.

3.5.2.1.4 Rotor Balancing

Two balancing problems occurred. A systematic vibration inspection implemented during start-up should prevent this type of event.

3.5.2.1.5 Thrust Bearing

In one case, the damaging of the pads might be related to significant vibrations, which led to increased rotor displacement.

3.5.2.1.6 Impellers

Some cracking of the extremity of a few impeller blades was seen after disassembly. The root cause was mainly a fatigue phenomenon that occurred at very specific locations. The blades involved in this failure were those that had been shop-machined to obtain suitable impeller balancing to the point that the remaining thickness was not sufficient for good mechanical behavior. This led to an extensive checking of impeller blade thickness and, therefore, to the replacement of all the undersized blades.

3.5.2.1.7 Wear Rings Locking Screws

Some screws loosened on several pumps despite the presence of a locking device. The root causes of this problem were:

- The screw material (AFNOR : Z2 CND 17 - 12 / AISI 316 L) was not suitable (yield strength and ultimate tensile strength too low).
- Some embrittlement occurred due to errors in manufacturing.
- The process implemented for the locking device led to the screws' embrittlement; therefore, this process is not suitable for this application.
- The required torque was too low.
- The locking devices were not of a captive design.

The following remedial actions were implemented for the loosened screws:

- Machining new screws in a material with high mechanical properties: AFNOR Z6 CND 16 - 04 (approximately ASTM A 564-83 grade 634)
- Machining instead of bending the locking devices
- Applying a higher torque

3.5.2.1.8 Shaft

Some fretting occurred at the thrust collar location. At one plant, the remedial action consisted of implementing a hardening surface treatment. Implementing this remedial action for the other EdF plants will be determined by its success at this first plant.

High vibrations close to the oil-lubricated bearing have been reported from several units. This problem might be related to a resonance phenomenon. In fact, the bearing has a frequency response close to a harmonic in relation with the number of blades of the impeller (five). The root cause of this is the lack of stiffness of the support frame. Some units mitigated the problem by implementing different adjustments of the alignment or tightening the pipes, supports, bolts, or flanges.

3.54.2.1.9 Loose Parts

During a functional test while a pump was operating with no flow rate, its discharge pressure was lower than requested, but its vibration amplitudes were lower than the alarm threshold. During the following operational test, the pump was unable to deliver either the required pressure (60 bar / 870 psi instead of 90 bar / 1305 psi) or the expected flow rate. After 90 minutes, the pump was stopped due to excessive vibration. Disassembly revealed the presence of a cleaning rag and a significantly deformed metallic part. These items originated from maintenance work carried out on pipes connected ahead of the pump.

The locking of the rotor was found on a pump during an inspection when checking its axial clearance. The pump disassembly revealed that the root cause of the locking was the presence of an unidentified particle at the first impeller leak-tight ring location.

3.5.2.1.10 Miscellaneous

Two pumps at one unit were badly damaged when both were operating at a low flow rate.

3.5.2.1.11 Oil Circuit

3.5.2.1.11.1 Oil Leaks

A number of pumps experienced oil leaks, mainly at the flange junctions and the threaded piping connections. The gaskets could not compensate for the lack of parallelism between the flanges, which was a result of their being welded to the pipes.

During transients, some failures of the metallic bellows of the pressure switches occurred. This led to replacement of the bellows with devices equipped with Perbunan diaphragms. Some casing deformations caused leaks at the casing flange; this problem was solved by machining the sealing surface. To mitigate these problems, it was decided to:

- Install new flange gaskets, with a careful inspection of the condition of the flange surface
- Apply a leak-tight paste to the threads of the threaded piping connections

3.5.2.1.11.2 Oil Temperature Increase

A change in the original design of the pump room (adding fire protection walls) resulted in a decrease of the pump air cooling capacity. This led to an increase in the room and oil temperatures and, consequently, to an increase in the oil-lubricated bearing temperature. Remedial actions (for example, a better air temperature or flow adjustment) are still in progress to solve this problem.

3.5.2.1.11.3 Loose Parts

Some problems occurring at the oil system of a pump have been related to the presence of a cleaning rag in the direct driven oil pump suction.

3.5.2.1.11.4 Oil Cooler Leakage

This has been a plant-specific issue due to the fact that the cooling water was the service water. The chiller tube bundle of the speed increaser lubrication system, originally made of copper, has been replaced with a copper-nickel base alloy.

3.5.2.1.12 Concrete Support Cracking

The cracking occurred at two locations:

- At some solid concrete build-up. The cracks were due to thermal shocks or to a lack of bonding of the build-up to the support.
- At the solid support itself. These cracks could be due to high stresses or vibrations.

3.5.2.2 1,300 MW Units

The reported problems and failures are described in relation to the involved component.

3.5.2.2.1 Multiplier

3.5.2.2.1.1 Misalignment

A misalignment between the pump shaft and the high-speed shaft of the multiplier, which was lower than the required position (1 mm / 40 mils instead of 0.09 mm / 3.6 mils), resulted in damage to the high-speed bearing and led to its replacement. The radial clearance at the high-speed bearing bushing location (toward the coupling) was 0.45 mm / 18 mils, for a design value of 0.15 mm / 6 mils.

3.5.2.2.1.2 Loose Parts

A multiplier was very noisy and showed high-amplitude vibration (120 µm / 4.8 mils) at the high-speed coupler bearing. After replacement, an examination of the damaged multiplier showed that the clearance between the teeth was exceeding the maximum specified value. The root cause was a lack of lubrication due to the presence of a weld particle plugging the oil injection area on the high-speed bearing side of the multiplier.

3.5.2.2.2 High-Speed Couplings

Most of the reported problems are related to lack of lubrication:

- A lack of oil occurred when glued O-rings for the sleeves were used instead of molded O-rings.
- Several teeth marks or wear marks and, in some cases, high vibration have been correlated with lack of oil. To mitigate these oil leakages or deficiencies, the same corrective action as for the 900 MW units (replacement of the original coupling with a flexible metallic disk) is planned.

3.5.2.2.3 Main Pump

3.5.2.2.3.1 Mechanical Seal/Package Failure

Crystallized boron was found on one pump at the Latty mechanical seal location. Inspection of the seal revealed some cracks and chips on the carbon ring, a significant wear at the carbide ring periphery, and some heating marks on the stuffing box shaft sleeve. The root cause of this was a misalignment of the pump; one of its axes was located 6 mm (0.24”) too high. This problem was solved by the seal replacement and a better pump / multiplier alignment.

3.5.2.2.3.2 Shaft

Some marks of “fretting corrosion” have been found at the thrust sleeve location. The remedial action consisted of carrying out a hardening surface treatment at this location.

3.5.2.2.3.3 Loose Parts

A rear hydrostatic bearing inspection showed some scratches located on the bearing surface of the sleeve. The root cause was the presence of rubber pieces that were plugging two out of five diaphragms. This led to the bushing shell replacement.

3.5.3 Conclusion

The main problem remaining on these pumps is the lack of lubrication of the hydrostatic bearing during start-up. Once the pressure is established in the system, the bearing lubrication is efficient. This is the reason why the more suitable parameter to take into account to assess the state of the pump and the inspection intervals is not the number of running hours but the number of start-ups.

3.5.4 Charging Pump Maintenance

3.5.4.1 900 MW Units

Due to the differences in operating conditions, the pumps have been divided into two classes:

- The **operating pumps**, which run more than 500 hours within a fuel cycle. In the following text, the maintenance activities related to these pumps will be referenced with an O.
- The **safety/standby pumps**, which run less than 500 hours within a fuel cycle; their maintenance activities will be referenced with an S.

The maintenance operations are implemented according to Figure 3-13.

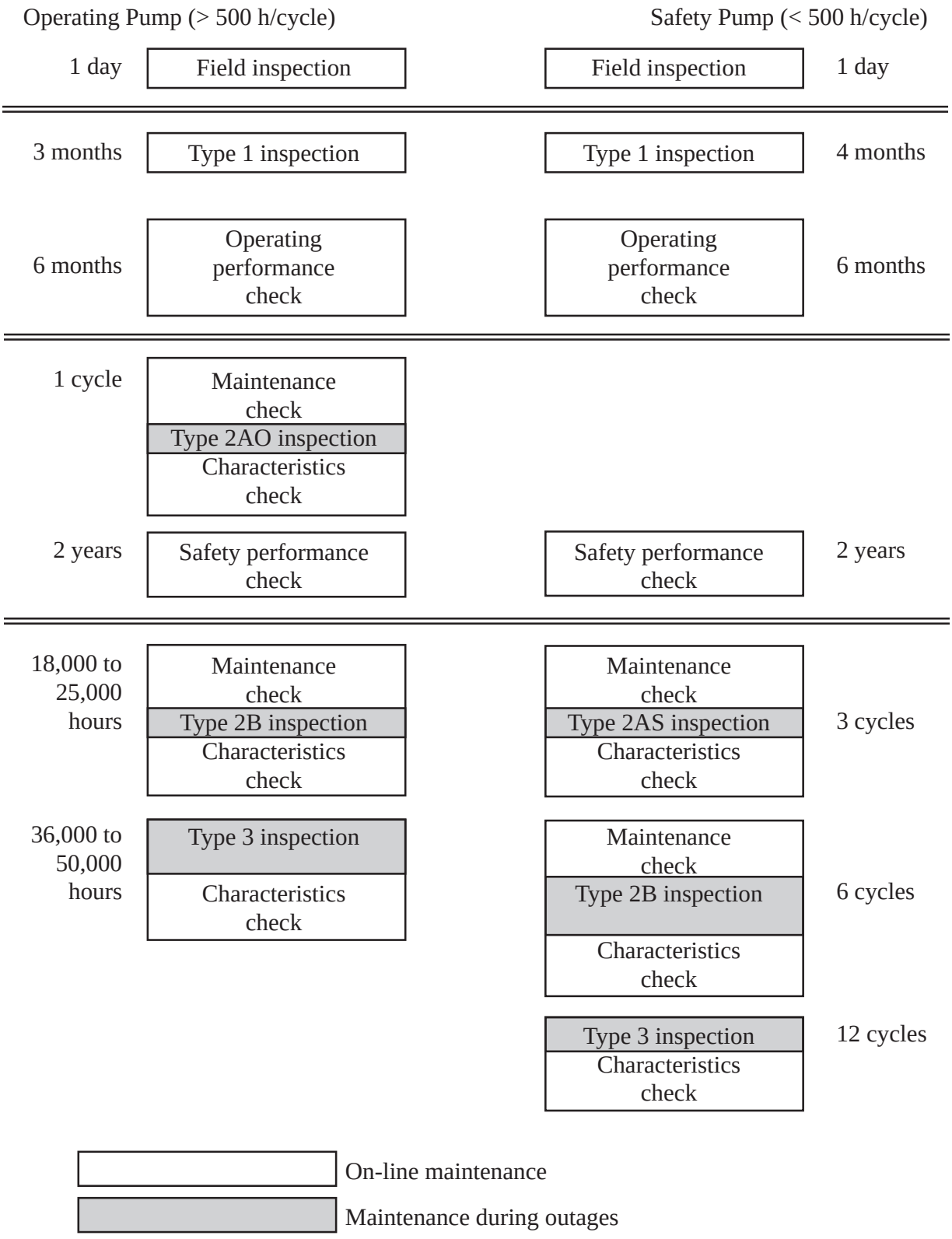


Figure 3-13
Maintenance Operations Chart

The following describes the detail of the chart activities.

- **Field inspection**. On a daily basis, all the pumps in operation are inspected. For the main pump, the inspection is focused mainly on determining that there are no oil and water leaks. For the lubrication system, the following are inspected: the oil levels, the tightness, the temperatures, the noises. For the multiplier, the inspection concerns the temperatures and the leak-tightness.
- **Type 1 inspection**. The operational parameters such as vibration amplitude (main pump and multiplier), temperature (oil system), flow and fouling (oil system), and the tightness are monitored and recorded. The results are compared to the standard values. If any variation occurs, remedial actions must be taken.
- **Operating performance check**. The pump characteristics are recorded for the standard operating conditions. The goal of this monitoring is to track any variations from the reference conditions.
- **Safety performance check**. The pump characteristics are recorded for the safety operating conditions. The goal of this monitoring is to track any variations from the reference conditions.
- **Maintenance check**. Overall pump state assessment. This global assessment is performed two to three months before a type 2 inspection. The goal of this checking is to take advantage of the planned pump outage to carry out some minor repairs of things that could lead to a breakdown or a failure in the next cycle. This inspection tries to look for minor degradations and determine if they will need remedial actions. The typical items monitored or inspected are: leaks, temperature and vibration (main pump and multiplier), noise, flow, fouling, oil level and chemical analysis (lubrication system), and the rundown time for the whole pump.
- **Characteristics check**. Qualification and recording of the pump operating parameters after a type 2 or 3 inspection. These records set the new reference operating parameters for the next period of operation. For the multiplier, these items are looked for or recorded: leaks, vibration, and temperature. For the main pump: leaks, vibration, temperature, head, and flow. For the lubrication system: flow, noise, leaks, and temperature. For the whole rotating equipment: rundown time.
- **Type 2AO inspection**. During this short inspection, the following maintenance operations are performed (not exhaustive) for the main pump: visual examination, cleaning, and axial clearance of the thrust bearing. For the multiplier: visual examination and grease replacement. For the lubrication circuit: visual examination and cleaning.
- **Type 2AS inspection**. Same as type 2AO plus the dye penetrant testing of the rear hydrostatic bearing and the measurement of the axial clearance of the hydraulic section.
- **Type 2B inspection**. Extended inspection. Same as type 2AS plus the examination or checking of: bearing, mechanical seal, disassembly of the low-speed coupling, multiplier gear train free motion, oil pump, air cooling chiller, and rear hydrostatic bearing.

- **Type 3 inspection**. General inspection. During this inspection, the typical maintenance work carried out is (this listing is not exhaustive):
  - On the **main pump**: visual examination of the overall aspect and of the surfaces; dimensional measurements; replacement of gaskets, springs, bellows, bolts, nuts, and nut locking devices; rotor balancing; and dye penetrant tests of the stellite
  - On the **multiplier**: visual examination; axial and radial clearance measurements; replacement of gaskets, nut locking devices, bearings, and grease
  - On the **lubrication system**: replacement of gaskets, filters, screws / nut locking devices, pumps, oil, and grease
  - For the **entire pump**: alignment checking

3.5.4.2 1,300 MW Units

Figure 3-14 presents the maintenance operations performed on the 1,300 MW unit pumps.

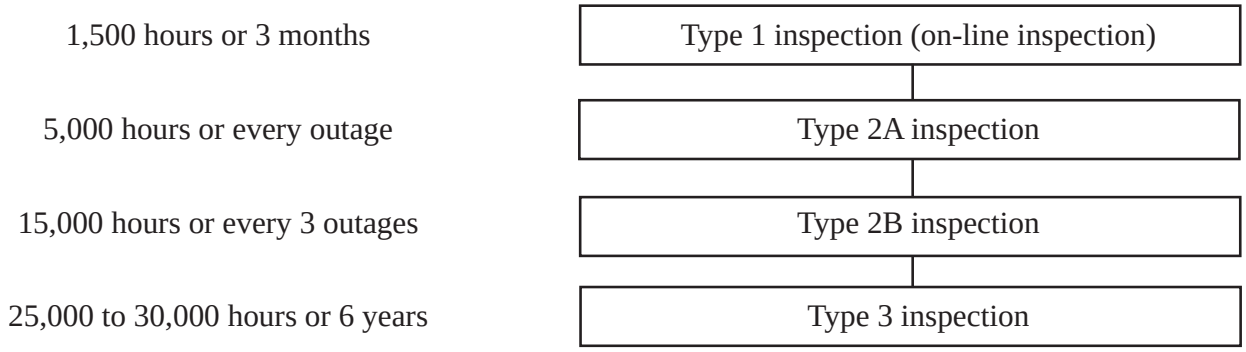


Figure 3-14
Maintenance Operations Chart

The following describes the maintenance work done for each type of inspection.

- **Type 1 inspection**. Checking the vibration amplitudes, bearing temperatures, oil level, and grease pump pressure
- **Type 2A inspection**. Same as type 1 plus: rundown time measurement, inspection (visual examination) of the high-speed coupling and the multiplier; checking the oil level, the axial clearance, and the alignment; cleaning (oil filter, cyclone); oil chemical analysis
- **Type 2B inspection**. Same as type 1 plus: rundown time measurement; oil chemical analysis; visual examination of the low to high-speed coupling, the multiplier, the direct driven oil pump coupling; axial clearance measurements and the alignments; disassembling the mechanical seals; inspecting the thrust bearing; cleaning

- **Type 3 inspection**. Same as type 1 plus: disassembly of the whole pump, followed by the rotor balancing

Moreover, after 10–12 years of operation, the grease pump, the multiplier, and the low- and high-speed couplings may be replaced, according to their condition.

4

CHARGING PUMP MAINTENANCE

4.1 Predictive—Centrifugal Pump Conditions To Be Monitored

The applicable predictive maintenance techniques identified through RCM analysis are:

- Vibration monitoring
- Temperature monitoring
- Pressure monitoring
- Flow monitoring
- Oil analysis
- Motor load monitoring

4.1.1 Vibration

Vibration monitoring of centrifugal pumps can be used to detect hydraulic forces caused by hydraulic imbalance, cavitation, recirculation, and impeller / diffuser interaction and mechanical forces caused by:

- Impeller imbalance
- Impeller instability
- Improper alignment
- Bent shaft
- Pipe stresses
- Cracks
- Wear
- Improper mounting of pump assembly components acting on the pump and its components

Figure 4-1 shows the frequency response for periodic vibration, while Figure 4-2 shows the same for a compound harmonic periodic vibration.

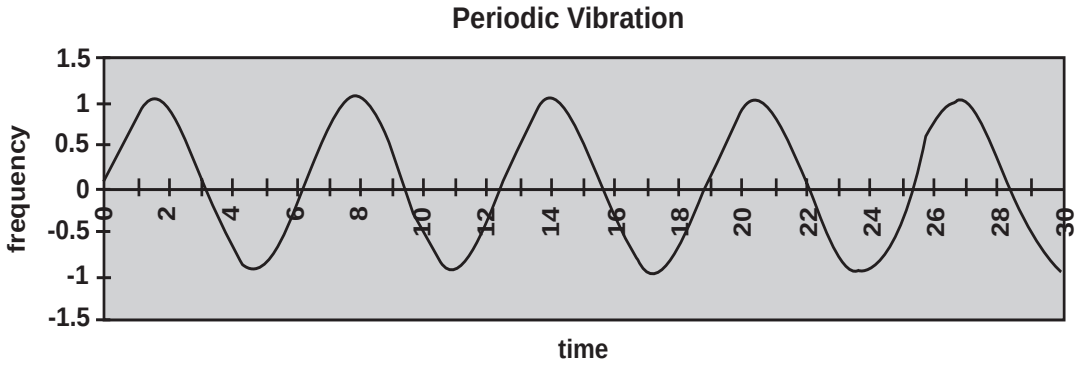


Figure 4-1
Frequency Response for Periodic Vibration

Vibration monitoring and analysis of both amplitude and frequency are widely used to assess pump condition.

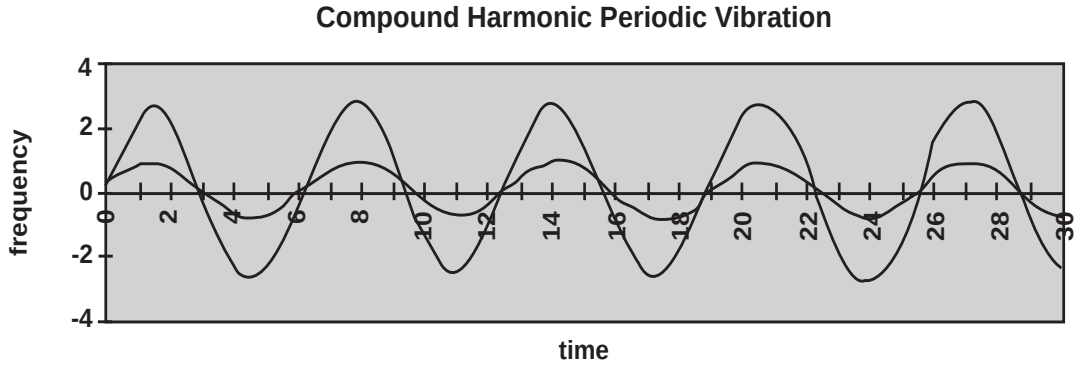


Figure 4-2
Frequency Response for Compound Harmonic Periodic Vibration

Vibration amplitude, expressed as displacement, velocity, or acceleration, is a measure of vibration magnitude or severity. Normally running pumps present a low and stable vibration amplitude. A change in amplitude indicates a change in pump condition. Frequency response of displacement, acceleration, and velocity transducers differ. If misapplied, the results can be misleading. Figure 4-3 shows the relationships between displacement, acceleration, and velocity with respect to frequency.

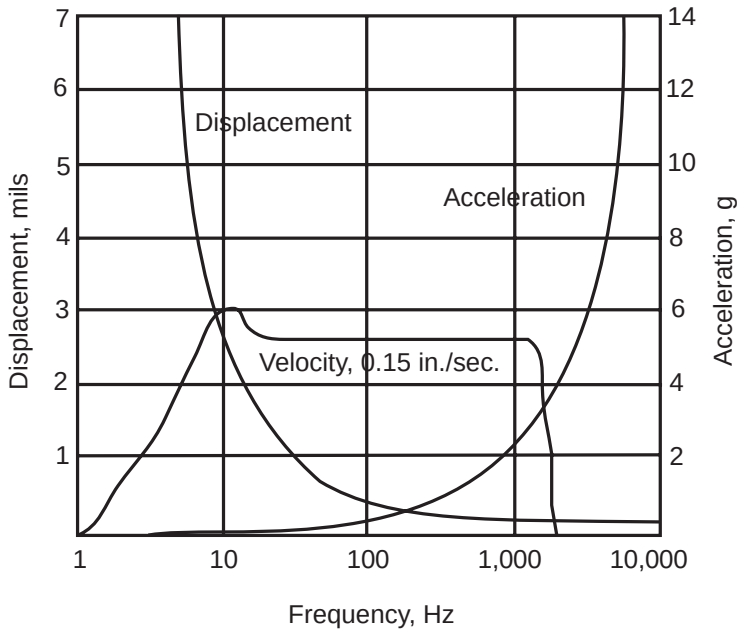


Figure 4-3
Transducer Frequency Response

Displacement is useful if measured relative to a reference. For example, proximity probes located at or near bearings provide the relative displacement of the shaft to the bearing. These measurements can be compared to design bearing / shaft clearances, and bearing wear can be determined. Monitoring systems can be set to alarm if relative displacement (clearances) exceeds a set value.

Velocity is the usual method of expressing vibration amplitude because of the relationship between velocity and vibration severity at any given frequency.

Acceleration is used to express vibration in order to put a value on the force causing the vibration:

$$F = ma$$

Vibration frequency, expressed as cycles per second, is not directly related to vibration severity. Frequency is used in determining the cause of the vibration. Frequency (*f*) is often expressed in multiples of rotating speed (rpm):

$f = 1 \times rpm$	Frequency of vibration is equal to rpm
$f = 2 \times rpm$	Frequency of vibration is twice rpm
$f = 1/2 \times rpm$	Frequency of vibration is one-half rpm
$f = N \times rpm$	Frequency of vibration equals the vane-pass frequency (where N is the number of impeller vanes)

Table 4-1 summarizes the vibration frequencies associated with the forces that can be measured, while a graphical depiction of these forces is shown in Figure 4-4.

Table 4-1
Frequency of Hydraulic and Mechanical Forces

Hydraulic Forces	Frequency
Hydraulic imbalance	f = 1 x rpm
Cavitation	f = 1 x rpm (f_1 = N x rpm)
Recirculation	f = 1 x rpm (f_1 = 1/2 to 3/4 x rpm)
Vane pass frequency	f = N x rpm
Mechanical Forces	**Frequency**
Impeller imbalance	f = 1 x rpm
Resonance	f = N x rpm
Rotational misalignment	f = 2 x rpm

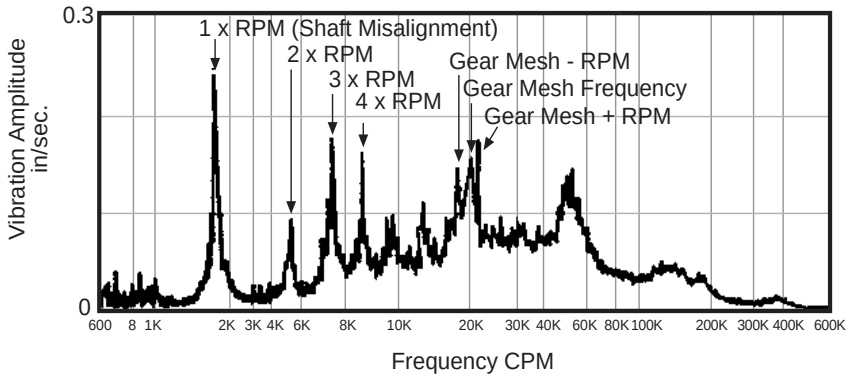


Figure 4-4
Alignment Vibration

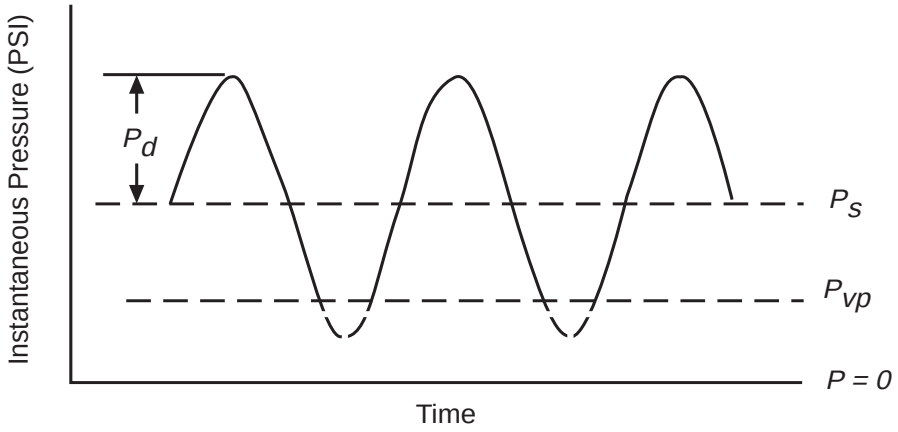
4.1.1.1 Hydraulic Imbalance

Hydraulic imbalance is induced by the fluid flow and is usually caused by poor suction piping arrangement / design. A restriction or an elbow too close to the pump suction (within about 30 pipe diameters) causes fluid to assume different velocities within the pipe. If these velocities cannot equalize prior to reaching the impeller, the hydraulic imbalance will impose a high axial vibration with a frequency equal to pump rpm (f = rpm).

Balance drum flow is typically 27–30 gallons per minute (gpm). This flow is a good indicator of wear on the balance drum, sleeve, and ring. When flow increases to approximately 40–45 gpm, this indicates wear of the internal pump components.

4.1.1.2 Cavitation

Cavitation is caused by pressure drops within the pump that allow vapor cavities to form or by the introduction of a gas or vapor into the pump suction. When the vapor cavities contact the high-pressure side of the pump, they collapse and fill with fluid, causing noise, vibration, and pitting of the impeller / diffuser. Fluid vapor cavities can occur only when the pressure drops below the vapor pressure corresponding to the fluid temperature. Multistage centrifugal pumps can experience cavitation between stages if sufficient heat of friction is added to the fluid as it is pumped. Cavitation can usually be avoided by maintaining a sufficient NPSH and by maintaining subcooled fluid at the impeller suction. Figure 4-5 shows an example of cavitation due to pulsation pressure.



If $P_d > P_s - P_{vp}$, then cavitation will occur.

P_s = Static Pressure

P_d = Dynamic Pulsations, 0 – p

P_{vp} = Vapor Pressure

Figure 4-5
Cavitation Due to Pulsation Pressure

Gas or vapor entrained or siphoned (due to poor pipe joints) into the suction fluid is compressed as it flows through the pump. Some pumps are designed for flow recirculation, from discharge to suction, in order to stabilize flow demand; and some pumps recirculate high-pressure fluid back into the pump to a balancing drum to counteract thrust. In these types of pumps, it is possible for cavitation to occur even if proper fluid

vapor pressure / temperature relationships are maintained. The expansion of the entrained gas forms "voids" followed by collapse and rapid filling of the cavities. Figure 4-6 shows the effects of entrained air or gases on centrifugal pump performance.

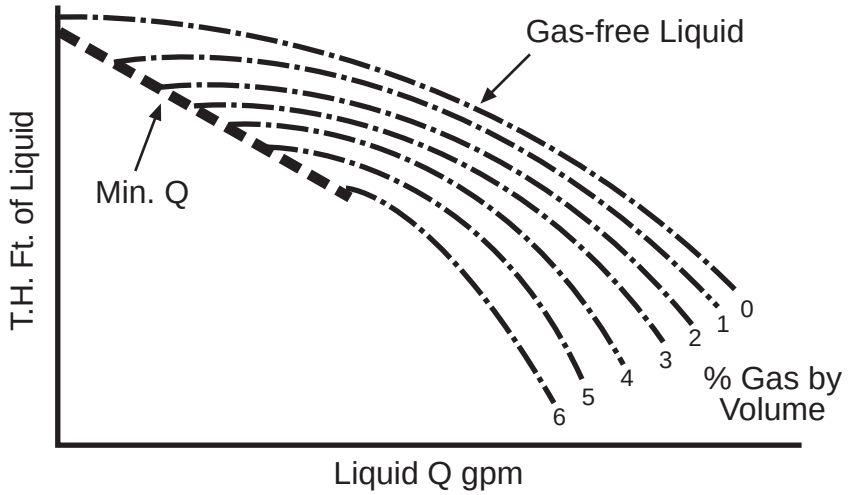


Figure 4-6
Effects of Entrained Air or Gases on Centrifugal Pumps

Cavitation produces a vibration with a frequency equal to pump rpm (f = 1 x rpm) and harmonics up to the vane-pass frequency (f_1 = N x rpm). A broadband vibration consisting of loud, steady, cracking noises is present in the pump, usually above the vane-pass frequency.

4.1.1.3 Recirculation

Recirculation is caused by high suction pressure pulling fluid back through seals and impeller clearances. It frequently occurs if a high suction pressure is accompanied by reduced flow. It can be reduced by increasing flow or installing a bypass line from the discharge to the suction.

Recirculation produces vibrations with a frequency equal to pump rpm (f = rpm) and subharmonics of speed, from 1 / 2–3 / 4 of operating frequency (f_1 = 1 / 2–3 / 4 x rpm). Audible indications of recirculation are loud, random, cracking noises in the suction piping; however, both suction and discharge pressures tend to pulse during recirculation. Location of damage on the impeller determines whether the recirculation is on the suction or discharge side of the pump, as shown in Figure 4-7. An example of the effects of cavitation on the pump head is shown in Figure 4-8.

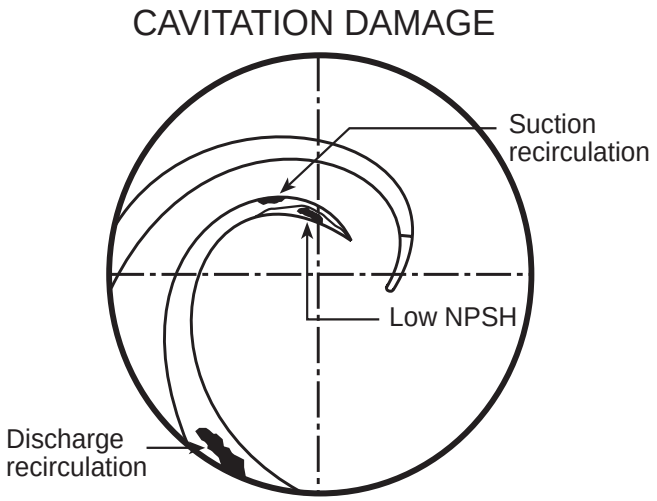


Figure 4-7
Diagnosis of Cavitation Damage

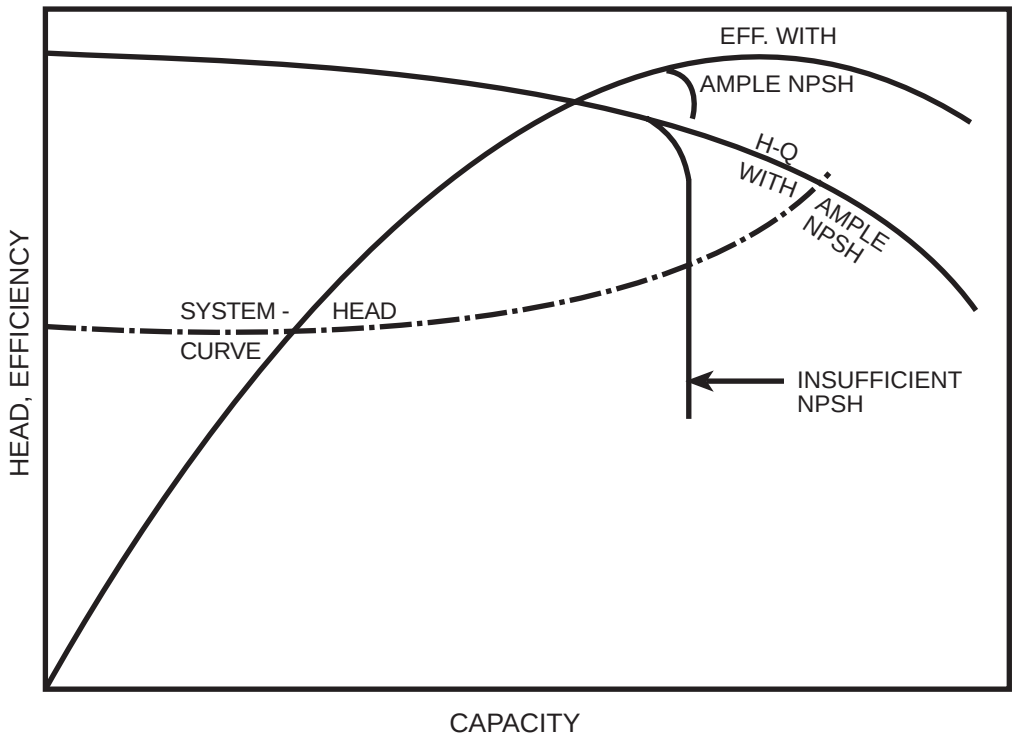


Figure 4-8
Effect of Cavitation on Pump Head

4.1.1.4 Impeller/Diffuser Interaction

Interaction between the impeller and the diffuser is caused by the pump being operated at less than rated capacity. At rated flow, the fluid meets the diffuser at an angle designed to minimize internal turbulence. At less than rated flow, the fluid interacts with the diffuser to create turbulence and axial vibrations of subsynchronous frequencies.

In a pump that has been in service for a prolonged period or with excessive clearances between the impeller and casing, random vibrations below the operating frequency of the pump occur.

Further evidence of impeller / diffuser interaction is acoustic resonance in the discharge piping caused by pressure pulsations. The piping resonates at the vane-pass frequency, regardless of the number of vanes on the diffuser.

4.1.1.5 Impeller Imbalance

Pumps always run with some degree of impeller imbalance, usually caused by errors in design, manufacture, assembly, initial balancing, or impeller damage. Imbalance can lead to excessive cavitation, bearing wear, and seal leakage and is usually detected as radial vibrations with a frequency equal to pump rpm (f = 1 x rpm). Impellers should be statically and dynamically balanced so that the maximum residual imbalance is less than 1 x W / rpm, where W = impeller weight.

4.1.2 Temperature

Viscosity is the resistance of a fluid to shear motion—its internal friction. The molecules of a liquid have an attraction for each other. They resist movement and repositioning relative to each other. This resistance to flow is expressed as the viscosity of the liquid. The viscosity of a liquid varies directly with temperature. Liquid viscosity, and hence temperature, are very important in analyzing the movement of liquids through pumps, piping, and valves.

A change in viscosity alters liquid handling characteristics in a system; more or less energy can then be required to perform the same amount of work. Increased temperatures can cause a pump to work harder, lose efficiency, and cavitate. At shutoff conditions, the brake horsepower all goes toward heating the liquid contained within the pump casing. The rate of temperature rise depends on the amount of liquid contained in the casing and the power losses (shutoff horsepower) of the pump. For a high-head, low-capacity pump, the temperature rise could be very rapid, while for a low-head, high-capacity pump, the temperature rise could be quite slow. Restrictions in the piping, partially closed valves, and other flow restrictions have similar effects on the pump.

Liquids evaporate because of molecules escaping from the liquid surface. The vapor molecules exert a partial pressure in the space, known as *vapor pressure*. If the space above the liquid is confined, after a sufficient time the number of vapor molecules

striking the liquid surface and condensing are equal to the number escaping in any time interval, and equilibrium exists. Because this phenomenon depends upon molecular activity, which is a function of temperature, the vapor pressure of a given fluid depends on temperature and increases with it. When the pressure above a liquid equals the vapor pressure of the liquid, boiling occurs.

In many situations involving the flow of liquids, it is possible that very low pressures are produced at certain locations in the system. Under such circumstances the pressures may be equal to or less than the vapor pressure. When this occurs, the liquid flashes into vapor; this is called cavitation. A rapidly expanding vapor pocket, or cavity, forms, which is usually swept away from its point of origin, and enters the regions of the flow where the pressure is greater than vapor pressure. The cavity collapses. This growth and decay of vapor bubbles affect the operating performance of the pump and can result in erosion of the metal parts in the region of cavitation.

4.1.3 Pressure

Transients in piping systems result from rapid changes in operational functioning of components, such as pumps and valves. They can produce rapid momentum changes as the column of water in the system is suddenly stopped or started. Large momentum changes can subject system components to severe force transients. Piping transients can result in the formation of vapor cavities when the system pressure is reduced to the vapor pressure of the fluid.

During a transient condition, pressure waves are generated throughout the system. In a water system, fluid-induced forces are exerted in a pipe segment by a pressure force acting normal to the inner surface of the pipe and a frictional force between the fluid and the pipe wall. The combination of these two elements is called a *segment force*, and it is collinear with the pipe axis.

Typical transients for a water-filled system include rapid valve closure, pump startup, and pump trip. The water hammer phenomena in a piping system can result in the formation of vapor pockets at locations where the pressures are reduced to or below vapor pressure. This phenomenon is normally called *column separation*. The subsequent collapse of these vapor pockets can develop significant pressure spikes. Some of the major causes of these conditions are pump startup, stopping, or seizure; pump startup with inadvertently closed discharge valves; valve opening, closing, and instability; check valve closure or delayed opening; and column separation and subsequent rejoining. Some of these events can be prevented by implementing appropriate operating procedures.

Certain types and severity levels of transients can be accommodated by the pump, but might not be accommodated by the system. NPSH transients at a pump's suction can produce a reduction in total pressure below the minimum required to satisfy a pump's inlet requirement or even below the minimum pressure required by the temperature of

the fluid. This reduction in pressure produces voids in the suction of the pump. It is the production of these vapor pockets and the subsequent collapse as the pressure is developed in the impeller that generate the cavitation which produces the noise, vibration, and pitting attack on the impeller's surfaces. The measurable consequences of cavitation on a pump's performance are a loss in flow and head resulting from the voiding of vane passages, turbulence, and two-phase flow.

Short-term transients can be considered momentary events where the available NPSH falls below the required NPSH at a particular moment. These events probably have no measurable effects on the pump mechanically. An example of a short-term transient occurs during the normal startup of a centrifugal pump where, in establishing a stable operating point, the pump would briefly run out past its design flow as the system fills. The NPSH available at run out flows can be less than the requirements of the pump, initiating a transient condition that is corrected as the system fills and flow stabilizes.

4.1.4 Flow

4.1.4.1 Discharge

Discharge flow encounters resistance such as friction, static head, and backflow. Flow rates vary depending on several factors, including variations in static head, valve alignment, system demands or loads, and flow control valve operations.

Flow (the volume of liquid that a centrifugal pump can move per unit time) is a function of the peripheral velocity of the impeller and the cross-sectional area of the impeller and its casing. The larger the passage area, the greater the flow rate. Consequently, the physical size of the pump increases with higher flow requirements for a given operating speed. The liquid flow rate is directly proportional to the area of the pump passages and can be expressed by:

$$Q = (V \times A) / 0.321$$

where: Q = flow, (expressed in gpm)

V = velocity, (expressed in fps)

A = area, (expressed in sq. in.)

4.1.4.2 Seal Leakage

Mechanical seals are considered failed when leakage past the seal causes excessive loss of fluid or excessive reduction in pressure in the sealing system. Normal leakage is sometimes misinterpreted as seal failure by personnel unfamiliar with the operation of a liquid seal. It is a common misconception that mechanical seals have a zero-leakage criteria. Actually, all mechanical seals leak, but the leakage is not visible because it evaporates. Leakage is dependent on size, pressure, and the viscosity of the liquid. This

leakage gets in between the seal faces and develops into a lubricating film. Lack of lubrication between the sealing faces causes rapid heat generation while the unit is operating, and this excess heat without cooling damages the sealing faces. Once the sealing faces are damaged, the leakage between the faces increases. Most seal failures can be attributed to face leakage.

The tapered plugs in the seals might not be part of the replacement seal. If the replacement seal is installed and a set screw is used in place of the tapered plug, the set screw provides a leakage path to the outside. Also on the replacement seal, two spacers might be required instead of the one normally supplied with the seal to prevent cocking and misalignment of the seal. When the seal is properly installed, there should be no leakage from the seal area with the pump running or shut down. If boron buildup is excessive, the backing plate might need to be cleaned with demineralized water to remove the boron crystals.

Normal operating pressure on the seal is 35–40 psi. However, once a year the seal operating pressure is increased to 150 psi for four hours. When the operating pressure is reduced to normal, the seal leaks for approximately 10–15 minutes until a steady state condition returns. The auxiliary packing should be replaced every time the seal is replaced.

NOTE: An important area regarding seal leakage is the surface finish of the hub of the gland plate where the bellows is installed. This surface should be free of scratches.

4.1.5 Oil Analysis

Oil analysis compares used lubricants to new lubricants to determine the condition of the used lubricant, the presence of contaminants in the oil (both liquid and solid), and the condition of the wearing surfaces of the machine. Although oil analysis can be used after pump failure or during final factory testing, it is the routine periodic trending, alarming, and diagnosis of oil analysis results that pay the biggest dividends.

Oil analysis has proven itself to be an ideal complement to vibration monitoring. It is suited to monitoring all rotating machinery. Because rotating machinery is often the most critical machinery in the plant, oil analysis “sees” the rotating machine from a different perspective.

Wear debris and contaminants are a separate phase in the oil and tend to settle out of the oil. When a sample is taken, it is usually warm or even hot. The presence of a significant quantity of water produces a cloudy or milky emulsion. After several minutes or hours, the water can separate out, depending on the additives in the oil. Some additives are designed to cause the oil and water to separate quickly. Water can be a constant contamination threat in such an oil. Fine metal particles of the less-than-5-microns size tend to stay suspended, while larger particles tend to settle out. If an oil is heavily contaminated, indicating severe wear, particles can settle out rapidly. They are visible at

the bottom of the sample bottle after it sits for a time. If shiny metal particles are visible, this is often an indication of an impending failure. "Normal" wear debris and even many severe wear particles are not visible to the eye. They can slip through the fingers with little or no indication of the severity of the wear on the machine.

Viscosity is the most important physical property of a lubricant and should be checked periodically. An abnormal viscosity is an excellent indicator of a problem. Some of the more common oil analysis methods are listed below, with a short description of each.

- Atomic emission spectroscopy - An oil sample is introduced into a hot arc, flame, or plasma where it is vaporized. The metal particles in the oil, at least the dissolved metals and very fine debris, absorb heat and vaporize, entering a "higher" atomic state. When the highly excited metal atom cools at the edges of the flame, the atom suddenly emits energy in the form of light in order to attain its former stable condition. Each metal releases light of wavelengths or frequencies that are characteristic of the particular metal. This "blast" of light is then carried by fiber optics into a diffraction grating that, like a light prism, orders and spreads the light into a "rainbow of colors." By locating photomultipliers where the "colors" fall on a curved wall, the presence of each metal of interest is measured and quantified in parts per million (ppm). There is a severe limitation on this type of analysis. The spectrometer sees only dissolved and very fine debris that is generally associated with good lubrication and benign wear. Coarse particles are generally characteristic of severe wear in machines.
- Automatic particle counter - There are many designs of particle counters for use in the lab and field, including on-line instruments. They all count and size particles, usually reporting them by number in each of several size categories.
- Spectrometric oil analysis (SOA) - This method provides information on the wear of the mechanical components together with information on the depletion of additives within the oil and the presence of contaminants within the oil. This type of analysis is limited to detecting particles of less than 8 microns. Industrial equipment in a healthy state frequently produces particles with an average size of 30 microns, thus limiting the application of SOA.
- Ferrographic oil analysis - This method consists of the physical analysis of wear debris particles, as opposed to the chemical analysis of SOA. The premise of this technique is that different wear mechanisms produce wear debris of differing appearances. To a skilled observer, the amount, shape, and appearance of debris in an oil sample provide an accurate indication of the health of the system from which it was obtained. This technique is criticized because the analysis is very skills-dependent and subjective.

Oil analysis is the starting point for developing a reliable approach that incorporates a tribology program, rather than simple lubricant analysis. *Tribology*, classically defined as the study of surfaces in relative motion, provides a different focus from basic oil analysis. It encompasses all influences within a system where two or more surfaces are interacting. In reality, it represents the study of a mechanical "sandwich," where there

are many influences such as the nature of the surfaces, the surface chemistry, and the load and duty cycle of the equipment being maintained, rather than simply the lubricant. From a maintenance perspective, the main concern is with the practical elements of what occurs within this tribological interaction—wear, friction, system cleanliness, and the thickness and stability of the fluid film. A typical oil analysis form is shown in Figure 4-9.

DATE		HOURS		OIL ADDED	PHYSICAL PROPERTY			SPECTROCHEMICAL ANALYSIS METALS: PARTS PER MILLION BY WEIGHT																
SAMPLED	RECEIVED	OIL	UNIT		VIS. AT 100°F	TOTAL ACID NO.	WATER	IRON	LEAD	COPPER	CHROMIUM	ALUMINUM	NICKEL	SILVER	TIN	SILICON	BORON	PHOSPHORUS	SODIUM	ZINC	CALCIUM	BARIUM	MAGNESIUM	TITANIUM
1/25	1/28	48.3	48.3	–	29.2	0.0	<.05	2.0	2.0	0.0	2.0	0.0	0.0	0.0	0.0	0.0	0.0	63.0	0.0	20.0	0.0	0.0	0.0	0.0
2/1	2/10	76.3	76.3	–	28.9	0.03	<.05	4.0	2.0	0.0	2.0	0.0	1.0	0.0	0.0	11.0	1.0	81.0	3.0	20.0	50.0	20.0	0.0	0.0

Figure 4-9
Oil Analysis Form

4.2 Centrifugal Charging Pump Maintenance Practices

Preventive maintenance actions that can be accomplished by plant personnel include:

- Pump disassembly and inspection
- Bearing disassembly and inspection

Corrective maintenance actions that can be accomplished by plant personnel include:

- Outboard seal replacement
- Inboard seal replacement
- Rotating element replacement
- Coupling alignment
- Complete pump / case replacement

4.2.1 Inspection Techniques

4.2.1.1 Operational Inspection

Assessing the condition of operating machinery is within the capabilities of qualified, experienced, and attentive operators. Inspection instruments include common hand tools:

- Flashlight
- Mirror
- Magnifying glass
- Vibration monitoring equipment
- Wiping cloths

These inspections include temperatures, pressures, leakage checks, and vibration checks.

4.2.1.2 Off-Line Inspection

Off-line inspections are normally intrusive in nature, requiring disassembly and measurement of internal wear surfaces. Inspection points and applicable inspection criteria are *italicized* in the disassembly procedure outlined in Section 4.2.2. These inspections include wearing rings, bearings, shafts, and casings.

1. Casing Inspection and Measurement Checks

    a. Before removing the casing, visually inspect the suction and discharge nozzles, auxiliary connections, sealing joints, and overall casing condition. The estimated time to perform the inspections is 20 minutes and includes the following areas:

        - Suction nozzle inspection - Note any signs of cracks, gasket surface damage, or other irregularities. Inspect the inside surface for erosion, corrosion, or wear. On pumps with nozzles 3” or larger, check the wearing ring clearance. This clearance check provides a good indication of wear and casing alignment or shaft deflection problems. The relationship between impeller and casing wearing rings is shown in Figure 4-10.
        - Discharge nozzle inspection - Note any signs of cracks, gasket surface damage, or other irregularities. Inspect the inside surface for erosion, corrosion, or wear.

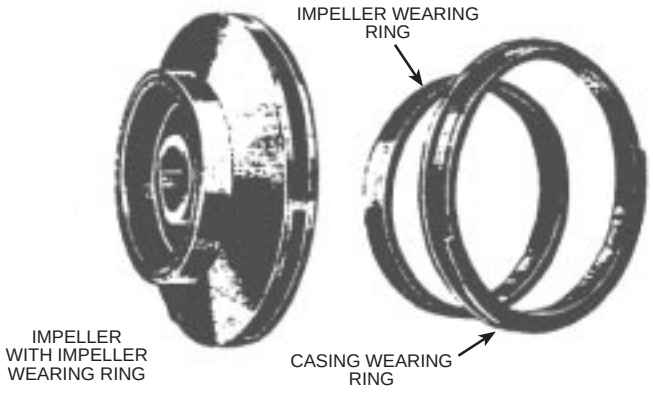


Figure 4-10
Impeller and Casing Wear Ring Relationship

b. After the casing is removed, perform additional inspections of the casing internal impeller running surface area, wear ring area, diffuser, and gasket surface. The estimated time required to perform the internal visual inspections is 20 minutes and includes the following areas:

- Wear ring area - Note any cracks, wear erosion, or corrosion. Inspect the wear ring locking mechanism for signs of cracks. The wear ring should be replaced if the clearance is greater than 150% of normal clearance. If normal diametrical clearance is not available from the manufacturer, apply the API clearance specifications of 0.010” plus 0.001” per inch of ring diameter for rings up to 12”. For rings over 12”, add 0.0005” per inch of ring diameter. Also, an added clearance of 0.005” is required for galling materials, and for temperatures over 500°F, add 0.010”.

  As running clearances increase with wear, a greater portion of the “gross” capacity of the pump is short-circuited through the clearances and has to be pumped again. The effective, or net, capacity delivered by the pump against a given head is reduced by an amount equal to the increase in leakage. Figure 4-11 shows the effect of increased leakage on the shape of the head-capacity curve of a pump.

- Internal surfaces of casing - Inspect the internal surfaces of the casing, and replace the casing if any surface damage in excess of 0.125” deep is found. Additionally, gasketed surfaces necessitate casing replacement if the surface damage exceeds 0.030”.

c. Upon completion of the visual inspection, the casing should undergo nondestructive testing (NDT) procedures to identify any cracks. Depending on the type of dye penetrant used, testing time can be as short as 30 minutes.

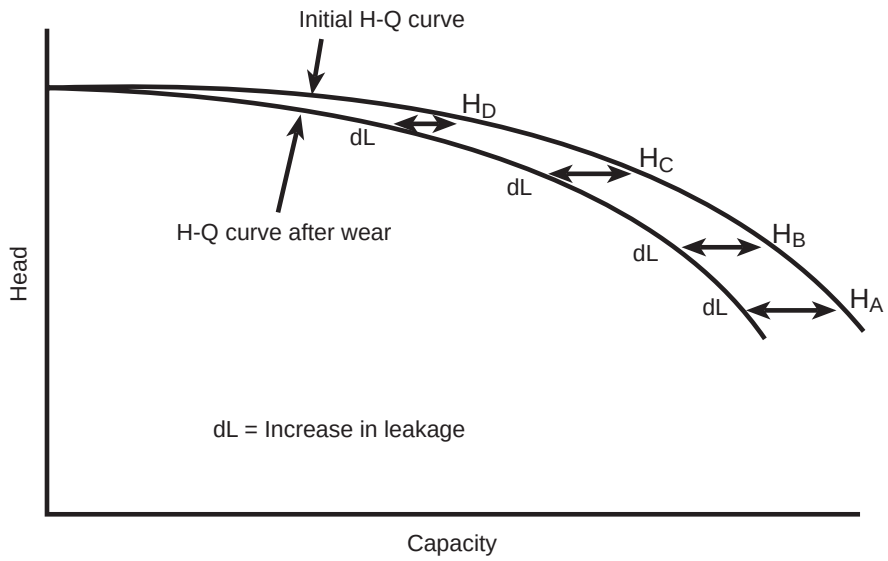


Figure 4-11
Effect of Increased Leakage on Head-Capacity Curve

4.2.2 Disassembly

Disassembly of the 11-stage Pacific pump is required for thorough inspection, and for repair or rebuild and replacement of the rotating element, bearings, and seals. Refer to Figure 4-12 for part identification during disassembly.

4.2.2.1 Disassembly for Inspection

Tools and equipment that might be needed during pump disassembly include:

- Assembly frame
- Seal removal and installation guide rods
- Assembly rail
- Feeler gauge, set
- Gloves, heatproof
- Allen wrench, set

General procedural steps for disassembly are as follows:

1. Remove the interference.
    - 1.1 Remove the supports and the restraints (hangers).
    - 1.2 Remove the inboard and outboard oil supply and return lines.
    - 1.3 Remove the water lines.
    - 1.4 Remove the coupling guard.

2. Remove the coupling.

   2.1 Remove the coupling key.

3. Rotate the shaft (no rubbing, tightness, or binding).

4. Measure the shaft axial end play (0.010“ to 0.014“).

5. Align the internal element for disassembly.

   5.1 Loosen the inboard deflector ring set screw, and move the deflector ring toward the pump casing (until clear of the bearing housing).

   5.2 Loosen the outboard deflector ring set screw, and move the deflector ring toward the pump casing (until clear of the bearing housing).

   5.3 Rotate the shaft coupling key slot to the top of the shaft.

   5.4 Rotate the inboard seal retainer plate until it is engaged in the shaft sleeve groove, and tighten the cap screws. (Note the location of the seal retainer plate in the groove on the sleeve.)

   5.5 Rotate the outboard seal retainer plate until it is engaged in the shaft sleeve groove, and tighten the cap screws. (Note the location of the seal retainer plate in the groove on the sleeve.)

6. Disassemble the inboard bearing.

   6.1 Remove the upper inboard bearing housing.

   6.2 Remove the upper stationary oil baffles.

   6.3 Roll out the lower stationary oil baffles.

   6.4 Remove the upper inboard sleeve bearing.

   6.5 Position the plastigage on top of the pump shaft, and install the inboard sleeve bearing.

   6.6 Position the plastigage on top of the inboard sleeve bearing, and install the upper inboard housing (with the gasket).

   6.7 Install the bolts in the upper inboard bearing housing and tighten them.

   6.8 Remove the bolts from the upper inboard bearing housing.

   6.9 Remove the upper inboard bearing housing.

NOTE: In addition to taking plastigage readings, it is also recommended that blue checks be performed before disassembly to determine “as found” conditions.

6.10 Measure the plastigage to determine the diametrical clearance of the inboard sleeve bearing to inboard bearing housing (0.0005–0.0015“). See Figure 4-13 for a visual representation of measuring the plastigage.

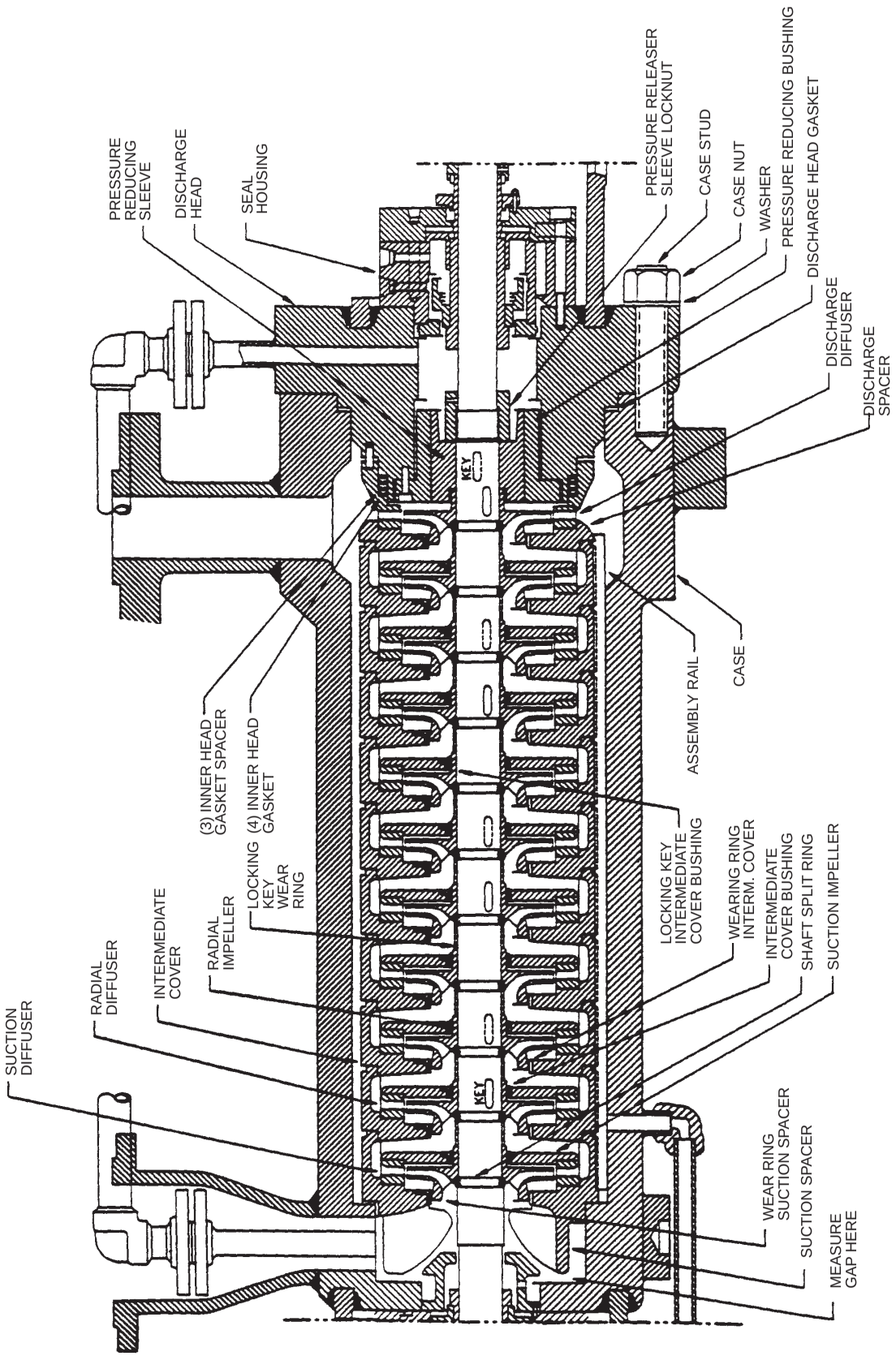


Figure 4-12
Cutaway of a Centrifugal Pump

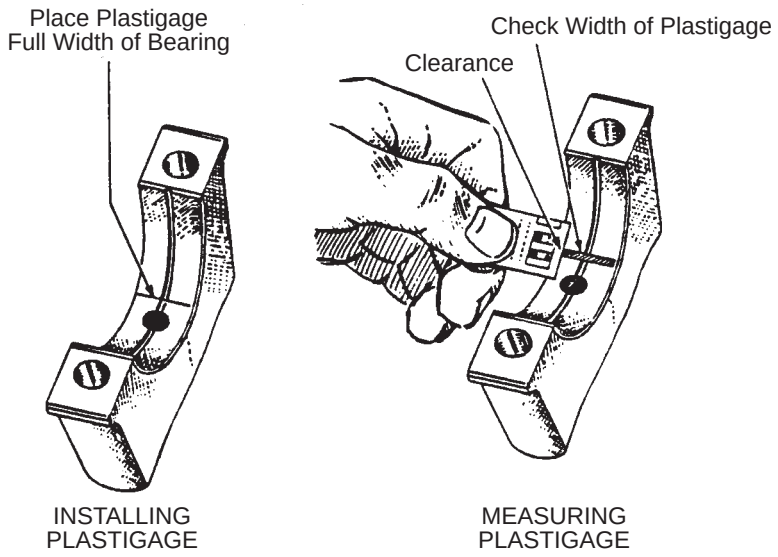


Figure 4-13
Checking Bearing Clearance Using Plastigage

6.11 Remove the upper inboard sleeve bearing.

6.12 Measure the plastigage to determine the diametrical clearance of the inboard sleeve bearing to the shaft journal (0.004–0.008“).

7. Disassemble the outboard bearing.

7.1 Remove the outboard bearing end cover.

7.2 Remove the outboard bearing housing.

7.3 Remove the upper outboard bearing housing.

7.4 Remove the upper stationary oil baffle.

7.5 Roll out the lower stationary oil baffle.

7.6 Remove the worm and the worm key from the pump shaft.

7.7 Remove the worm gear and key from the gear oil pump.

7.8 Remove the reservoir cover plate.

7.9 Remove the gear oil pump shaft housing from the outboard bearing housing, and lower the gear oil pump into the reservoir.

7.10 Remove the upper oil control ring.

NOTE: The lower half of the thrust bearing assembly consists of the base ring and three thrust shoes. The upper half also consists of a base ring and three shoes. The upper and lower half base rings are manufactured as matched pairs. The upper and lower halves, when installed, must have the same matching numbers. These numbers are inscribed on the outer surfaces of each cage. Do ***not*** *intermix parts.*

7.11 Remove the outer thrust bearing shoes. Wrap each shoe to protect the babbitt faces.

7.12 Remove the outer thrust bearing retainer and the retaining plate.

7.13 Remove the oil seal ring and the retainer.

7.14 Remove the inner thrust bearing shoes. Wrap each shoe to protect the babbitt faces.

7.15 Remove the inner thrust bearing retainer.

7.16 Roll out the lower oil control ring.

7.17 Remove the locking screw from the thrust collar nut, and remove the nut from the shaft (left-hand threads).

7.18 Remove the thrust collar and the key from the shaft.

7.19 Remove the thrust collar spacer and the inner retainer plate from the shaft.

7.20 Remove the upper outboard sleeve bearing.

7.21 Position the plastigage on top of the pump shaft, and install the upper outboard sleeve bearing.

7.22 Position the plastigage on top of the outboard sleeve bearing, and install the upper outboard housing (with the gasket).

7.23 Install the bolts in the upper outboard bearing housing and tighten them.

7.24 Remove the bolts from the upper outboard bearing housing.

7.25 Remove the upper outboard bearing housing.

NOTE: In addition to taking plastigage readings, it is also recommended that blue checks be performed before disassembly to determine "as found" conditions.

7.26 Measure the plastigage to determine the diametrical clearance of the outboard leeve bearing to the outboard bearing housing (0.0005–0.0015").

7.27 Remove the upper outboard sleeve bearing.

7.28 Measure the plastigage to determine the diametrical clearance of the outboard sleeve bearing to the shaft journal (0.004–0.008").

8. Remove the inboard and outboard lower bearing housing.

CAUTION: *If the shaft is rotated when the bottom half of either bearing is removed, damage to the wear surfaces and parts results.*

8.1 Remove the lower inboard sleeve bearing.

8.2 Remove and tag the tapered pins from the inboard bearing housing.

8.3 Remove the lower inboard bearing housing nuts and the bearing housing.

8.4 Remove the inboard deflector ring.

8.5 Remove the lower outboard sleeve bearing.

8.6 Loosen the gear oil pump flex line union.

8.7 Turn the oil pump suction line until the oil pump drops out of the lower outboard bearing housing.

8.8 Remove the gear oil pump from the reservoir.

8.9 Draw an oil sample from the reservoir, and label it with the component name, component ID number, date of the sample, and sample location.

CAUTION: *Cover the oil reservoir to prevent foreign material from entering the oil system.*

8.10 Remove and tag the tapered pins from the outboard bearing housing.

8.11 Remove the lower outboard bearing housing nuts and the bearing housing.

8.12 Remove the outboard deflector ring.

9. Remove the outboard and inboard seals. Refer to Figures 4-14a and b for seal terminology.

*NOTE: When positioning the seal retainer plate, note the location of the groove versus the tip of the retainer plate. The ideal position is when the tip fits into the groove without moving the sleeve***.**

9.1 Before the pump or bearing is disassembled, fold the seal retainer plate into the slot on the sleeve, and tighten the cap screws.

NOTE: The purpose of using the guide rods is to support the weight of the seal cartridge and to reduce the potential for galling between the shaft sleeve and the shaft.

9.2 After the bearing housing is removed, remove the two top bolts that hold the gland plate against the housing, and install the guide rods to assist in the seal cartridge removal.

9.3 Loosen the set screws in the shaft sleeve collar.

9.4 Remove the collar from the shaft.

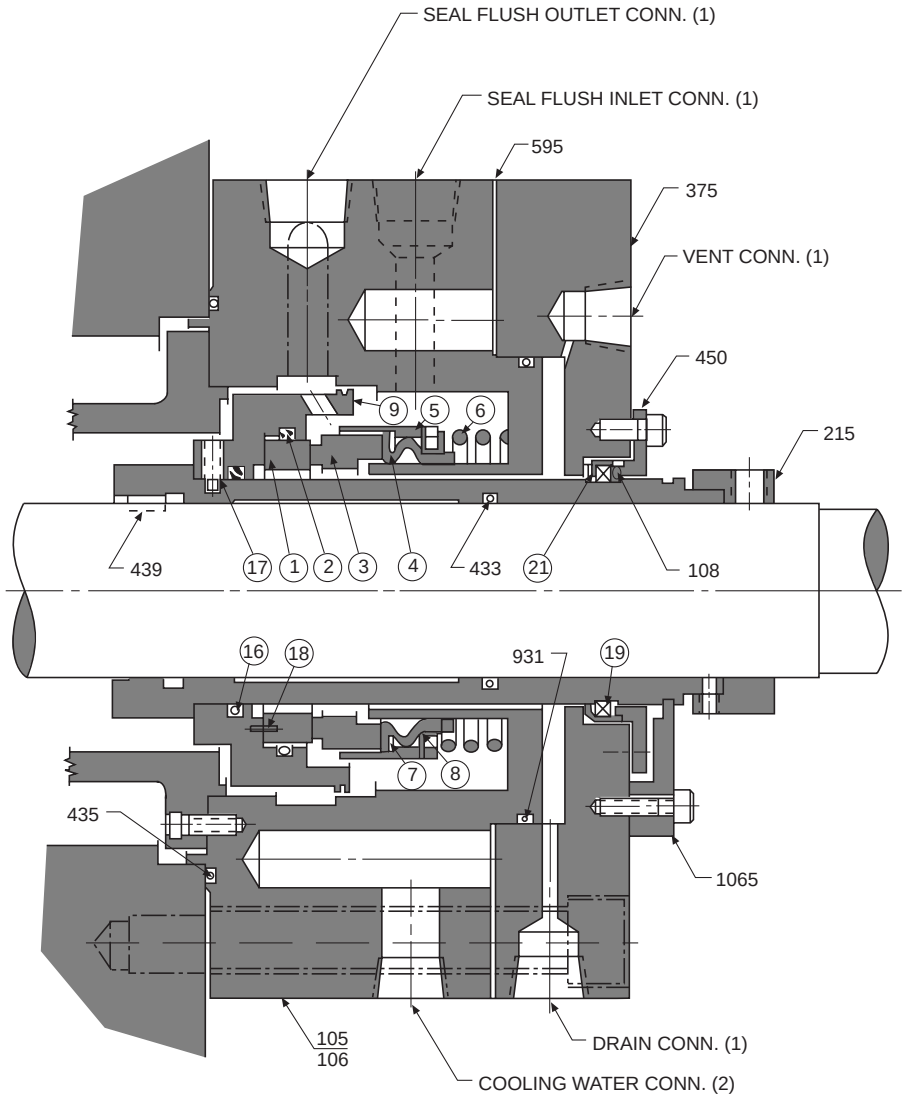
9.5 Clean all burrs from the shaft surface (especially where the set screw marks are).

9.6 Lubricate the surface of the shaft with glycerin.

9.7 Remove the two remaining bolts from the seal plate.

9.8 Remove the seal cartridge from the shaft carefully.

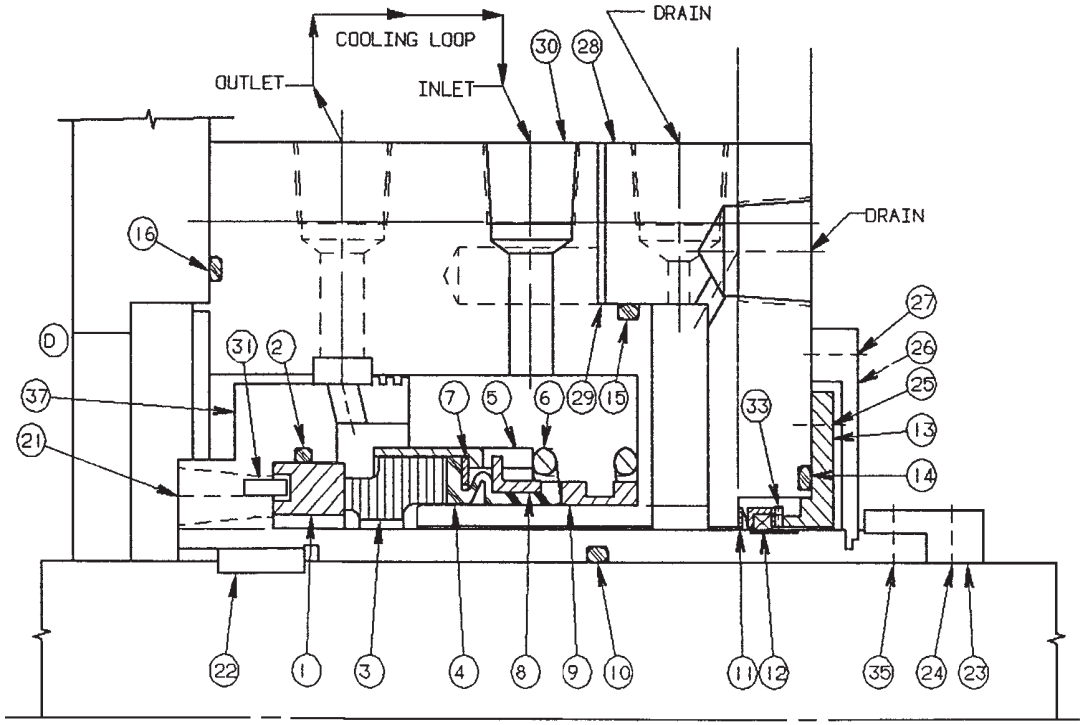
9.9 Move the seal assembly to a clean work area for disassembly.



PART NO.	PART NAME	NO. REQ'D
108	LOCK PIN	2
215	COLLAR-SHAFT SLEEVE	1
433	"O" RING-SHAFT SLEEVE	1
435	"O" RING-SEAL HOUSING	1
439	KEY-SHAFT SLEEVE	1
450	PACKING RING	1
595	GASKET-SEAL PLATE	1
931	"O" RING-SEAL PLATE	1
1065	SEAL RETAINER PLATE	1
1	SEAT	1
2	SEAT RING	1
3	WASHER	1
4	BELLOWS	1
5	RETAINER	1
6	SPRING	1
7	DISC	1
8	DRIVE BAND	1
9	PUMPING RING	1
16	"O" RING	1
17	DOG POINT SET SCREW	4
18	DRIVE PIN	1
19	AUXILIARY PACKING ASS'Y	1
21	SPRING	1

NOTE: QUANTITIES SHOWN ARE FOR EACH SEAL ASSEMBLY.

Figure 4-14a
Typical Mechanical Seal Installation
(John Crane First Generation Seal—Obsolete Design)



ITEM	PART NAME	NO. REQ'D	ITEM	PART NAME	NO. REQ'D	ITEM	PART NAME	NO. REQ'D
1	MATING RING	1	13	AUX. GLAND	1	25	CAP SCREW	
2	O-RING	1	14	O-RING	1	26	SPACER	
3	PRIMARY RING	1	15	O-RING	1	27	CAP SCREW	
4	BELLOWS	1	16	O-RING	1	28	GLAND PLATE	
5	RETAINER	1	17	PUMPING RG. ASS'Y.	1	29	GASKET	
6	SPRING	1	18	SET SCREW (dog point)	4	30	GLAND PLATE	
7	DISC	1	19	O-RING	1	31	PIN (.125 D.)	2
8	DRIVE BAND	1	20	O-RING	1	32	AUX. GLAND	
9	SPACER	1	21	PLUG	3	33	PIN (.062 D.)	
10	O-RING	1	22	DRIVE KEY		34	GASKET	
11	SPRING	1	23	COLLAR		35	SET SCREW	
12	PACKING ASS'Y.	1	24	SET SCREW		36	SLEEVE W/0 P.R.	
						37	PUMP. RING/SLV.	

Figure 4-14b
John Crane Second Generation Mechanical Seal

NOTE: There is a deflector plate installed against the front face of the inboard seal housing. This plate must be separated from the housing before the seal cartridge can be removed. Otherwise, the seal removal is identical on both the inboard and outboard sides of the pump.

9.10 After separating the deflector plate from the front face of the inboard seal housing, remove the inboard seal by repeating steps 9.1 through 9.9 above.

NOTE: The following steps involve disassembly of the seal. These steps will probably be performed at a later time after disassembly of the pump has been completed.

9.11 Loosen the cap screws, and rotate the retainer plate out of the sleeve.

9.12 Remove the cap screws and the retainer plate from the back of the housing.

9.13 Slide the sleeve out of the housing, and check the face of the mating ring for any unusual wear.

NOTE: Tapping out the mating ring should be done with the mating ring removal tool by inserting the three pins into the holes on the back side of the pumping ring portion of the sleeve (where the plugs were removed) and tapping lightly with a mallet. If this tool is not available, a punch that fits the holes can be used.

9.14 Remove the plugs from the pumping ring, and gently tap out the mating ring.

9.15 Remove the primary ring from the retainer, and check the face for any irregularities.

NOTE: Use a tool similar to a bearing puller to remove the seal head assembly from the hub of the gland plate. For Steps 9.16.1 through 9.16.8, use the existing tool designed by Pacific Pump.

9.16 Remove the rest of the seal head from the housing, using a hook-type tool similar to a bearing puller. Ensure that the surface of the hub is not damaged during removal.

9.16.1 Remove the primary ring from the retainer.

9.16.2 Remove the retainer with a pair of pliers.

9.16.3 Place the seal puller plug against the front of the hub.

9.16.4 Insert the hook pullers through the slots of the strongback, and hook them into the coils of the spring.

9.16.5 Slide the spacers over the threaded end of the hook puller, and tighten the nuts.

9.16.6 Screw the threaded rod into the strongback until it touches the plug.

9.16.7 While rotating the rod against the plug, the spring and the bellows will slide out from the cavity.

9.16.8 Slide out the spacer from the bottom of the cavity. Make sure the spacer slides out easily and does not damage the surface of the hub.

9.17 Remove the cap screws from the auxiliary gland, and slide out the auxiliary gland, packing retainer, and the wave spring. Make note of any unusual wear or discoloration.

9.18 Remove the cap screws, and separate the gland plate from the seal housing. Note any discoloration or irregularities.

9.19 The seal disassembly is now complete.

10. Remove the discharge head.

10.1 Install an eyebolt in the pump head flange, and couple it to an overhead lifting device.

NOTE: Before loosening the casing nuts, check the gap between the head and the casing. This can be done by using a feeler gauge at points between the studs. The clearance should be the same at all points. The specification for this gap is 0.040–0.045", with "out-of-parallelism" at any point between head and case faces no more than 0.002". In some installations, a 0.035" gap has been recommended by the manufacturer to achieve good sealing.

This check is repeated on reassembly; however, doing it at disassembly allows the technician to find any anomalies early and repair them while the pump is disassembled, not holding up the reassembly process.

10.2 Remove the case nuts.

NOTE: When a pump has been in service for a long time, it can be difficult to loosen the head with the jackscrews. If this is the case, the head can be loosened by mechanical agitation with a mallet while tightening the jackscrews. As a last resort, a steel wedge can be driven in the clearance between the head flange and the end of the case. The wedge should be driven at three or more points equally spaced around the circumference of the head. If using the wedge, be extremely careful to avoid damaging the flange faces.

10.3 Using jackscrews, loosen the head. Jackscrews must be taken up evenly to prevent cocking or jamming the head. Back out the head far enough so that it can be supported by the overhead lifting device.

CAUTION: *The discharge diffuser is located on the discharge head with two dowel pins. Be careful during head removal that the discharge diffuser remains attached to the head and does not fall, causing damage to itself, other parts, or personnel.*

10.4 Tension the lifting device so that the head is fully supported without binding against the casing. Work with the jackscrews until the head is completely free of the casing, being careful not to damage the shaft, pressure-reducing sleeve, or pressure-reducing bushing.

10.5 Place the head out of the immediate working area for later disassembly and inspection.

11. Remove the internal element.

NOTE: The internal element includes all stationary and rotating parts in the pump case, except those parts removed with the head assembly. The internal element must be removed and also installed as a complete assembly.

11.1 Position, align, level, and tightly bolt the assembly skid to the pump case.

CAUTION: *The track of the assembly can have sharp edges or burrs. Use a cloth or glove when applying lubricant to the assembly skid tracks.*

11.2 Lubricate the tracks to ease the sliding of the internal element over the assembly skid.

CAUTION: *Apply pulling force to the shaft* ***only*** *in line with the shaft axis or the stationary components. Any force exerted on the shaft perpendicular to the axis can damage the shaft.*

11.3 Slide the internal element out of the case and onto the assembly skid.

11.4 Remove the assembly to the most convenient location for disassembly.

4.2.2.2 Disassembly for Rotating Element Replacement

Tools and equipment that might be needed during disassembly of the rotating element include:

- Hoist, 2-ton capacity
- Lifting straps
- Blocks, wooden
- Balance drum wrench, spanner
- Roll and heel bars, matched
- Pry bars, matched
- Pyrometer

- Acetylene torch, rosebud tip (#10)
- Gloves, heatproof
- Hose, water
- Pipe clamps, matched
- Allen wrench, set

The sequence of rotating element disassembly is based on the shaft design, which is stepped up at each impeller, from the discharge end to the suction end. Therefore, all components (except the suction spacer) are removed from the discharge end of the shaft.

1. Remove the final stage impeller.

NOTE: The pressure-reducing sleeve locknut has a left-hand thread and must be turned clockwise to remove it from the shaft.

1.1 Remove the pressure-reducing sleeve locknut.

1.2 Remove the pressure-reducing sleeve.

1.3 Remove the assembly rail.

1.4 Remove the suction spacer with the wearing ring.

1.5 Remove the locking keys, if installed.

CAUTION: *Before attempting to remove an impeller from the shaft, the impeller must be heated, beginning at the outer diameter and slowly moving the heat source in a spiral motion toward the hub.*

NOTE: The use of two torch tips to heat the impeller has been recommended by the manufacturer.

1.6 Remove the last stage impeller.

1.7 Remove the impeller key.

1.8 Remove the split ring.

1.9 Remove the discharge spacer with wearing ring.

1.10 Remove the intermediate cover bushing.

1.11 Remove the locking keys, if installed.

1.12 Remove the diffuser.

2. Remove the remaining stage impellers.

2.1 Remove the radial impeller.

2.2 Remove the impeller key.

2.3 Remove the split ring.

2.4 Remove the intermediate cover with the wearing ring.

2.5 Remove the locking keys.

2.6 Remove the diffuser.

2.7 Repeat steps 2.1 through 2.6 until all intermediate covers, impellers, and split rings are removed from the shaft.

4.2.2.3 High-Speed Gear Drive Disassembly

1. Unscrew the pipe unions in the inlet and outlet lines to the lube oil pump and in the oil line to the gear oil spray.
2. Remove the bolts holding the pump support end cap to the gear case, and remove the cap and the pump.
3. Remove the bolts holding the high-speed end caps to the unit, and remove the closed cap.

NOTE: Some newer design high- and low-speed shaft end caps are of the split design so that they can be removed from the shaft without removing the coupling if so desired.

4. Remove the bolts holding the low-speed end cap to the gear case.
5. Remove the bolts holding the upper case to the lower case.
6. Using the jacking screws, free the upper case, and remove it from the lower case.
7. Remove the upper half high- and low-speed bearings.
8. Lift out the high-speed pinion shaft, and remove the lower high-speed bearing.

NOTE: To remove either the low-speed shaft or the lower low-speed bearings, first lift the shaft extension—just high enough to pull up the bearing shell so that it clears the dowel pin—and rotate it 180° to the top of the shaft to take it from the case.

9. Remove the low-speed bearings and the low-speed shaft.

4.2.3 Repair

4.2.3.1 Casing Repair

NOTE: Excessive pitting and erosion of the pump casing can be corrected by weld buildup or electroplating buildup of the affected areas. If the buildup process is done incorrectly, however, the casing could be made unusable because of distortion and cracking.

1. Visually inspect the O-ring grooves and sealing surfaces for corrosion, pitting, and scratches.
2. Determine if the surfaces are satisfactory.
3. Using a profilometer, determine the surface finish. The maximum surface finish for grooves is typically 125 rhr; while for mating sealing surfaces, it is 63 rhr.
4. Perform weld buildup or electroplating, if necessary.
5. Hand polish the O-ring grooves and sealing surface to remove any sharp edges from acceptable imperfections.

4.2.3.2 Repairing a Damaged Shaft Journal

NOTE: Journal bearings with partial wiping or localized surface damage can often be repaired by scraping down the high spots on the babbitt surface.

1. Clean the bearing shells thoroughly, and remove all burrs.
2. Clean the bearing seats, removing rust, burrs, and dirt.
3. Install the clean, dry bearing. Ensure that all other bearings are in place.
4. Clean the shaft, and apply a light film of Prussian blue to the journal.
5. Place the shaft in its bearings, and rotate it one turn.
6. Lift the shaft, and remove the bearing shell. Observe the Prussian blue that remains on the high spot. Also note evidence of the shaft pressing unevenly to one side or one end of the bearing.

NOTE: In scraping soft babbitt, it is best to select a straight scraper that has a rectangular cross section and one that is long enough to be gripped with both hands. Avoid scrapers with knife-like edges and curved edges, such as the usual spoon-shaped scrapers. A good tool can be made by grinding all four sides of an "old" file parallel on a surface grinder. A piece of ground tool stock also makes a good scraper for babbitt, because all corners are 90°.

7. Carefully scrape down the points of contact. Recheck the scraped areas frequently.
8. Perform a blue check periodically until the bearing is satisfactory.

4.2.4 Reassembly

The reassembly procedures are for the Pacific 11-stage centrifugal pump.

4.2.4.1 Reassembly of the Rotating Element

1. Inspect all gasket surfaces, and thoroughly clean all mating surfaces.
2. Install the pressure-reducing bushing.

NOTE: Before it is installed, the pressure-reducing bushing must be subcooled in dry ice.

2.1 Cool the pressure-reducing bushing.

2.2 Measure the OD and the ID of the head to verify the proper clearance.

2.3 Push the pressure-reducing bushing into position until it bottoms out on the shoulder of the head.

2.4 Line up the bolt holes, and insert the cap screws. Do **not** tighten the cap screws fully at this time.

NOTE: Ensure that the pressure-reducing bushing has warmed to ambient temperature before fully tightening any cap screws.

2.5 Tighten all cap screws.

2.6 With the wrench in position, stake the cap screw metal into the slot.

3. Install the new inner head gaskets with spacers.
4. Replace the discharge diffuser, using the locating dowels to ensure proper positioning.

NOTE: Make up the rotating assembly, starting from the inboard or coupling end.

5. Assemble the rotating element.

5.1 Inspect the shaft surface and remove any burrs. Inspection should include a PT for cracks, runout (0.001” TIR), measurement of all diameters and fits for impellers and bearings, and all impeller bore IDs.

5.2 Inspect the bores and the end faces of all impellers, and remove any burrs.

5.3 Install the split rings and the key for the first stage impeller.

NOTE: Before installing the impeller, it is necessary to apply heat to the impeller hubs. The hub should be evenly heated to approximately 250°F and the impeller slid onto the shaft while the hub is at this temperature. It is important that no burrs remain on the shaft, impeller bore, and end faces of the impeller hubs. For convenience in assembling, most impellers have the stage number stamped on the reverse above the keyway.

5.4 Install the impeller.

5.5 Repeat steps 5.3 and 5.4 until all impellers are installed on the shaft.

NOTE: The clearance between the pressure reducing sleeve and the bushing is critical. The new clearance is 0.010–0.012". The maximum clearance allowed before replacement is 0.020". However, any clearance in excess of 0.012" results in reduced performance and efficiency.

5.6 Install the pressure-reducing sleeve.

CAUTION: *The pressure-reducing sleeve locknut has left-hand threads. Turn it counter-clockwise to tighten.*

5.7 Lubricate the shaft threads with Neolube, and install the locknuts.

6. Check the rotating assembly for concentricity.

6.1 Set the lower half of the sleeve bearings into "V" blocks. See Figure 4-15 for the V-block configuration.

NOTE: With the unit disassembled, the shaft TIR should be checked.

6.2 Clean the bearing halves, and apply a thin coat of oil to the lower half of the sleeve bearings.

6.3 Set the assembled rotor into the bearings.

6.4 Steady the dial indicator against the wearing surface.

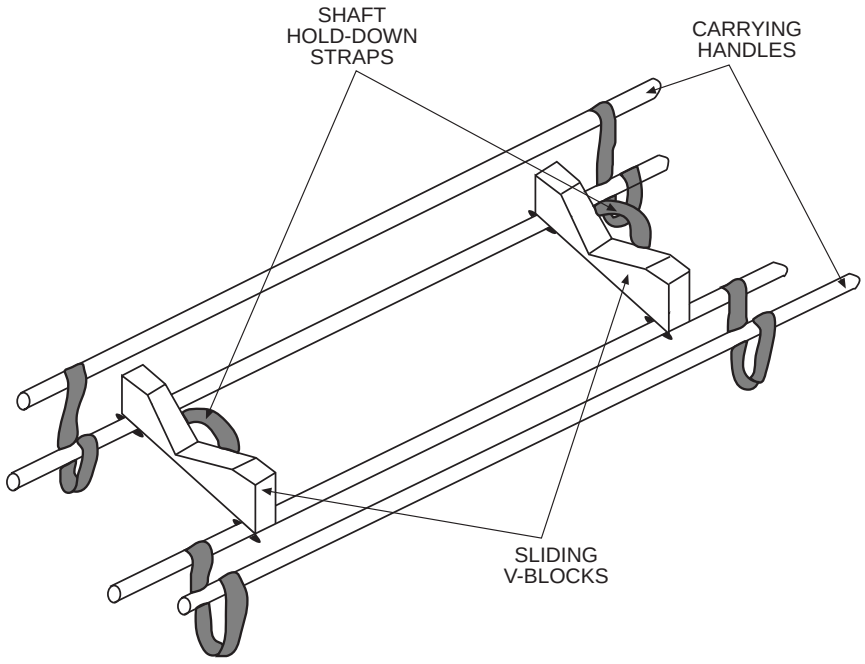


Figure 4-15
V-Block Configuration

NOTE: The following measurement should be repeated at several locations on the exposed shaft, near the center of the pressure reducing sleeve, and at the circumference of each impeller hub and skirt. If the indicator readings exceed the allowable runout, the cause must be found and corrected before proceeding. Possible causes are bent shaft, butting faces of the impeller hub, misaligned pressure-reducing sleeve, split rings, locknut not square with the shaft centerline, or burr or dirt particles on faces.

6.5 *Slowly turn the rotor, and note any variations in the dial indicator readings. The total indicator reading should not exceed 0.0015" on any surface.*

6.6 When the rotating assembly is verified true, remove the locknut, pressure-reducing sleeve, impellers, and split rings, starting from the outboard end of the shaft.

7. Assemble the unitized internal element.

NOTE: The internal parts should be assembled on the assembly frame, which allows the assembled unit to remain level and slide freely into the casing bore. To facilitate assembly, an assembly rail is fitted in a machined slot in the suction and discharge spacers and in each intermediate cover. During assembly, keep the slots off the vertical centerline so that they just clear the top of the assembly frame.

NOTE: Verify that a 0.040" x 20° chamfer exists at the step of the shaft. If no chamfer exists, grind an approximately 0.040" x 20° chamfer. This is required later when assembling/installing the seals, but it is much easier to do when the impellers are removed from the shaft.

7.1 Install the split ring and the key for the first stage impeller.

NOTE: To ease impeller installation, heat the hub to approximately 250°F before sliding the impeller on the shaft.

7.2 Install the first stage impeller.

7.3 Install the second stage intermediate cover with the wearing ring and bushing installed in it.

7.4 Install the second stage split ring.

7.5 Repeat steps 7.1 through 7.3 until the last impeller, all intermediate covers, and the discharge spacer are assembled on the shaft.

7.6 Install the split ring for the last stage impeller.

7.7 Install the pressure-reducing sleeve.

CAUTION: *The pressure-reducing sleeve locknut has left-hand threads. Turn it counterclockwise to tighten.*

7.8 Install the pressure-reducing sleeve locknut, and torque to the proper torque.

7.9 Install the suction spacer with its wearing ring.

7.10 Install the assembly rail in the slot provided.

8. Install the internal element.

NOTE: Before installing the internal element, ensure that the inside surfaces of the casing and the outside surfaces of the internal element are clean. Pay particular attention to the sealing shoulder at the inlet end of the case and the abutting face of the suction spacer. These surfaces must be perfectly clean to prevent internal leakage from the discharge to the suction side of the pump.

8.1 Slide the unitized internal assembly into the pump case.

8.2 Remove the assembly frame.

8.3 Check the position of the assembly rail. It should be between the assembly rail guides at the bottom center of the casing.

9. Install the pump discharge head.

NOTE: The gasketed joints on horizontal pumps can be difficult to seal. The casing gasket should extrude slightly where the bearing housing meets the casing (1/64–1/32"). The joint should also be treated with an RTV suitable for the fluid and temperatures at the time of assembly.

9.1 Install the new head gasket.

9.2 Install an eyebolt at the top of the discharge head, and lift the head into position so it is centered with the casing.

NOTE: After completing the tightening sequence on the casing nuts, the gap between the head and casing must be 0.050" ± 0.001". The feeler gauge should be used at points between studs, but not at any point where there are surface defects on the casing or the head. The clearance should be the same at all points. The correct gap is achieved by tightening each casing nut and moving around the head so that the studs are tightened in progression. This draws the pump into the head slowly and evenly. Progress should be measured with the feeler gauge. An uneven gap causes the bearing housing assembly to be tilted out of position.

9.3 Install the casing stud washers and nuts. Tighten them just enough to bring the discharge head and gasket up to the casing face.

9.4 Tighten all studs and nuts to achieve a discharge head to casing clearance of 0.040–0.045", with "out-of-parallelism" at any point between the head and case faces no more than 0.002". *In some installations, a 0.035" gap has been recommended by the manufacturer to achieve good sealing.*

10. Center the rotating element.

NOTE: The bearing housings must be properly located so that the rotating element is concentric with the stationary parts.

10.1 Temporarily install the lower halves of radial and thrust-bearing housings with only the lower half of the sleeve in each housing.

10.2 Install the bearing housing nuts finger tight.

10.3 Install the bearing housing alignment dowel.

10.4 Lightly oil the sleeves and the portions of the shaft that ride in the sleeves.

10.5 Using feeler gauges, reach in through the opening in the head on the discharge end of the pump, and check the clearance between the pressure-reducing sleeve and the pressure-reducing bushing. The variation in the clearance around the circumference should be no more than 0.001".

10.6 Check the clearance between the suction spacer wearing ring and the first stage impeller wearing skirt. The variation in clearance around the circumference should be no more than 0.001".

10.7 If the clearances are satisfactory, proceed to step 10.9. If the element is not centered properly, as evidenced by any unsatisfactory measurements in steps 10.5 or 10.6, remove the bearing housing dowels, and leave the bearing housing nuts finger tight.

NOTE: If the element is not properly centered, a correction can be made by using jacking screws or by placing a small hydraulic jack under the bearing housing. Measure the clearance between rotating and stationary parts. Move the bearing housing around until the clearance is within 0.001". When this is accomplished, tighten the bearing housing nuts. Repeat the procedure for the other bearing housing, and then recheck the concentricity at the first end. Both ends must be within the limits specified.

10.8 Using a small hydraulic jack, move the bearing housings around until the clearance between the rotating and stationary parts is within 0.001".

10.9 Clamp the dial indicator to the pump shaft.

10.10 Place the dial indicator tip against the casing bore and discharge head where the seal housings are installed.

10.11 Rotate the pump shaft, and note the variation in dial indicator readings.

10.12 After the shaft is centered, tighten all bearing housing stud nuts.

10.13 Recheck to ensure that the tightening of nuts did not disturb the shaft centering.

NOTE: When a slow initial rolling motion is imparted to the shaft, it can be maintained with a very slight hand-applied force if the rotating element is properly centered and is not binding against any of the stationary parts.

10.14 Rotate the shaft by hand to ensure that it rotates freely.

10.15 Ream out the bearing housing dowel holes.

NOTE: The installation of new dowels requires rechecking the shaft to ensure that it is still centered with the new dowels installed.

10.16 Install the dowels.

10.17 Recheck the shaft centering, if applicable.

11. Assemble the radial bearing.

NOTE: Before assembly, it is necessary to thoroughly clean all parts. It is assumed at this point that an inspection of all parts has been made and worn pieces replaced.

11.1 Slide the deflector ring onto the shaft and against the seal housing. Do **not** tighten the set screw.

NOTE: The gasket at the parting line of the upper- and lower-half bearing housing should be 0.014–0.016" thick and should be made from composition paper or sheet asbestos gasketing with a rubber binder. Gaskets must be cut to match the bearing housing at assembly.

11.2 Position the lower half of the bearing housing onto the mounting studs of the casing bearing bracket.

11.3 Install the nuts and tighten them finger tight.

11.4 Ensure that the rotating element is centered in accordance with step 10 above. Align the bearing so that the tapered dowel holes of the bearing housing and casing align.

11.5 Tap the taper pin dowel into position on each side of the bearing housing flange to align the housing.

11.6 Tighten the stud nuts.

NOTE: When the oil baffles and the lower-half sleeve bearing are rolled into the housing, a slight lift on the end of the shaft might be necessary to allow the parts to roll into position.

11.7 Oil the mating surfaces, and roll the oil baffles and lower-half sleeve bearing into the lower-half bearing housing.

11.8 Thoroughly oil the upper half of the sleeve bearing, and set it into position on the lower bearing half.

11.9 Oil the top half of the oil baffles, and push them into the upper half of the bearing housing.

NOTE: If new bearing sleeves are being installed, it is necessary to drill the upper half for the anti-rotation dowel. The hole is drilled using the upper-half bearing housing as a guide. Additionally, a hole might need to be drilled for the temperature probe. After drilling the hole(s), remove the upper half and thoroughly remove all debris. Install the dowel pin in the upper-half bearing sleeve, and reinstall the upper-half bearing housing with the upper-half oil baffles.

NOTE: When installing new bearings, assemble the bearings into the housing, and take three micrometer readings 120° apart on the bearings and the shaft journals to ensure that acceptable clearances are achievable. This should be done with the housings removed from the pump. If the clearances are not satisfactory, the new bearing needs to be scraped. Additionally, the dowel pin in the new bearings must be measured to ensure that it is not too long. When new holes are drilled for the dowel pin in the bearing housing, the use of a drill guide is recommended. Although a drill guide is not required, it allows more accurate drilling.

11.10 Install the upper half bearing housing with the oil baffles onto the lower-half bearing housing.

11.11 Install the cap screws and tighten them.

12. Assemble / install the thrust bearing.

12.1 Slide the deflector ring onto the shaft and against the seal housing. Do **not** tighten the set screw.

12.2 Position the lower-half bearing housing on the studs for the pump head bearing bracket.

12.3 Install the stud nuts finger tight.

12.4 Align the tapered dowel holes in the mounting flange with the holes in the pump head bearing bracket.

12.5 Tap the tapered dowels into each side of the mounting flange to align the housing.

12.6 Tighten the stud nuts with a wrench.

NOTE: The gasket material should be 1/64" thick and should be high-quality composition paper or sheet asbestos with a rubber binder. Bearing housing gaskets must be cut to fit the bearing housing at assembly.

12.7 Install the gasket at the parting line of the upper and lower halves of the thrust bearing housing.

12.8 Lubricate and roll into position the lower-half oil baffle, lower-half sleeve bearing, and lower-half oil control ring.

12.9 Install the inner retainer plate.

NOTE: The proper length of the thrust collar spacer is determined when the axial position of the rotor is checked.

12.10 Thoroughly oil and slide onto the shaft the thrust collar spacer, thrust collar key, and thrust collar.

NOTE: Depending on how long the pump has been in operation, the thrust collar locknut might turn past the point at which it was located before disassembly. If this is the case, it is necessary to redrill and tap a hole in the thrust collar to allow installation of the lock screw, which is mounted axially through the thrust collar locknut. After drilling and tapping, thoroughly clean away any metal particles and debris.

12.11 Screw the thrust collar locknut onto the shaft, and tighten it against the thrust collar.

NOTE: The shaft should be installed to within 1/16" of a point midway between the two extreme positions on the shaft in the following steps. The correct axial position of the shaft can be located by adjusting the length of the thrust collar spacer. The thrust bearing itself must have an end-play to allow for oil films and thermal expansion. The thrust bearing housing and parts are fitted to accomplish the desired end-play. Because in normal operation there is no wear, no adjustment to end-play is necessary. The thrust collar should be located to allow equal end-play on both sides, and the required length of the thrust collar spacer should be determined.

12.12 Place a mark on the thrust bearing housing and shaft to mark the relative locations, or use a dial indicator to take these measurements.

12.13 Push the pump shaft toward the coupling, and note the distance moved from the original mark.

12.14 Push the shaft toward the thrust end of the pump, and note the position relative to the mark.

12.15 Position the thrust collar spacer.

NOTE: After the locating element has been centered and located axially and the bearing housings are doweled in their proper position, they can be removed to install the seal assemblies.

12.16 Install the seal assemblies in accordance with step 13 below.

12.17 Install the thrust collar lock screw and tighten it.

12.18 Stake the lock screw in place by deforming the upper edge of the screw hole with a sharp punch.

NOTE: The lower half of the thrust bearing assembly consists of the base ring and three thrust shoes. The upper half also consists of a base ring and three shoes. The upper- and lower-half base rings are manufactured as matched pairs. The upper and lower halves, when installed, must have the same matching numbers. These numbers are inscribed on the outer surfaces of each cage. Do ***not*** *intermix the parts.*

12.19 Thoroughly oil and roll the lower half of the inner thrust bearing assembly into position.

12.20 Thoroughly oil and roll the upper half of the inner thrust bearing assembly into position.

NOTE: If proper runout cannot be attained, the thrust collar should be removed, and all mounting faces of the collar, thrust collar spacer, and thrust collar locknut checked for the presence of small particles or debris. All part faces must be square with respect to bores.

12.21 Clamp the dial indicator to the bearing housing, and place the tip against the outer face of the thrust collar.

12.22 While pushing the shaft toward the coupling end, rotate the shaft and note the variation in the indicator reading. Runout measured on the thrust collar bearing face should not exceed 0.002".

NOTE: If a new sleeve bearing is being installed, it is necessary to install the upper-half thrust bearing housing, and drill a dowel hole into the upper-half sleeve bearing, using the upper-half sleeve bearing housing as a guide. After drilling, remove and thoroughly clean the housing of all debris. Install a new dowel pin in the upper-half sleeve bearing.

12.23 Oil and install the upper-half sleeve bearing.

12.24 Install the upper-half oil control ring.

12.25 Oil and install the outer thrust bearing assembly (consisting of upper and lower base rings and thrust shoes).

12.26 Oil and install the outer retaining plate with the sealing ring.

12.27 Oil and install the oil seal ring and retainer.

12.28 Install the worm with the worm key, and replace the locknut and washer. Turn counterclockwise to tighten the locknut.

12.29 Install the main lube oil pump assembly.

12.30 Install the upper-half bearing housing with the upper-half oil baffle in place.

12.31 Install the upper-half bearing housing cap screws and tighten them.

12.32 Install the thrust bearing housing end cover with the new 1/64"-thick gasket.

12.33 Install the end cover cap screws and tighten them.

12.34 Tighten the thrust bearing deflector ring set screw installed in step 12.1.

12.35 Slide the radial bearing deflector ring against the bearing housing and back away 1/16".

12.36 Tighten the set screw and the stake set screw in place by deforming around the upper edge of the set screw hole with a sharp punch.

13. Assemble/install the inboard and outboard seal assemblies.

CAUTION: *The sleeves can easily gall on the shaft and can stick, necessitating removal by cutting. The shaft needs to be clean and lubricated prior to sleeve removal.*

13.1 Inspect the ID of the sleeve with a go/no-go gauge to ensure a proper fit. If possible, temporarily install the sleeve on the shaft before assembling the seal to verify that the sleeve will fit.

13.2 Check the pin extension and the depth of the pin hole in the mating ring with a depth micrometer and compare the results.

13.3 Check the flatness of the support surface inside the pumping ring portion of the sleeve before the O-ring and the mating ring are installed.

13.4 Cover the back side of the mating ring with tool and die blue.

13.5 Align the pin hole and pin, and place the mating ring onto the support surface. Ensure that the mating ring is not resting on the end of the pin and is not rocking in the bore.

13.6 Remove the mating ring from the bore, and check how much blue transferred onto the support surface.

NOTE: If the bluing does not transfer all around the surface or is not wide enough, remove the pins and put the sleeve in a lathe and rework the surface. After machining, replace the pins.

13.7 When the mating ring is ready to be installed, clean the entire bore.

NOTE: There are still a few installations remaining that utilize a separate pumping ring/ sleeve design. This design is the first generation seal design and is considered obsolete. If you have this design, the assembly of the complete seal cartridge is similar to this procedure except the assembly of the pumping ring with the sleeve. Follow steps 13.7 through 13.11 for installations with this design. If you do not have the separate pumping ring/sleeve design, proceed to step 13.12.

13.8 Inspect the pumping ring for sharp corners or burrs.

13.9 Lubricate the O-ring and install it in the O-ring groove on the ID of the pumping ring.

13.10 Inspect the sleeve for broken edges at the groove where the seal retainer plate fits. Cover the groove with cellophane before installing the pumping ring to avoid cutting the O-ring.

13.11 Slide the pumping ring on the sleeve, align the tapped holes with the drilled holes in the sleeve, and tighten the set screws until they bottom out. Back off the set screws 1/2 turn and stake them to prevent loosening during shaft rotation.

13.12 Mark the bore of the pumping ring, where the pins are located, with a black marker.

13.13 Mark the OD of the mating ring, where the pin holes are located, with a black marker.

13.14 Lightly lubricate the entire O-ring with an approved lubricant, and install the O-ring into the O-ring groove.

13.15 Lightly lubricate the OD of the mating ring with an approved lubricant, align the marks, and push the mating ring into the bore of the pumping ring.

13.16 Apply a light coat of Permatex, Neolube, or other approved lubricant on the threads of the plugs, and install them in the pumping ring.

13.17 Lubricate the sleeve O-ring, using an approved lubricant, and install it in the O-ring groove on the ID of the sleeve.

NOTE: This completes the assembly of the sleeve. The following steps assemble the gland plate with the seal housing and the seal head into the seal housing.

13.18 Lubricate the seal housing O-ring, and install it in the groove on the shoulder of the seal housing.

13.19 Place a gasket on the shoulder of the seal housing, and install the gland plate on the seal housing.

NOTE: Verify that a 0.062" x 45° chamfer exists on the inside edge of the spacer at either edge. If the chamfer is missing, machine a chamfer on the spacer.

13.20 Inspect the hub of the seal housing where the bellows fits for any sharp corners, burrs, or deep scratches. Repair any irregularities found, if possible.

13.21 Slide the spacer on the hub of the seal housing with the chamfer pointing toward the bottom of the bore. Ensure that it rests against the bottom of the bore and is not rocking.

13.22 Install the spring over the spacer.

NOTE: When installing the bellows/seal head assembly, be sure to protect the sealing face of the primary ring by placing a cardboard disc, supplied with the seal, over the sealing face.

13.23 Lightly lubricate the ID of the tail section of the bellows with glycerin or an approved lubricant.

13.24 Slide the seal head assembly onto the hub of the seal housing until it rests firmly against the spacer. Hold this position for approximately 15–30 minutes or until the bellows adheres to the hub of the seal housing at that position.

13.25 Clean the sealing faces with alcohol and a lint-free cloth.

13.26 Apply demineralized water to the sealing faces.

13.27 Carefully slide the sleeve inside the seal housing, and rotate the entire assembly to a vertical position with the gland on top.

NOTE: The first generation seal design uses a gasket, while the second generation seal design uses an O-ring to seal between the gland plate and the auxiliary gland.

13.28 Install an O-ring or gasket on the back face of the gland plate.

13.29 Insert the wave spring into the bore of the gland plate.

13.30 Place the packing assembly, consisting of a packing retainer and one ring of packing, on top of the wave spring.

NOTE: When installing the auxiliary gland, ensure that the drive pin engages the slots of the packing retainer. Earlier seal designs used studs and nuts instead of cap screws.

13.31 Install the auxiliary gland and tighten the cap screws.

13.32 Engage the retainer plate into the groove on the sleeve to properly locate the sleeve and set the seal.

13.33 Secure the retainer plate to the gland plate with the cap screws.

13.34 Place the drive collar onto the sleeve, but do not tighten the set screws.

13.35 Install the retainer plate on the back of the seal housing.

13.36 Install an O-ring in the groove on the front face of the seal housing.

NOTE: This completes the assembly of the seal cartridge. The remaining steps cover seal installation in the pump. If the ID of the sleeve was not checked during the assembly of the seal, use a go/no-go gauge now to check the ID of the sleeve. If the seal cartridge has been stored for an extended period, separate the sleeve from the seal housing and check the condition of the seal faces. If they are satisfactory, clean both faces with alcohol, and apply demineralized water on the sealing faces before reassembly.

13.37 Inspect the shaft surface for burrs, especially where the sleeve will be installed. Use stones, Scotch-Brite, and emery paper to remove any high spots or burrs.

13.38 Inspect the ID of the sleeve, ensuring that the O-ring is in the groove.

13.39 Lubricate the ID of the sleeve with glycerin. Apply a light film of lubricant the entire length.

13.40 Cover the surface of the shaft where the sleeve will be installed with Neolube.

13.41 After the Neolube dries, lubricate the surface of the shaft with glycerin.

NOTE: The deflector plate has to be installed on the shaft first and then assembled to the front face of the seal plate—only on the inboard side. The use of the guide rods is optional.

13.42 Install the seal cartridge on the shaft.

13.43 Install the bolts in the housing, and tighten them in a criss-cross pattern.

13.44 Torque the bolts to the proper torque.

13.45 Install the collar on the shaft, but do not tighten the set screws until the bearings are set.

13.46 Tighten the set screws, and disconnect the seal retainer plate only after the bearing installation and pump alignment are completed.

14. Install the pump half of the coupling.

NOTE: The pump half of the coupling assembly consists of the coupling key, coupling hub, and coupling sleeve.

14.1 Slide the sleeve onto the shaft.

NOTE: In order to slide the hub onto the shaft, the following step must be accomplished rapidly because thermal growth of the shaft quickly reduces the clearance and prevents the hub from sliding onto the shaft. If there are match marks on the coupling hub and sleeve, they must be aligned when the gear teeth are engaged.

NOTE: It is preferable that the coupling hub be heated in a hot oil bath. If a hot oil bath is not available, the hub can be heated by a bearing heater or by careful use of an acetylene heating torch. Carefully play the torch on the end faces and bore of the coupling. Do ***not*** *play the flame directly on the gear teeth or the coupling will be unfit for further use. Before heating, inspect the shaft and coupling bore for cleanliness and freedom from burrs. The coupling will not slide into position if burrs are left on the bore of the coupling or shaft.*

14.2 Heat the coupling hub to 250–350°F.

14.3 While the sleeve is held in position centered on the shaft, slide the hub onto the shaft and coupling key.

15. Verify the pump-shaft-to-gear-shaft alignment. Re-align if necessary before proceeding.

NOTE: The coupling spacer and coupling sleeve have match marks that must be paired together before bolting the spacer to the sleeve. Failure to observe the matching of these marks causes vibrations during operation. Couplings that utilize keys must have the keys placed 180° apart.

16. Lubricate the coupling, and install the coupling spoolpiece to the coupling sleeve.

17. Reinstall the interference removed in step 1.

17.1 Reinstall the coupling guard.

17.2 Reinstall the water lines.

17.3 Reinstall the inboard and outboard oil supply and the return lines.

17.4 Reinstall the restraints and supports (hangers).

4.2.4.2 Overhaul Gear Reducer

The following tools are required to overhaul the gear reducer:

- Dial indicator and magnetic base
- Outside micrometer, 1“
- Rymple cloth
- Lubricating oil
- Coupling grease
- Plastigage, 0.020 thickness

1. Remove the gear reducer.

   1.1 If the gear reducer is not uncoupled from the pump, place a match mark on the flange sleeves to aid in reassembly.

   1.2 Remove the lube plugs.

   1.3 Remove the coupling flange locknuts.

   1.4 Remove the coupling flange bolts.

   1.5 Remove the flange gasket.

2. If the gear reducer is not uncoupled from the motor, repeat steps 1.1 through 1.5 for the motor coupling.

3. If the reducer is going to be physically removed, remove the cap screws holding it to its base.

4. Remove the gear reducer covers. Figure 4-16 shows a typical gear reducer.

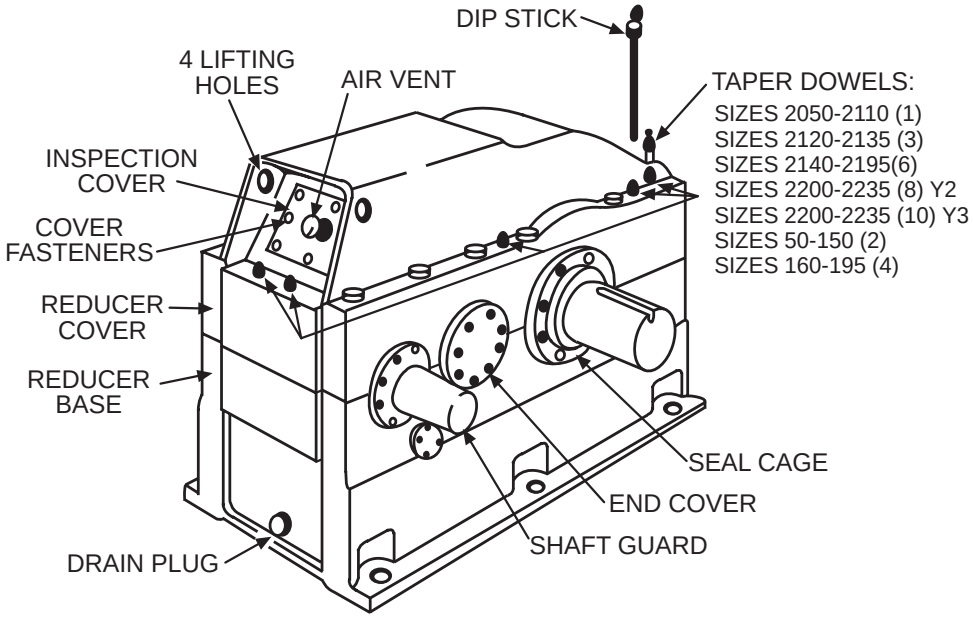


Figure 4-16
Typical Gear Reducer

   4.1 Drain the oil and remove the dipstick.

NOTE: A gear puller can be used to remove the flex hubs. The removal of the coupling hub might require the use of heat. Do ***not*** *exceed 275°F. Monitor the temperature with a temp stick or a contact pyrometer.*

4.2 Remove the coupling hub.

4.3 Remove the coupling flange.

4.4 Remove the high-speed shaft guard.

4.5 Remove the fan guards.

4.6 Remove the fans.

4.7 Remove the reducer cover fasteners.

4.8 Remove all cap screws above the housing split from the end covers and the seal cage assemblies.

4.9 Loosen, but do **not** remove the cap screws on the end covers, and seal the cage assemblies below the housing split three or four turns.

4.10 Tighten the nuts on the taper dowels, and lift out all dowels.

4.11 Lift the reducer cover straight up without damaging the gear teeth.

5. Remove the shaft assemblies.

5.1 Place a sling around the high-speed shaft. Take up the slack just enough to take the weight off the bearings.

5.2 Remove the cap screws and seal cage assembly, along with the shim gaskets.

5.3 Wire the shims to the seal cage and end cover so they are available for reference when reassembling the unit.

5.4 Lift the shaft assembly out of the base without damaging the teeth.

5.5 Place a sling around the intermediate speed shaft. Take up the slack just enough to take the weight off the bearings.

5.6 Remove the cap screws and both end covers along with the shim gaskets.

5.7 Wire the shims to their mating end cover so they are available for reference when reassembling the unit.

5.8 Lift the shaft assembly out of the base without damaging the teeth.

5.9 Place a sling around the low-speed shaft. Take up the slack just enough to take the weight off the bearings.

5.10 Remove the cap screws, seal cage assembly, and end cover along with the shim gaskets.

5.11 Wire the shims to the seal cage and end cover so they are available for reference when reassembling the unit.

5.12 Lift the shaft assembly out of the base without damaging the teeth.

6. Disassemble, repair, and reassemble the seal cage assembly.

6.1 Remove the seal baffle.

6.2 Block up the seal cage and press or drive out the seal.

6.3 Remove the old sealing compound from the seal cage bore.

6.4 Coat the seal cage bore with Permatex 2.

NOTE: New seals leak if the seal's lips are cut or if the seal's rubbing surface on the shaft has been altered.

6.5 Position the seal and place a cylindrical tool against the seal. Drive or press the seal until it is fully seated.

6.6 Coat the seal lips with an approved grease.

7. Inspect and replace the bearings if necessary.

7.1 Inspect the bearings carefully for any signs of wear.

7.2 If the bearings need to be replaced, remove the bearings with a wheel puller, or press them off by applying force to the inner face only.

NOTE: Flame should not be applied directly to the bearings, and the bearings should not be allowed to rest on the bottom of the heated container.

7.3 If new bearings are to be installed, heat the bearings to a maximum of 275°F in a bearing heater or oven, and slide or press the bearings onto the shaft tight against the shaft shoulder.

7.4 Thoroughly coat all bearings with oil.

8. Assemble the gear reducer.

NOTE: During assembly, all seal cages and end cover guards should be positioned with the word "TOP" in the upright position to permit circulation of the lubricant.

8.1 Verify the cleanliness of the gearbox.

8.2 Measure the thickness of the shim gaskets attached to the seal cage for the low-speed shaft, and replace the shims with new shims of equal thickness.

NOTE: The shaft, gear, and bearings should be maintained in a level position during installation.

8.3 Rig the low-speed shaft for installation.

8.4 Lower the low-speed shaft into the housing until the bearings are in place but are not supporting the shaft.

8.5 Install the new shim gaskets and the seal cage with the housing bottom-half cap screws.

8.6 Install the end cover without shims with the housing bottom half cap screws, but do not tighten the cap screws.

NOTE: Steps 8.7 through 8.13 measure the axial float. Enough force must be applied to the shafts to fully seat the tapered roller bearings when measuring axial float. This can be accomplished by using pry bars against the shaft ends. It could be helpful to space three feeler gauges under the end cover to keep the gap even until the correct float is obtained.

8.7 Set up a dial indicator to measure the axial end play.

8.8 Tighten the cap screws on the end cover evenly until the axial float is between 0.000–0.001".

8.9 Measure the gap, as indicated in Figure 4-17.

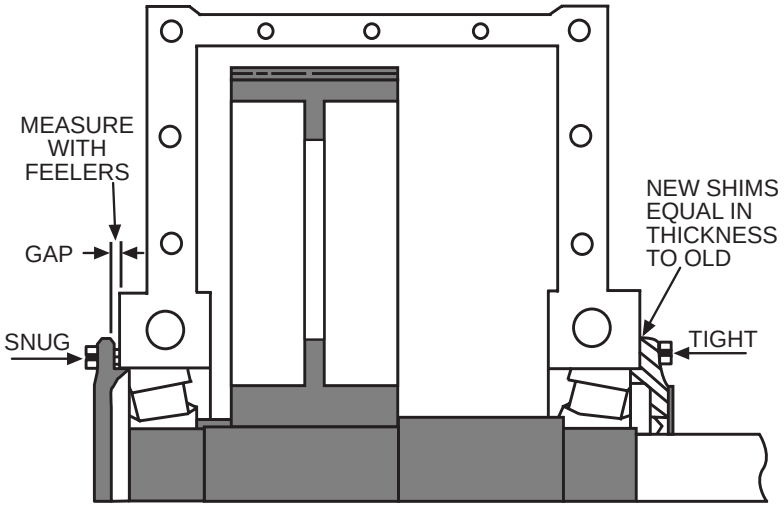


Figure 4-17
Gear Reducer Gap

8.10 Determine the starting shim amount by increasing the gap measurement by 10%.

8.11 Remove the cap screws and the end cover, and reinstall with the starting shim gaskets.

8.12 Measure the axial float, and add shim gaskets until the axial float is between 0.0005–0.002".

8.13 Remove 0.007" of shims to achieve a 0.000–0.006" preload of the shaft.

8.14 Remove the dial indicator.

8.15 Lower the shaft onto the bearings and remove the sling.

8.16 Measure the thickness of the shim gaskets attached to the end cover on the motor side (small shaft side) of the intermediate-speed shaft, and replace them with new shims of equal thickness.

8.17 Lower the intermediate-speed shaft into the housing until the bearings are in place.

8.18 Install the new shim gaskets and the input side end cover with the housing bottom-half cap screws.

8.19 Install the other end cover without shims with the housing bottom-half cap screws.

8.20 Set up a dial indicator to measure the axial float.

8.21 Tighten the cap screws on the end cover evenly until the axial float is between 0.000–0.001".

8.22 Measure the gap, as indicated in Figure 4-17.

8.23 Determine the starting shim amount by adding the gap measurement to 0.005" and increasing this sum by 10%.

8.24 Remove the cap screws and end cover, and reinstall with the proper starting shim gaskets.

8.25 Measure the axial float, and adjust the shim amounts until the axial float is 0.005–0.008".

8.26 Measure the thickness of the shim gaskets attached to the seal cage from the input side of the high-speed shaft, and replace them with new shims of equal thickness.

8.27 Lower the high-speed shaft into the housing until the bearings are in place.

8.28 Install the new shim gaskets and the seal cage with the housing bottom-half cap screws.

8.29 Install the other seal cage without shims with the housing bottom-half cap screws.

8.30 Set up a dial indicator to measure the axial float.

8.31 Tighten the cap screws on the output side seal cage evenly until the axial float is between 0.000–0.001".

8.32 Measure the gap, as indicated in Figure 4-17.

8.33 Determine the starting shim amount by adding the gap measurement to 0.005" and increasing this sum by 10%.

8.34 Remove the cap screws and output side seal cage, and reinstall them with the proper thickness of shim gaskets.

8.35 Measure the axial float, and adjust the shim amounts until the axial float is 0.005–0.008".

8.36 Remove the dial indicator.

9. Assemble the gear reducer cover to the base.
    - 9.1 Loosen the cap screws holding the seal cages, shaft guards, and end covers to provide clearance for the reducer cover while holding the upper portions of the shims to their adjoining parts with thin wire or string.
    - 9.2 Coat the base joints, air vent, and oil drain threads with Form-a-Gasket, ensuring that excessive quantities are not deposited near the bearings.
    - 9.3 Lower the cover carefully onto the base without bumping the gear teeth.
    - 9.4 Drive in the taper dowels.
    - 9.5 Remove all wires or strings holding the shims.
    - 9.6 Install and torque to the proper torque the cap screws for the seal cages, end covers, and shaft guards.
    - 9.7 Insert the cover bolts, and torque to the proper torque.
    - 9.8 Remove the inspection cover cap screws and inspection cover.
    - 9.9 Verify that the axial float is still present.
    - 9.10 Reinstall the fan and fan guards.
    - 9.11 Turn the gear train over by hand as a final check for possible interference, and readjust as necessary.
    - 9.12 Fill the reducer with an approved lubricant to the level indicated on the dipstick.
    - 9.13 Install the inspection cover, cap screws, and plugs, and tighten them snug tight.
10. If the coupling halves were removed, reassemble the coupling.
    - 10.1 Install the flange sleeves with the seal rings on the shafts.
    - 10.2 Mount the flex hubs on their respective shafts so that each hub face is flush with the end of its shaft.

4.3 Predictive—Positive Displacement Pump Conditions To Be Monitored

The applicable predictive maintenance techniques identified through RCM analysis are:

- Vibration monitoring
- Temperature monitoring
- Suction stabilizer pressure monitoring
- Discharge dampener pressure monitoring
- Discharge pressure monitoring
- Water block packing leakage
- Slip

- Plunger surface condition
- Crankcase oil analysis
- Gear condition

4.3.1 Vibration

In all rotating or reciprocating machinery, a condition of mechanical vibration is present because of the force and energy levels inherent in the equipment. Every machine fault, whether it is mechanical or electrical in nature, generates vibration at a specific frequency. For example, unbalance occurs at a frequency equal to shaft rotation speed. This is because the heavy spot of the unbalanced rotor passes the vibration sensor once per shaft revolution. Similarly, frequencies can be calculated corresponding to many other common machine faults: misalignment, cavitation, soft foot, bearing deterioration, and many others. A spectrum is a calculated data display of frequency versus amplitude. The frequency helps diagnose the source of the vibration, while the amplitude helps determine the severity of the problem.

A successful vibration monitoring program includes measurements for the following characteristics, which are needed to define the vibration:

- ***Vibration Frequency*** - The amount of time required to complete one full cycle of the vibration is called the *period* of the vibration. Vibration frequency is simply a measure of the number of complete cycles that occur in a specified period of time. Frequency is related to period of vibration by this simple formula:

  Frequency = 1 / Period

  It is not necessary to determine the frequency of the vibration by observing the vibration time waveform, noting the period of the vibration, and then calculating the inverse of the period to find the frequency. Nearly all data collectors and vibration analyzers provide a direct readout of the vibration frequencies being generated by the machine.

  The forces that cause vibration are usually generated through the rotating motion of the machine's parts. Because these forces change in direction or amplitude according to the rotational speed (revolutions per minute or RPM) of the machine's components, it follows that most vibration problems have frequencies that are directly related to the rotational speeds. Vibration analysis is truly a process of elimination. Additional tests and measurements can be taken to reduce the number of causes. However, knowing the frequency of vibration and how the frequency relates to the rotating speed of the machine components is usually the first step in the analysis process.

  Vibration frequency is a diagnostic tool needed to help identify specific mechanical or operational problems. Whether vibration frequency analysis is necessary depends on the magnitude of the vibration, expressed as its vibration amplitude. Vibration amplitude can be measured and expressed as displacement, velocity, or acceleration.

- ***Vibration Displacement*** - Vibration displacement is the total distance traveled by the vibrating part from one extreme limit of travel to the other. This distance is also called the *peak-to-peak displacement*. The use of vibration displacement can be very cumbersome, however, because it must be used with vibration frequency to have any meaning. Machinery vibration is not always simple or at one frequency; in more complex cases, it is nearly impossible to use vibration displacement to judge the overall condition of a machine.
- ***Vibration Velocity*** - Vibration velocity is a measurement of the speed of movement of a machine or machine component because velocity undergoes oscillating motion. Each source of vibration contributes to the fatigue of machine components, and the overall condition can be determined only by taking into account all frequencies of vibration. This is accomplished by measuring vibration velocity.

  Velocity is definitely a characteristic of the vibration, but since it is constantly changing throughout the cycle, the highest or "peak" velocity is selected for measurement. The advantages of measuring vibration velocity instead of vibration displacement include:

  – Vibration velocity is a direct indicator of fatigue because it takes into account both displacement and frequency.

  – It is not necessary to know the frequency of vibration in order to evaluate the severity of vibration velocity because frequency is already a part of velocity.

  – A measurement of overall vibration velocity is a valid indicator of the overall condition of a machine whether the vibration is simple (one frequency) or complex (more than one frequency).

  Although vibration velocity provides many positive functions, there is no absolute vibration level that divides continuous operation and immediate failure.
- ***Vibration Acceleration*** - Vibration acceleration is another important characteristic of vibration that can be used to express the amplitude or magnitude of vibration. Technically, acceleration is simply the rate of change of velocity. Since the value of vibration acceleration is constantly changing, the highest or peak acceleration is selected for measurement. Evaluating vibration acceleration measurements is similar to evaluating vibration displacement measurements in that it is necessary to know the specific frequency of vibration.

  Although the vast majority of machinery failures are due to fatigue, which is directly related to vibration velocity, there are two other mechanisms of machine failure: stress and force. These two parameters are directly related to vibration displacement and vibration acceleration, respectively. Due to brittleness, machine components can crack or break if bent or deflected (displaced) beyond a certain limit. High amplitudes of vibration displacement can cause mounting bolts to snap, welds to give way, or foundations to crack—not because of fatigue, but because they were deflected beyond their yield points. High amplitudes of vibration displacement that result in stress failures typically occur at very low vibration

frequencies, generally below 600 CPM (10 Hz). Whenever it is anticipated that vibration frequencies might be present at frequencies below 600 CPM, measurements of vibration displacement are recommended.

As a general rule, fatigue failures result from vibration frequencies between approximately 600 CPM and 120,000 CPM. Therefore, when vibration frequencies within this range are anticipated, measurements of vibration velocity are recommended.

The easiest way to demonstrate force (and acceleration) as a cause of trouble is to consider striking an object with a hammer. The impact might not cause significant displacement or velocity; however, resultant damage can be considerable. Since force equals mass times acceleration, it is apparent that vibration acceleration is directly proportional to vibratory force. And, since vibration acceleration increases proportionally to the square of vibration frequency, very large vibratory forces can occur at high frequencies of vibration even though the displacement and velocity amplitudes might be quite small.

Excessive forces generally cause deformation of the surfaces of machine components, such as gear teeth and rolling element bearings. High forces can also cause the lubricating film to break down, resulting in friction, heat generation, and ultimate failure. Because of the importance of vibratory forces at high frequencies, vibration acceleration measurements are recommended whenever vibration frequencies above 120,000 CPM (2000 Hz) are anticipated. The most common source of such high frequencies are gear mesh frequencies and harmonics or multiples of gear mesh frequencies on gear drives.

4.3.2 Temperature

Variations in temperature of the fluid pumped cause changes in the specific weight and viscosity of the fluid, with resultant changes in the performance of the pump. Reduced viscosity has an influence on efficiency by increasing the internal leakage losses, reducing friction losses, and reducing flow losses. The net effect of a reduction in viscosity due to higher temperature depends on specific speed and on the design of the pump.

4.3.3 Dampener Pressure

4.3.3.1 Suction Stabilizer Pressure

Suction stabilizer pressure assists in maintaining sufficient NPSH to the suction of the charging pump. Typical pressures (nitrogen or hydrogen) maintained in the suction stabilizers range from 15–30 psi. Maintaining a pressure that is too low in these stabilizers can result in gases coming out of solution, causing cavitation, valve damage, and damage to other internal pump components, as well as a reduction in the capacity of the pump.

4.3.3.2 Discharge Dampener Pressure

The discharge dampeners utilize a gas pressure acting on a bladder against system pressure to minimize fluctuations in pump output. There are two modes of operation for these dampeners: low-pressure and high-pressure. A low-pressure operation, typically used during plant shutdown, precharges the dampener to approximately 200 psi. A high-pressure operation, typically the normal operating mode, precharges the dampener to approximately 1380 psi. These pressures can vary slightly from plant to plant, depending on normal system operating pressures.

4.3.4 Flow

4.3.4.1 Discharge

Reciprocating pumps inherently have periodic impulses. However, multipiston pumps reduce the magnitude of these variations. Pulsation, although appearing as a fluctuation in pressure, actually arises as a fluctuation in flow rate because pressure is a dependent quantity. The discharge of a positive displacement pump can be calculated as follows:

$$Q_d = DN\text{-}C_s\Delta PD/\mu$$

where: Q_d = discharge flow

D = displacement

N = speed

C_s = slip coefficient

ΔP = pressure differential

μ = viscosity

The system into which the liquid is pumped has some resistance (R), where $R = \Delta P / Q_d$ or $Q_d = \Delta P / R$. Equating, $Q_d = \Delta P / R = DN\text{-}C_s\Delta PD/\mu$. Noting that $\Delta P = P_{out} - P_{in}$, it can be shown that this relationship between pressure, flow, and discharge resistance applies:

$$P_{out} = (DN)/(1/R + C_sD/\mu) + P_{in}$$

If N, R, C_s, and P_{in} remain constant, the discharge pressure and flow are constant. But C_s depends on the cube of the pump internal clearances and can vary during each revolution, thus introducing large flow fluctuations.

4.3.4.2 Water Block Packing Leakage

Pumps using compression packing require both the plunger and stuffing box surfaces to be in good condition to provide a satisfactory seal. When packing leakage becomes excessive, some of the packing glands in older design pumps can usually be adjusted by

tightening the packing gland adjusting nuts. More recent design pumps have no adjustment, and the packing must be replaced if leakage becomes excessive. Care must be exercised to ensure that the packing adjusting nuts are not tightened too much. Tightening the packing gland adjusting nuts too much can result in the packing seizing to the shaft and overheating, necessitating replacement. When replacement is necessary, the packing should undergo a run-in period, if possible, to ensure proper sealing. The adjusting nuts should be tightened finger tight when the packing is replaced and tightened in small increments until the packing leak rate is satisfactory, if your design allows adjustment. If the design does not allow adjustment, the packing gland nuts must be torqued to the proper torque.

4.3.5 Slip

Slip is the capacity loss as a percentage of suction capacity and can be calculated with the following equation:

$$S = Ve_1 + B_1 + V_1$$

where: Ve_1 = volumetric efficiency loss

B_1 = stuffing box loss (usually negligible)

V_1 = valve loss (usually 2–10% depending on valve design and conditions)

Volumetric efficiency loss can be calculated by using the formula $Ve_1 = 1 - Ve$, where volumetric efficiency (Ve) is the ratio of discharge volume to suction volume.

4.3.6 Plunger Surface Condition

Although plunger clearance measurements are important, the surface finish of the plunger is the most critical condition. Unsatisfactory plunger surface finish results in the premature wearing out of the packing, necessitating replacement. In-service plungers should not exceed a surface finish of 16 RMS, while the criteria for new plungers is 6–10 RMS and can be measured with a profilometer. Hardness, unless otherwise specified, should be a minimum of Rockwell C 35.

4.3.7 Crankcase Oil Analysis

Oil samples should be taken periodically to ensure that the oil is not contaminated with foreign material, thereby reducing its lubricating ability. Typically, oil analysis is done on a periodic basis, such as every six months and prior to overhaul or startup.

4.3.8 Gear Condition

Because larger gear boxes run at slower speeds, their life is normally longer than smaller, high-speed gear boxes. Most problems in large gear boxes occur with gears and bearings. Gears usually fail because an interruption in operation causes wear and fatigue. Several common problems in large gear boxes are listed below:

- ***Metal-to-metal contact during operation*** is the result of misalignment or poor lubrication.
- ***Oxidized oil*** forms carbon deposits and loses lubricity.
- ***Fine abrasive dirt and foreign material in the lubrication system*** are commonly debris from gears and bearings, scale, rust, and other abrasive material.
- ***Corrosion of the gear surfaces*** is caused by a chemical reaction of the active ingredients in the oil, such as acid, moisture, and other chemicals.
- ***Ambient moisture*** decreases the lubricity of the oil and forms rust on inside surfaces of the gear box housing. The rust flakes with time and causes surface scratches and dents in rotating elements.

Surface fatigue is common in heavy-duty gear boxes, where slow speeds create high torques and stresses on gear teeth. When gear material is stressed beyond its endurance limit, the resulting surface fatigue causes spalling and pitting. Stresses occur when mating surfaces do not conform to each other. Lack of proper alignment creates poor load distribution resulting in surface cracks. As a result, large chunks of material can flake off.

Thrust bearings are the most important components in large gear boxes. Because of their large size, installation is important. Failures are similar for all bearings. Flaking starts with the development of small fatigue cracks in the surface of the ring or roller elements. These cracks cause material fragments to break loose from the raceways. The first flakes are often very small. The flaked area spreads by breaking off additional pieces of the surface until flaking covers the entire surface of the loaded zone. Causes of flaking include:

- Excessive operating loads
- Overloading due to heat expansion
- Housing misalignment
- Improper installation
- Excessive vibration without rotation
- Contamination

Cracks and fractures in bearings occur as a result of heavy loads applied for a short time, high-speed load variations, and overstrain. An overstrain, which can lead to fracture, results when an inner ring is driven too far onto a tapered seat.

Indentations in bearing raceways result when foreign particles are present inside the bearing and are pressed between the rolling elements and rings. Heavy or impact loads cause local deformation at the point of roller contact. Adhesive wear is common in heavy-duty gear boxes, where metal from outside sources such as bolts, nuts, and tools is crushed and fused to the bearing or its shaft. Abnormal bearing wear occurs when housing seals fail. Abrasive particles enter the bearings and damage raceways and flanges. Poor lubrication, lack of lubrication, and contaminants such as metal flakes and rust also cause wear.

4.4 Positive Displacement Charging Pump Maintenance Practices

Preventive maintenance practices that can be accomplished by plant personnel are:

- Pump disassembly and inspection
- Bearing disassembly and inspection

Corrective maintenance practices that can be accomplished by plant personnel are:

- Packing replacement
- Motor-to-gearbox and gearbox-to-pump coupling alignment
- Plunger replacement
- Crosshead and connecting rod repair or replacement
- Removal and replacement of crank shaft
- Suction and discharge valve replacement
- Fluid cylinder replacement

4.4.1 Inspection Techniques

4.4.1.1 Operational Inspection

Assessing the condition of operating machinery is within the capabilities of qualified, experienced, and attentive operators. Inspection instruments include common hand tools:

- Flashlight
- Mirror
- Magnifying glass
- Wiping cloths

4.4.1.2 Off-Line Inspection

Off-line inspections are normally intrusive in nature, requiring disassembly and measurement of internal wear surfaces. Inspection points and applicable inspection criteria are *italicized* in the disassembly procedure outlined in Section 4.4.2.

4.4.2 Disassembly

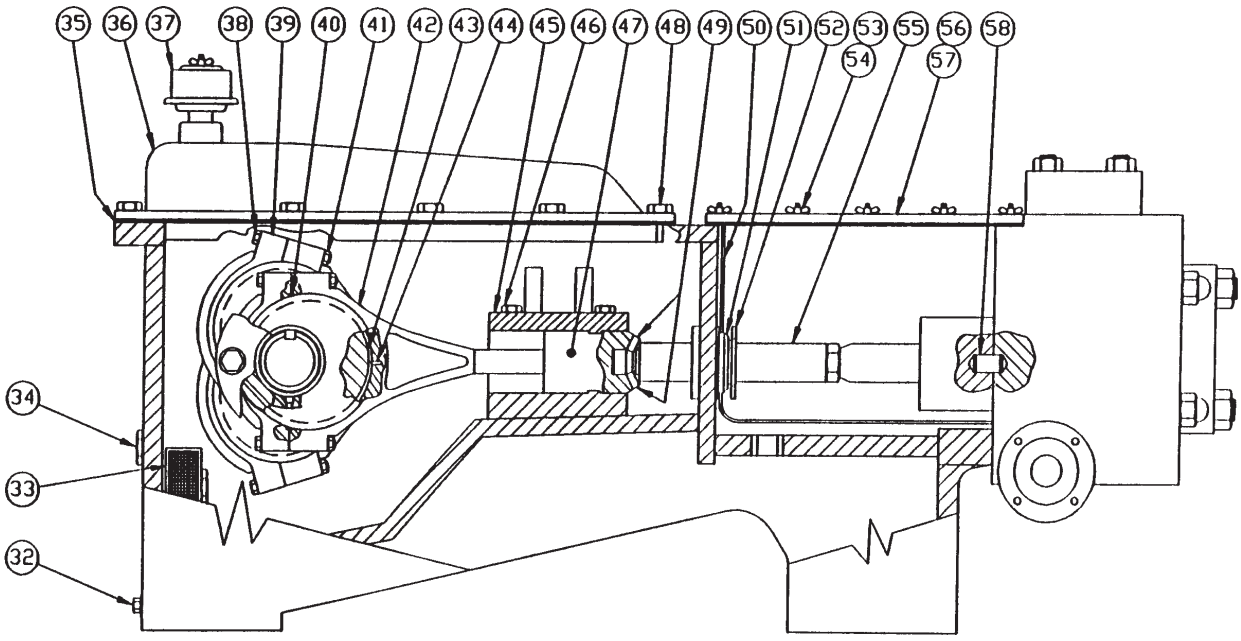
4.4.2.1 Power End Disassembly for Inspection

Tools and equipment that might be needed for pump power end and pump end disassembly include:

- Hoist, 2-ton capacity
- Lifting straps
- Socket set, 3/4" drive
- Socket drive handles, 1/2" and 3/4"
- Socket drive extension, 1/2"
- Valve fastener gripping tool
- Adjustable wrench
- Packing removal tool
- Crosshead bearing spanner wrench
- Baffle gland wrench
- Valve removal tool assembly
- Valve fastener removing tool
- Allen wrench, set
- Acetylene torch, rosebud tip (#10)
- Gloves, heatproof

General procedural steps for disassembly are as follows (refer to Figure 4-18):

1. Drain the oil from the pump.
2. Remove the base cover screws and the base cover.
3. Disconnect the couplings.
    - 3.1 Remove the cap screws on the coupling guard between the gear reducer and the motor.
    - 3.2 Remove the cap screws on the coupling guard between the gear reducer and the pump.
    - 3.3 Remove the coupling guards on both couplings.
    - 3.4 Matchmark the coupling hubs.
    - 3.5 Remove the bolts, nuts, and lockwashers holding the coupling flanged sleeves together on both couplings.
    - 3.6 Slide the flanged sleeves back on their respective shafts.



32 Oil Drain Petcock
33 Oil Suction Strainer
34 Oil Level Gauge
35 Base Cover Gasket
36 Base Cover
37 Oil Filter Cap
38 Connecting Rod Bolt
39 Connecting Rod Cap
40 Connecting Rod Dowel
41 Connection Rod Nut
42 Connecting Rod
43 Connecting Rod Bearing
44 Connecting Rod Bearing Dowels
45 Crosshead Guide Cover
46 Crosshead Cover Cap Screw
47 Crosshead
48 Base Cover Screws
49 Plunger Adapter Set Screws
50 Well Lining
51 Baffle Stuffing Box
52 Packing Adjusting Screw
53 Well Cover Stud
54 Well Cover Wing Nut
55 Plunger Adapter
56 Well Cover
57 Well Cover Gasket
58 Cylinder Dowel

Figure 4-18
Pump Power End

4. Remove the gear reducer.

NOTE: Any shims used on the gear reducer should be kept together and marked for location to make assembly easier.

4.1 Remove the cap screws that hold the gear reducer to the sub-base.

4.2 Remove the gear reducer from the sub-base, using the proper lifting equipment.

***CAUTION:** Do not apply a flame from the torch directly to the gear teeth. The maximum temperature should not exceed 275°F.*

5. If the shaft contains a flexible hub, remove the hub with a gear puller while slowly heating the hub.

6. Remove and disassemble the connecting rods and crosshead assemblies. A typical configuration drawing for these components is shown in Figure 4-18.

   6.1 Remove the crosshead cover cap screws.

   6.2 Remove the crosshead cover.

   6.3 Loosen, but do ***not*** remove, the crosshead bearing set screws.

   6.4 Using the crosshead bearing tool, unscrew the crossheads from the crosshead bearing retainers.

   6.5 Remove the front cylinder caps from the cylinder.

   6.6 Slide the crosshead assemblies as far to the front of the pump as possible.

   6.7 Rotate the shaft until one of the connecting rods is positioned as far to the rear as possible. This will provide the necessary space for removal of the connecting rod from the crosshead.

   6.8 Remove the connecting rod nuts.

NOTE: Ensure that each connecting rod is stored with its respective cap as a unit to prevent the mixing of parts.

   6.9 Remove the connecting rod cap and the connecting rod.

   6.10 Remove the other two connecting rods by repeating steps 6.7 through 6.9.

   6.11 Remove the bearing halves from the connecting rod caps.

   6.12 Remove the bearing halves from the connecting rods.

   6.13 Inspect the bearing surfaces on the cams for roughness or scratches. Figures 4-19 and 4-20 show normal bearing wear patterns, while Figure 4-21 shows abnormal wear patterns.

   6.14 Clean the bearing surfaces as necessary.

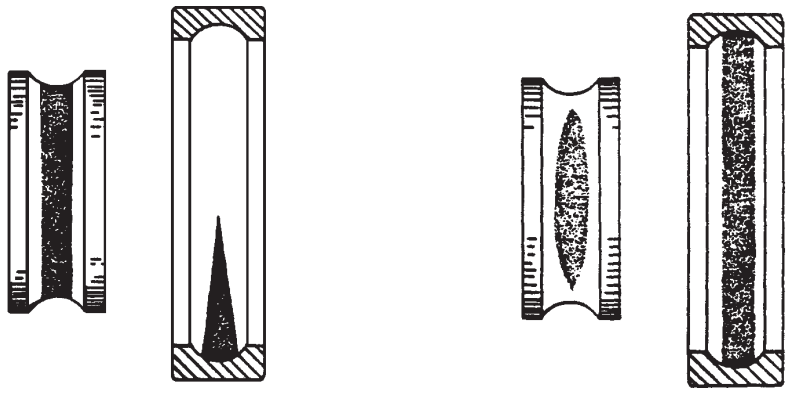


Figure 4-19
Bearing Wear Patterns for a Normal Radial Load

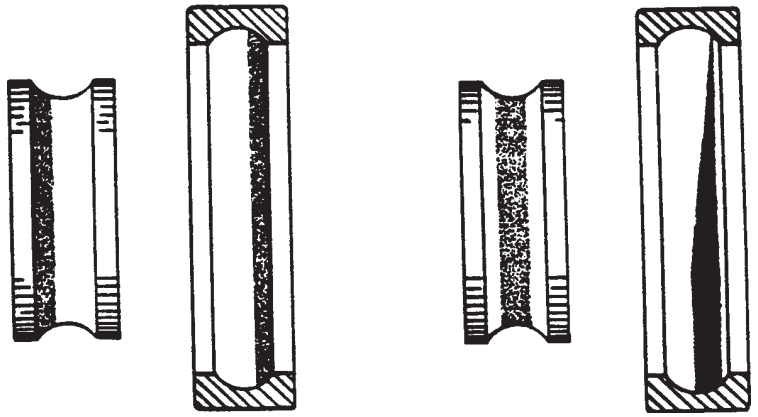


Figure 4-20
Bearing Wear Patterns for a Normal Thrust Load

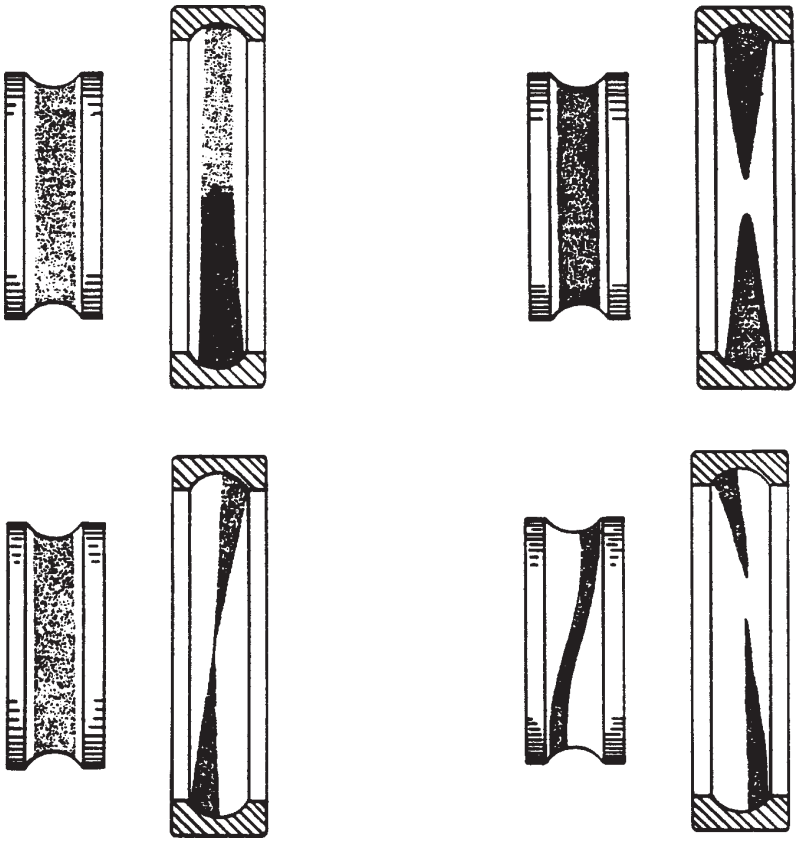
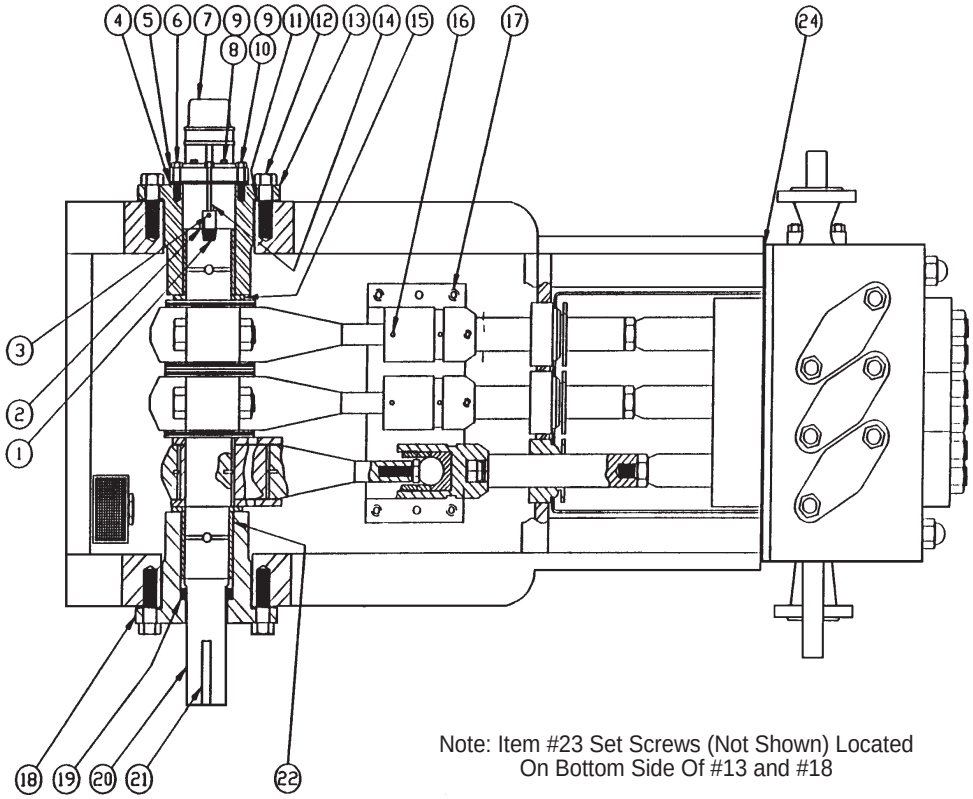


Figure 4-21
Abnormal Bearing Wear Patterns

7. Remove both oil lines to the oil pump. Refer to Figure 4-22.
8. Remove the bearing retainer cap screws.
9. Support the weight of the shaft, using the proper lifting equipment.
10. Slide the bearing retainers from the shaft.



1. Eccentric Shaft Pipe Plug
2. Oil Pump Coupling
3. Oil Pump Coupling Set Screw
4. Retainer Flange Gasket
5. Eccentric Bearing Retaining Flange
6. Oil Pump Gasket
7. Oil Pump
8. Oil Pump Mounting Bolts
9. Retainer Flange & Oil Pump Lockwasher
10. Eccentric Bearing Retaining Flange Bolts
11. Eccentric Bearing Retainer Gasket
12. Eccentric Bearing Retainer Cap Screw
13. Eccentric Bearing Retainer
14. Oil Pump Key
15. Eccentric Shaft Thrust Washer
16. Crosshead Bearing Set Screw
17. Crosshead Cover Dowel
18. Eccentric Bearing & Seal Retainer
19. Eccentric Shaft Seal
20. Eccentric Shaft Bearing
21. Eccentric Shaft Key
22. Eccentric Shaft Bearing
24. Cylinder Gasket

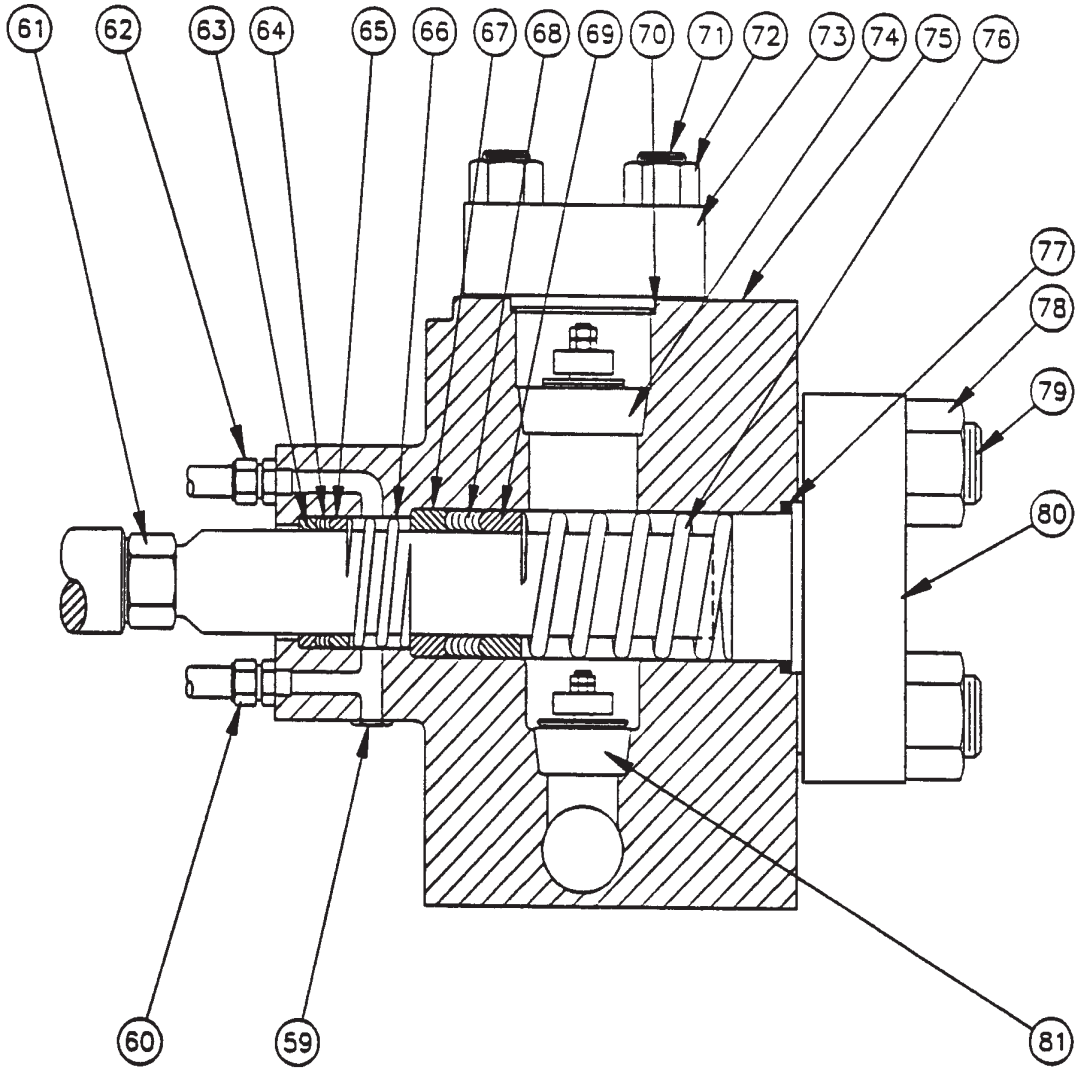
Figure 4-22
Typical Positive Displacement Pump Configuration

NOTE: Be extremely careful when removing the shaft to prevent the eccentric shaft from contacting the pump base, because this could potentially damage the journals.

11. Tip the shaft to an angle that allows its easy removal from the base.
12. Remove the shaft from the base.

4.4.2.2 Pump End Disassembly for Inspection

1. Disassemble the pump cylinder assembly. Figure 4-23 shows a typical positive displacement pump end configuration.



59. Plug
60. Packing Cooling System Outlet
61. Plunger
62. Packing Cooling System Inlet
63. Secondary Packing Adjusting Ring
64. Secondary Packing
65. Secondary Plunger Rong
66. Secondary Packing Spring
67. Primary Packing Adjusting Ring
68. Primary Packing
69. Primary Plunger Ring
70. Upper Cap Gasket
71. Upper Cap Stud
72. Upper Cap Stud Nut
73. Upper Cap
74. Discharge Valve
75. Cylinder Block
76. Primary Packing Spring
77. Front Cap Gasket
78. Front Cap Stud Nut
79. Front Cap Stud
80. Front Cap
81. Suction Valve

Figure 4-23
Typical Positive Displacement Pump End Configuration

1.1 Remove the upper cylinder cap nuts.

1.2 Remove the upper cylinder caps and the cap gaskets.

1.3 Remove the front cylinder cap nuts.

1.4 Remove the front cylinder caps and the gaskets.

1.5 Disconnect the plungers from the plunger adapters and remove the plungers.

1.6 Remove the packing adjusting spring.

NOTE: To use the packing removal tool, first remove the knurled nut from the tool. Slide the tool through the packing assembly, replace the knurled nut, and pull the tool forward to remove the packing. Figure 4-24 shows a list of the special tools required for packing and valve removal.

NOTE: Placing some shim stock (either a 0.003" or 0.005" piece) inside the packing spring bore in the block keeps the packing material from falling into the suction valve bore.

1.7 Remove the packing assembly.

1.8 Remove the suction and discharge valves as follows:

1.8.1 Remove the fastening hex socket cap screw with the valve fastener removing tool and 1/2" drive extension.

1.8.2 Remove the retainer, springs, and plate from the valve with the valve fastener gripping tool.

NOTE: If the valve seat is to be removed, screw the extraction stud into the tool stem, and screw the exposed end of the stud into the valve seat.

1.8.3 Assemble the pressure plate thrust washer, nut, and drive cap to the stem. Turn down on the stem to put approximately 20 ft.-lb. of torque on the valve seat.

NOTE: Either dry ice or conventional ice can be used to pack around the valve seat to assist in the removal of the seat. Wait for the ice to cool the valve before removing the seat. Waiting time is approximately 7–10 minutes for dry ice and 15–20 minutes for conventional ice.

1.8.4 Pack ice around the top of the valve seat.

1.8.5 Tap the end of the stem smartly with a small hammer. The valve should pop out. If the valve sticks, increase the torque on the nut and tap again. Repeat this process until the valve seat is removed.

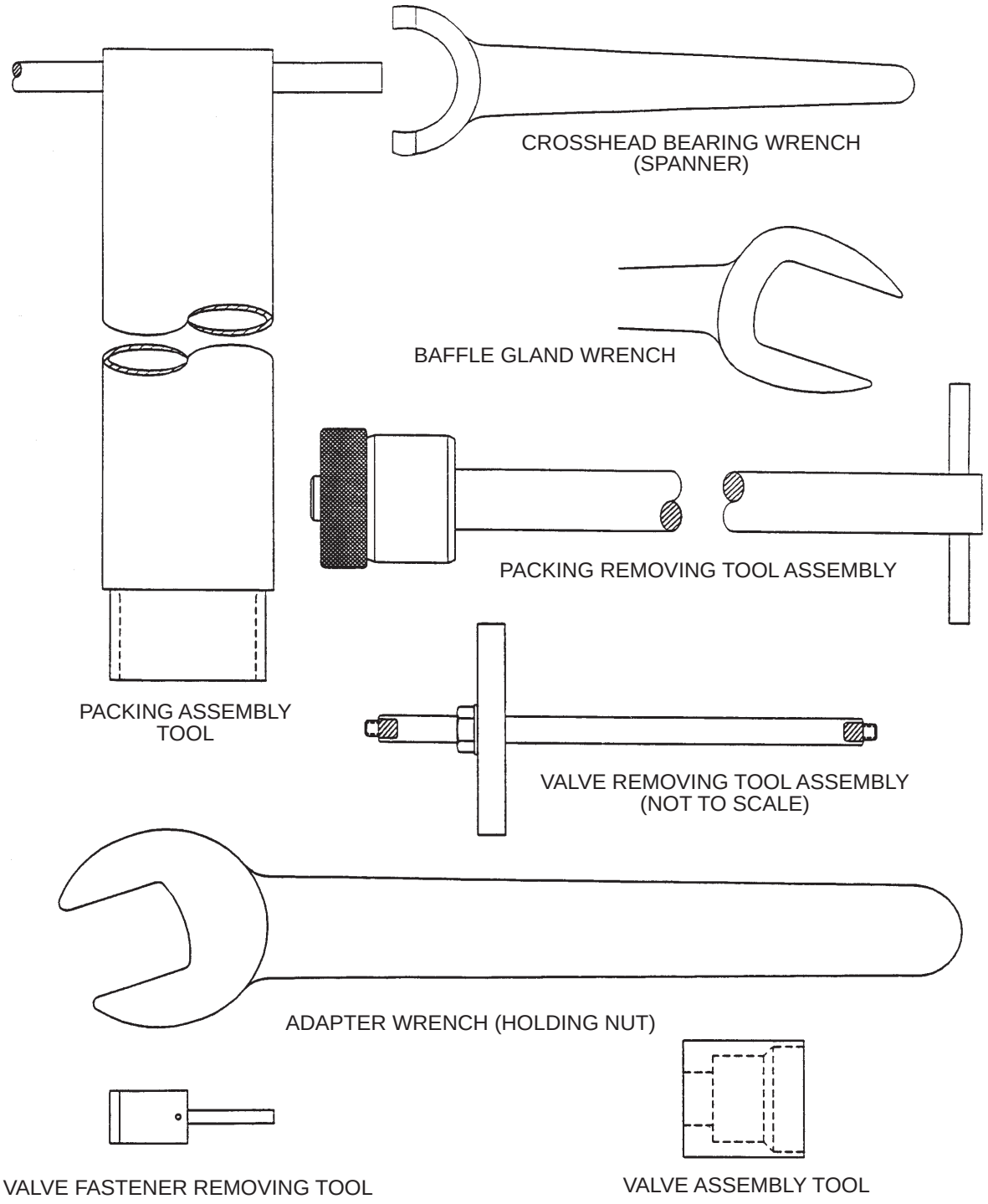


Figure 4-24
Special Tools for Valve and Packing Removal

4.4.2.3 Plunger Packing Replacement

The following tools are required to replace the packing:

- 0–1” depth micrometer
- 2–3” snap gauge
- 2–3” outside micrometer
- Dial indicator
- Neolube with applicator
- Rymple cloth
- 14” pipe wrench
- Adjusting nut spanner wrench, special tool
- 3/4” combination wrench
- 1/8” Allen wrench
- Pry bars (2)
- Packing puller, special tool
- Scribe
- Display-traverse unit
- Pick-up
- Pick-up cable
- Two pick-up holders
  - Unbiased (round)
  - Bias/reversed (square)
- Plunge cradle
- Profilometer surtronic 3P

1. Ensure that the cavity drain is not clogged.
2. Open the drain valve for the packing seal water tank and drain the tank.
3. Remove the packing (some designs use cartridges).

NOTE: The cartridges are referred to as No. 1–3, with No. 1 being closest to the motor. These numbers are stamped on the cartridges and are used as a reference point for recording the location of packing failure. Cartridges can be removed in any logical order. Steps 3.1 through 3.11 are applicable to cartridge No. 1.

3.1 Rotate the crankshaft-to-motor coupling until the plunger is at the maximum travel out of the cartridge (gland).

3.2 Separate the intermediate rod and the plunger.

3.3 Slide the plunger forward into the cartridge (gland).

3.4 Remove the bolts from the intermediate rod seal retainer flange.

3.5 Remove the intermediate rod and seal retainer from the pump as an assembly.

3.6 Using the special adjusting nut spanner wrench, remove the stuffing box adjusting nut.

3.7 Record the location (for example, 12 o'clock) of the cartridge identification number.

3.8 Matchmark the secondary packing carrier to the cartridge at the 12 o'clock position.

3.9 Remove the cartridge (packing) from the stuffing box.

3.10 Record the cartridge serial number.

3.11 Remove the cartridge seal.

3.12 Repeat steps 3.1 through 3.11 for cartridges No. 2 and 3.

4. Seal the water tank and line flush. Refer to Figure 4-25 for a schematic of a typical packing seal water system.

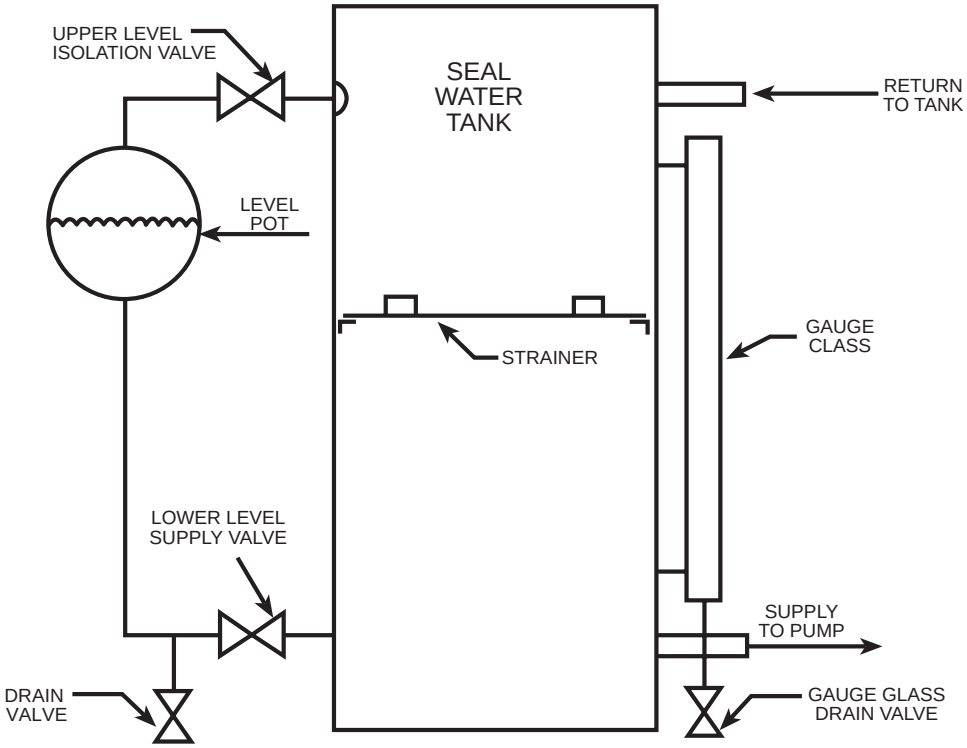


Figure 4-25
Typical Packing Seal Water System

4.1 Drain the seal water tank.

4.2 Remove the tank cover, and inspect the gasket.

4.3 Remove the strainer, and clean the strainer and the tank.

4.4 Remove and clean the overflow strainer. Inspect the strainer for damage, and repair or replace it as required.

4.5 Shut the drain valve.

4.6 Fill the seal water tank, and ensure that the tank and level pot are full.

4.7 Ensure that a full stream of water flows through each seal water supply line hole in the bottom of the stuffing boxes. If any lines are clogged, remove and unclog the lines.

4.8 Cover the holes of the seal water supply line in the bottom of the stuffing boxes with your fingers.

4.9 Ensure that a full stream of water flows through each seal water return line hole in the top of the stuffing boxes. If any lines are clogged, remove and unclog them.

4.10 Shut the seal water tank fill valve.

4.11 Open the drain valve for the seal water tank and drain the tank.

4.12 Remove the debris from the bottom of the pump cavity.

4.13 Ensure that all seal water supply and return lines are free of obstructions.

4.14 Install the strainer.

4.15 Install the overflow strainer.

4.16 Install the cover, using a new gasket if required.

5. Inspect the stuffing box.

5.1 Inspect and clean the stuffing boxes of all foreign material. The minimum surface finish for the stuffing box bore is 63 RMS, unless otherwise specified.

5.2 Inspect the cartridge seal seating surfaces for nicks and scratches that could prevent the seals from sealing.

6. Remove the packing from the cartridges.

6.1 Remove the cartridge O-ring.

6.2 Loosen the set screw and withdraw the secondary packing carrier and the plunger from the cartridge.

6.3 Remove the plunger from the secondary packing carrier.

6.4 Remove the secondary packing carefully from the secondary packing carrier.

6.5 Remove the lube ring and the lube O-ring from the cartridge.

6.6 Remove the primary packing.

6.7 Repeat steps 6.1 through 6.6 for the remaining cartridges if necessary.

7. Clean all parts that are to be reused.

8. Inspect the cartridge. Figure 4-26 shows a typical packing cartridge.

8.1 Inspect the cartridge for signs of wear.

8.2 Measure and record the inside diameter of the cartridge primary packing area. The measurement should be 2.875“ (2.872–2.875”).

8.3 Measure and record the cartridge seal depth. The measurement should be 0.173” (0.168–0.173“).

8.4 Ensure that a 15° chamfer exists on the cartridge just below the primary packing area.

8.5 Inspect the secondary packing carrier for signs of wear.

8.6 Measure and record the inside diameter of the secondary packing area.

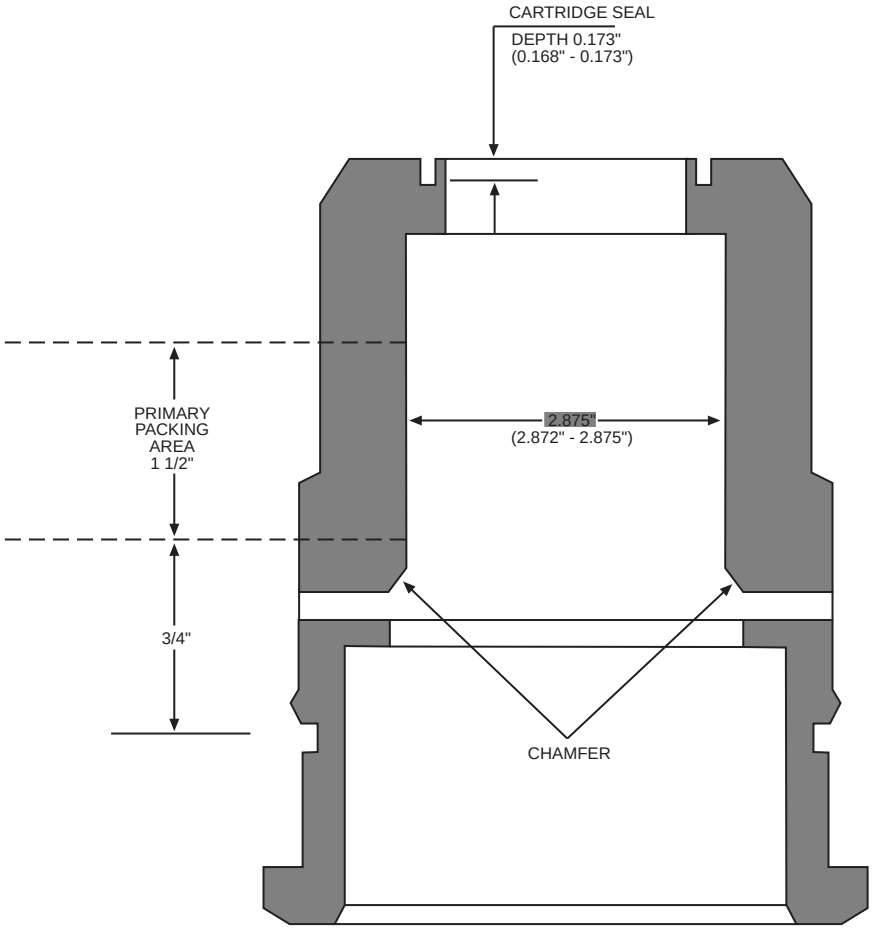


Figure 4-26
Typical Packing Cartridge

8.7 Inspect the lube ring for signs of wear.

8.8 Inspect the primary bronze top adapter for wear.

8.9 Inspect the secondary bronze top adapter for wear.

8.10 Repeat steps 8.1 through 8.9 for the remaining cartridges if necessary.

9. Inspect the plungers. Refer to Figure 4-27 for plunger inspection.

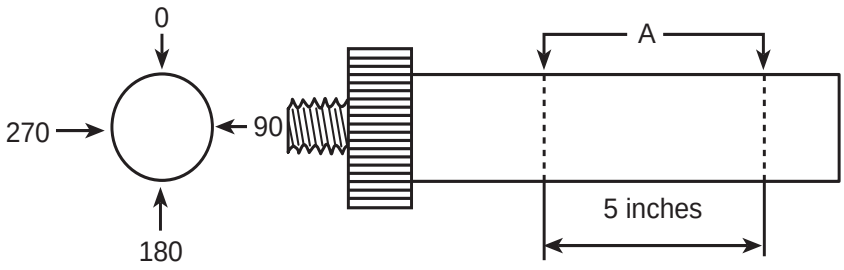


Figure 4-27
Plunger Measurement Diagram

9.1 Clean and inspect the plungers for signs of wear.

9.2 Place the plunger in the plunger cradle so that the knurled end of the plunger is hanging off the end and the plunger is flat in the bottom.

9.3 Place the profilometer feet in the sliding grooves next to the plunger.

9.4 Install the pick-up, pick-up cable, and pick-up holder on the display traverse unit.

9.5 Set the display traverser unit as follows:

9.5.1 The range switch should be down to 9999 micro-inch.

9.5.2 The parameter switch should be set on Ra.

9.5.3 The cut-off switch should be set on 0.8 mm / 0.03 in.

9.6 Ensure that the pick-up is in the 5” stroke area of the plunger and is supported by the skid, not the stylus.

9.7 Ensure that the end of the pick-up marked 03° is square with the plunger.

9.8 Inspect the surface with the profilometer, and verify that the finish is equal to or less than 16 RMS.

9.9 Obtain at least two profilometer readings axially in the stroke area at the 0°, 90°, 180°, and 270° positions.

9.10 Verify that none of the readings exceed 16 RMS.

10. Install the packing in the cartridge.

10.1 Mark a line on one side of the flat face of the top adapter.

10.2 Lubricate the primary packing by dipping it in water and install it. Ensure that the V-ring lips are pointed toward the bottom of the cartridge and that the line on the top adapter is in line with the cartridge number in the 12 o'clock position.

10.3 Install the lube ring into the cartridge.

10.4 Lubricate the lube ring O-ring with an approved lubricant.

10.5 Install the lube ring O-ring in the groove on the upper edge of the lube ring.

10.6 Lay the secondary packing carrier on a suitable flat surface with the larger opening facing up and the smooth flange facing down.

10.7 Lubricate the secondary packing by dipping it in water and install it. Ensure that the V-ring lips are pointed toward the bottom of the cartridge and that the line on the top adapter is in line with the cartridge number in the 12 o'clock position.

NOTE: Lubricating the plunger with water before installation eases the assembly process.

10.8 Insert the plunger through the secondary packing carrier until the shoulder contacts the secondary packing smooth flange. Ensure that the packing is not pushed out.

NOTE: It might be necessary to use an arbor press to press the secondary packing carrier into the cartridge.

10.9 Install the plunger and the secondary packing carrier assembly carefully into the cartridge and through the primary packing until the set screw can engage the slot in the secondary packing carrier.

NOTE: The set screw should not be bottomed out against the secondary packing carrier.

10.10 Install the set screw and adjust it so that it is flush with the outside diameter of the cartridge.

10.11 Position the plunger in the cartridge until about 6" protrude from the intermediate rod end.

NOTE: Applying a light coat of lubricant on the cartridge seal in four places helps to hold the cartridge seal in place while installing it in the stuffing box.

10.12 If the cartridge seal is to be installed in the pump at this time, lubricate the cartridge seal and the cartridge O-ring with an approved lubricant.

10.13 Install the cartridge O-ring around the groove in the cartridge.

10.14 Install the cartridge seal on the bottom of the cartridge.

10.15 Repeat steps 10.1 through 10.14 for the remaining cartridges if necessary.

11. Install the cartridge in the stuffing box.

NOTE: If previously rebuilt spares are being used, cartridges must be submersed in demineralized water for several minutes to ensure that the packing is not dry during installation and startup.

11.1 Verify that the proper cartridge is being installed.

11.2 Insert the cartridge carefully into the stuffing box, ensuring that the cartridge seal is not disturbed and that the cartridge number is on top (12 o'clock).

11.3 Lubricate the stuffing box threads and adjusting nut threads with Neolube.

11.4 Install the adjusting nut.

11.5 Inspect the rubber baffle plate on the intermediate rod for damage or wear, and replace it if it is damaged or worn.

11.6 Inspect the mating surfaces of the plunger, intermediate rod, and crosshead for nicks, burrs, or high spots.

11.7 Install the intermediate rod and the rod seal retainer flange together.

11.8 Bolt the intermediate rod seal retainer flange to the pump, ensuring that the drain hole is down.

11.9 Screw the intermediate rod into the crosshead and tighten, using a pipe wrench.

11.10 Tighten the plunger into the intermediate rod, using the pipe wrench.

11.11 Ensure that the stuffing box adjusting nut is still tight.

11.12 Install the remaining cartridges if necessary.

4.4.2.4 Charging Pump Suction and Discharge Valve Replacement

The following tools are required to replace the suction and discharge valves:

- Torque wrench, 0-250 ft.-lb. with multiplier
- Torque wrench, 0-600 in.-lb.
- Neolube
- Special valve removal tool
- Hydraulic jack with spacer, for use on stuck valves
- 5-pound maul
- Special seating tool with nylon insert, for seating valves

- Acid brush
- Slide hammer

1. Remove the suction and discharge valves.

   1.1 Remove the valve cover nuts and the cover.

   1.2 Remove the following from the discharge valve: valve hex nut, valve guard or retainer, spring, valve disc, and sleeve.

   1.3 Bag and tag all parts removed from the pump cavity.

   1.4 Screw the special valve-pulling tool onto the discharge valve stud and tighten firmly. Ensure that the tool has full engagement with the stud.

NOTE: Step 1.5 should be used to remove the valve with a slide hammer. If the valve does not come out with the slide hammer, steps 1.6 and 1.7 or 1.8 should be performed.

   1.5 Remove the discharge valve with the slide hammer. Unscrew the special valve-pulling tool from the discharge valve stud, and remove the discharge valve seat.

WARNING: *Ensure that all personnel are clear of the valve while the hydraulic jack is in use to avoid injury if the valve breaks free.*

   1.6 Apply tension to the stud, using a hydraulic jack.

   1.7 If the discharge valve seat breaks free, unscrew the special valve-pulling tool from the discharge valve stud, and remove the discharge valve seat.

   1.8 If the discharge valve seat does not break free, continue applying tension with the hydraulic jack while rapping simultaneously on top of the tool with a maul to loosen the discharge valve seat.

   1.9 When the discharge valve seat breaks free, unscrew the special valve-pulling tool from the discharge valve stud, and remove the discharge valve seat.

   1.10 Repeat steps 1.1 through 1.9 for the remaining discharge valves and all suction valves if applicable.

2. Clean and inspect the suction and discharge valves.

   2.1 Clean the valve bore / seat of the suction and discharge valves.

   2.2 Perform dye penetrant examination of the valve bores.

3. Install the suction and discharge valves.

   3.1 Ensure that the suction and discharge valve bores / seats are clean.

   3.2 Install a new suction valve assembly into the suction valve bore / seat.

   3.3 Center the nylon inserted end of the seating tool onto the suction valve stud, and seat the valve into the bore / seat, using a maul as necessary.

3.4 Loosen the suction valve hex nut, and re-torque to 25 ft.-lb. (20–25 ft.-lb.).

3.5 Repeat steps 3.1 through 3.4 for the remaining suction valves and all discharge valves if applicable.

3.6 Ensure that the valve cover and the cylinder seating surfaces are clean.

3.7 Apply two coats of Neolube to cover the studs.

3.8 Install the valve cover and the new cover gasket.

3.9 Install the cover nuts and torque to 550 ft.-lb. (520–580 ft.-lb.) in three increments (180, 360, and 550 ft.-lb.).

3.10 Repeat the final pass until no further movement is evident.

4.4.2.5 Discharge Pulsation Dampener Overhaul

The following tools are required to overhaul the discharge pulsation dampener:

- Leak tec
- Rymple cloth
- Neolube
- Charging pump tool box and key
- Bladder puller
- Special spanner
- Valve core removal tool
- Chainfall (1/2-ton minimum)
- 6-foot strap (2)
- 4-foot strap

1. Remove the pulsation dampener. Refer to Figure 4-28 for the discharge pulsation dampener details and Figure 4-29 for details on the discharge dampener seal locations.

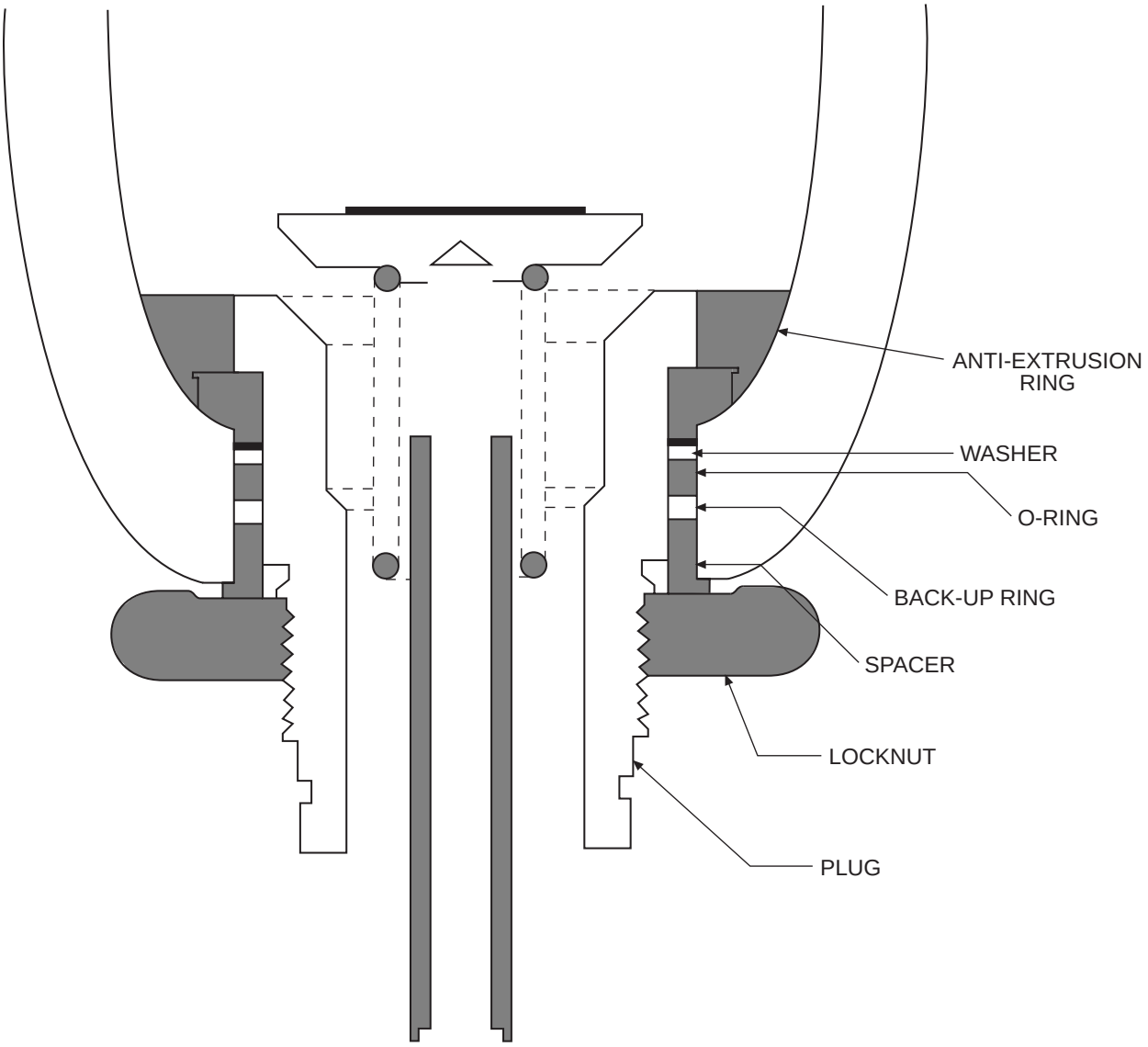


Figure 4-28
Discharge Dampener Seals

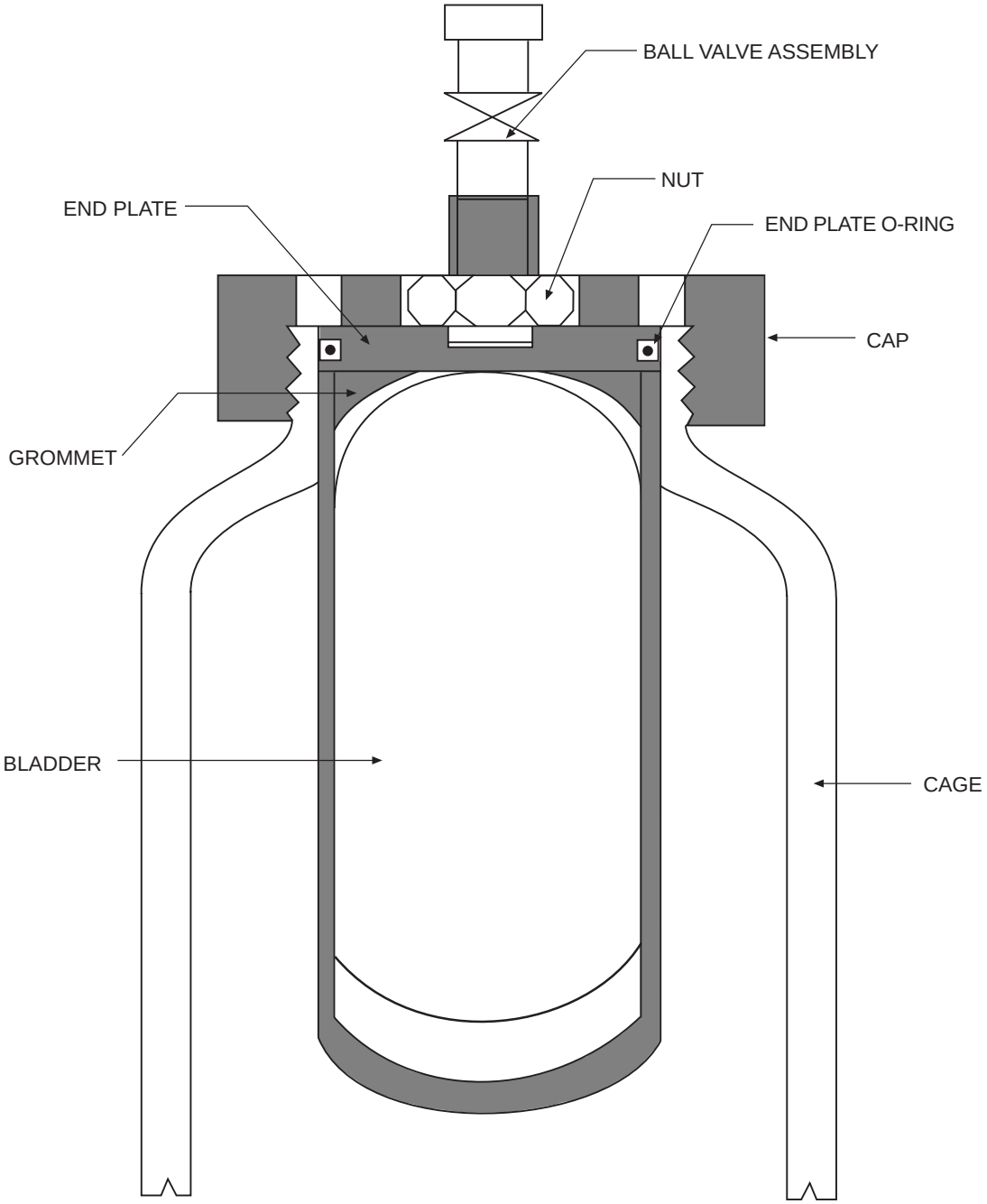


Figure 4-29
Discharge Dampener Seal Locations

1.1 Install the quick check tool, and open the ball valve.

1.2 Open the discharge pulsation dampener ball valve to bleed the residual nitrogen pressure from the bladder.

1.3 Remove the quick check tool.

1.4 Remove the ball valve assembly with the O-ring.

1.5 Remove the cap.

1.6 Remove the O-ring and the back-up ring.

NOTE: The shell assembly can be removed by using a chainfall rigged to an overhead lift point. When the shell assembly is removed from the flange assembly, the locating pin may drop. Care must be taken to prevent the pin from dropping into the system.

1.7 Rig to lift the shell assembly with a chainfall and lift the shell assembly clear of the flange assembly.

1.8 Place the shell assembly on a suitable flat surface.

2. Disassemble the discharge dampener.

2.1 Remove the O-rings, back-up rings, locknut, and spacer.

2.2 Push the plug and poppet assembly gently into the shell.

2.3 Remove the back-up ring, O-ring, and washer.

NOTE: The anti-extrusion ring must be bent at the middle and held at an angle to clear the opening of the shell. The anti-extrusion ring can be manipulated by inserting your hand into the shell.

2.4 Remove the anti-extrusion ring.

NOTE: Do not allow the weight of the poppet assembly to strike the face of the shell assembly opening.

2.5 Remove the plug and poppet assembly from the shell.

2.6 Remove the nut from the valve stem.

2.7 Pull the bladder from the shell.

2.8 Inspect the bladder, O-rings, back-up rings, and extrusion ring for deterioration, wear, tears, or other damage, and replace as required.

2.9 Inspect the shell for internal cleanliness and damage.

3. Assemble the pulsation dampener.

3.1 If a new bladder is used, remove the valve core from the new bladder and discard it.

3.2 Squeeze the bladder to remove as much air as possible.

3.3 Apply a light coat of an approved lubricant to the shell assembly opening.

NOTE: A threaded rod can be used with the bladder puller to aid in inserting the bladder.

3.4 Install the bladder into the shell and guide the bladder valve stem through the small opening in the top of the shell.

NOTE: Excessive force applied to the stem nut could twist or damage the valve stem.

3.5 Install the stem nut and tighten it.

3.6 Install the plug and the poppet assembly into the shell.

NOTE: The anti-extrusion ring must be bent at the middle and held at an angle to clear the opening of the shell. The anti-extrusion ring can be manipulated by inserting your hand into the shell.

3.7 Install the anti-extrusion ring in the shell against the shoulder on the plug and poppet assembly.

3.8 Apply a light coat of an approved lubricant on the shell opening.

3.9 Pull the plug and poppet assembly with the anti-extrusion ring into position against the shell opening.

3.10 Install the washer, O-ring, back-up ring, spacer, and locknut.

3.11 Apply a light coat of an approved lubricant to the O-ring grooves on the plug and poppet assembly.

3.12 Install the O-ring and the back-up rings.

4. Install the pulsation dampener.

4.1 Lift the shell assembly with a chainfall into position over the flange assembly.

NOTE: The locating pin holes should be aligned before setting the shell into the flange assembly. After the shell with its O-rings is inserted into the flange, it is difficult to rotate.

4.2 Lower the shell assembly into the flange assembly as far as possible.

4.3 Apply a light coat of an approved lubricant to the inside diameter of the plug for the O-ring seat.

4.4 Apply a light coat of an approved lubricant to the O-ring grooves in the cap.

4.5 Screw the cap into the plug to seat the flange.

4.6 Remove the cap.

4.7 Install the locating pin.

4.8 Install the O-ring and back-up ring.

4.9 Lubricate the cap threads with Neolube; install and tighten the cap.

4.10 Install the ball valve assembly with the new O-ring.

4.11 Precharge the bladder.

4.11.1 Turn the handle on the air chuck counterclockwise until it stops.

4.11.2 Mount the air chuck on the valve stem and tighten it.

4.11.3 Verify that the nitrogen fill valve is shut.

4.11.4 Verify that the valve between the test gauge and accumulator is open.

4.11.5 Turn the handle on the air chuck clockwise until the shaft depresses the valve core.

4.11.6 Mount the charging assembly regulator to the nitrogen bottle.

4.11.7 Verify that the regulator is backed out.

4.11.8 Open the valve on the nitrogen bottle.

4.11.9 Open the valve on the fill line of the charging assembly.

CAUTION: *The pressure must be adjusted slowly to prevent rupturing the bladder. For low-pressure operations, the bladder should be precharged to 200 psi (190–210 psi). For high-pressure operations, the bladder should be precharged to 1380 psi (1370–1390 psi).*

4.11.10 Slowly adjust the regulator valve to obtain the desired pressure.

4.11.11 Shut the valves between the test gauge and the accumulator and the nitrogen bottle and accumulator.

4.11.12 Bleed the pressure from the charging assembly with the bleeder valve.

4.11.13 Turn the handle on the air chuck counterclockwise, and remove the air chuck.

4.11.14 Remove the charging assembly regulator from the nitrogen bottle.

NOTE: If the valve core is leaking, depress the valve core, and quickly release it until the valve reseats. This sometimes stops the leak.

4.12 When the bladder has been precharged, perform a leakage inspection (soap test) on the ball valve assembly. No leakage is allowed.

4.13 Tighten the gas valve cap hand tight.

4.14 Install the dampener valve guard hand tight.

4.4.2.6 Suction Stabilizer Overhaul

The following tools are required to overhaul the suction stabilizer:

- Leak Tec
- Neolube
- Rymple cloth
- Charging pump toolbox and key
- Bladder puller
- Special spanner
- Valve core removal tool

1. Disassemble and inspect the bladder, using Figure 4-29 as a guide.
    1.1 Install the quick check tool
    1.2 Open the quick check vent valve.
    1.3 Open the suction stabilizer ball valve to remove any residual nitrogen.
    1.4 Remove the quick check tool.

NOTE: When pulling the bladder assembly out, some resistance can be expected as a result of the O-ring fit between the end plate and the housing. A reasonable amount of force can be used.

    1.5 Remove the ball valve assembly.
    1.6 Remove the machine cap
    1.7 Lift the end plate, with the bladder assembly attached, out of the shell assembly.
    1.8 Remove the nut, end plate, grommet, and end plate O-ring from the bladder assembly.
    1.9 Install the ball valve assembly into the bladder, and inflate to 3 psig with dry nitrogen.
    1.10 Inspect the bladder and the valve assembly for leakage (soap test).
    1.11 Deflate the bladder, and remove the ball valve assembly.
    1.12 Replace the ball valve assembly and / or the bladder if they showed any signs of leakage in step 1.10.
    1.13 Inspect the bladder, end plate O-ring, ball valve assembly, grommet, end plate, machine cap, and nut, and replace them if necessary.
2. Assemble the bladder.
    2.1 If a new bladder is used, remove the valve core from the new bladder and discard it.

2.2 Install the special bladder puller into the valve stem.

2.3 Lubricate the valve stem assembly with an approved lubricant.

2.4 Lubricate the end plate O-ring with an approved lubricant, and install it into the groove on the end plate.

2.5 Place the grommet behind the end plate with the curved surface facing down.

2.6 While holding the grommet in place, insert the bladder puller through the center hole of the end plate, and pull the stem through.

2.7 Install the nut over the valve stem and tighten snug.

2.8 Remove the special bladder puller.

2.9 Install the bladder and end plate assembly into the cage, ensuring that the grommet enters the cage and that the end plate O-ring remains in place.

2.10 While holding the valve stem, tighten the nut.

2.11 Lubricate the cap and the housing threads with Neolube.

2.12 Install the ball valve assembly with the new O-ring.

2.13 Precharge the bladder with a charging rig similar to that shown in Figure 4-30.

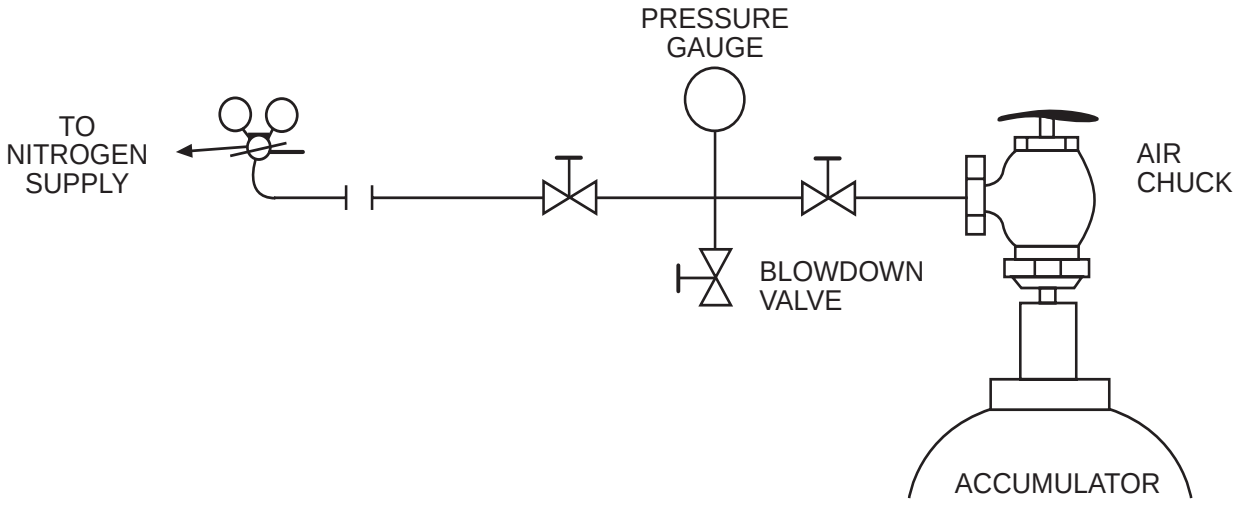


Figure 4-30
Charging Rig

2.13.1 Turn the handle on the air chuck all the way counterclockwise.

2.13.2 Mount the air chuck on the valve stem and tighten it.

2.13.3 Verify that the nitrogen fill valve is shut.

2.13.4 Verify that the valve between the test gauge and accumulator is open.

2.13.5 Turn the handle on the air chuck clockwise until the shaft depresses the valve core.

2.13.6 Mount the charging assembly regulator to the nitrogen bottle.

2.13.7 Verify that the regulator is backed out.

2.13.8 Open the valve on the nitrogen bottle.

2.13.9 Open the valve on the fill line of the charging assembly.

CAUTION: *The pressure must be adjusted slowly to prevent rupturing the bladder. Depending upon the installation, proper precharge pressure should be somewhere between 15–30 psi.*

2.13.10 Slowly adjust the regulator valve to obtain the desired pressure.

2.13.11 Shut the valves between the test gauge and the accumulator and the nitrogen bottle and accumulator.

2.13.12 Bleed the pressure from the charging assembly with the bleeder valve.

2.13.13 Turn the handle on the air chuck counterclockwise, and remove the air chuck.

2.13.14 Remove the charging assembly regulator from the nitrogen bottle.

2.14 When the bladder has been precharged, inspect the ball valve assembly for leakage (soap test). No leakage is allowed.

2.15 Tighten the gas valve cap hand tight.

2.16 Install the valve guard hand tight.

4.4.2.7 Connecting Rod Bearings Removal and Replacement

NOTE: Connecting rod subassemblies consist of two halves of the connecting rod, a set of two steel-backed, babbitted bearing halves, two bolts with locknuts, a crosshead bearing retainer nut, and a ball. Parts from one connecting rod must ***not*** *be used on another connecting rod.*

1. Remove the base cover screws and the base cover.
2. Remove the connecting rod nuts and bolts.
3. Remove the connecting rod cap, and pry the bearing half from it, being careful not to scratch the journal.

4. Slide the connecting rod to the front of the machine, and remove the other bearing half.
5. Inspect the bearing surface on the eccentric shaft for roughness or scratches, and smooth as necessary.

*NOTE: Bearing halves **must** be used as a set, but they can be interchanged on the same connecting rod. Bearings are automatically located by dowel pins. Ensure that dowels are bottomed in the holes provided so that the bearings are fully seated in connecting rod parts.*

6. Install the new bearing halves if required.
7. Lubricate the rod journal and the associated connecting rod with oil.
8. Pull the connecting rod into place on the shaft.
9. Lubricate the connecting rod cap.
10. Install the connecting rod cap.
11. Install the two connecting rod bolts and snug the nuts. Torque the connecting rod bolts to the proper torque.
12. Repeat steps 1 through 11 for the other connecting rods if necessary.
13. Clean the mating surfaces on the base cover, and install a new gasket if necessary.
14. Install the base cover.
15. Install the base cover screws and tighten them evenly.
16. Ensure that the oil is at the correct level.

4.4.2.8 Plunger Adapter Removal and Replacement

1. Remove the base cover screws and the base cover.
2. Remove the well cover nuts and the well cover.
3. Remove the crosshead oil lines.
4. Remove the crosshead cover cap screws and the crosshead guide cover.
5. Rotate the crosshead as necessary, and loosen both plunger adapter set screws.
6. Rotate the shaft until the crosshead separates from the plunger adapter.
7. Carefully lift the connecting rod and the crosshead out of the way.
8. Loosen the packing adjusting screw.
9. Unscrew the plunger adapter from the plunger.
10. Remove the plunger adapter by sliding it through the baffle stuffing box.
11. Remove the packing adjusting screw.

12. Remove the baffle wiper washer and the baffle seal packing.
13. Install the new plunger adapter by sliding it into position through the baffle stuffing box.
14. Lower the connecting rod and the crosshead into place.
15. Rotate the shaft until the plunger adapter engages the crosshead and bottoms out.
16. Tighten both plunger adapter set screws.
17. Repeat steps 1 through 16 for the other two plunger adapters if necessary.
18. Install the crosshead guide cover and the crosshead cover cap screws.
19. Install the oil lines on the crosshead cover.
20. Install the base cover gasket.
21. Install the base cover.
22. Install the base cover screws and tighten them evenly.
23. Clean the baffle stuffing box, plunger adapter, and packing adjusting screw, using Figure 4-31 as a guide.

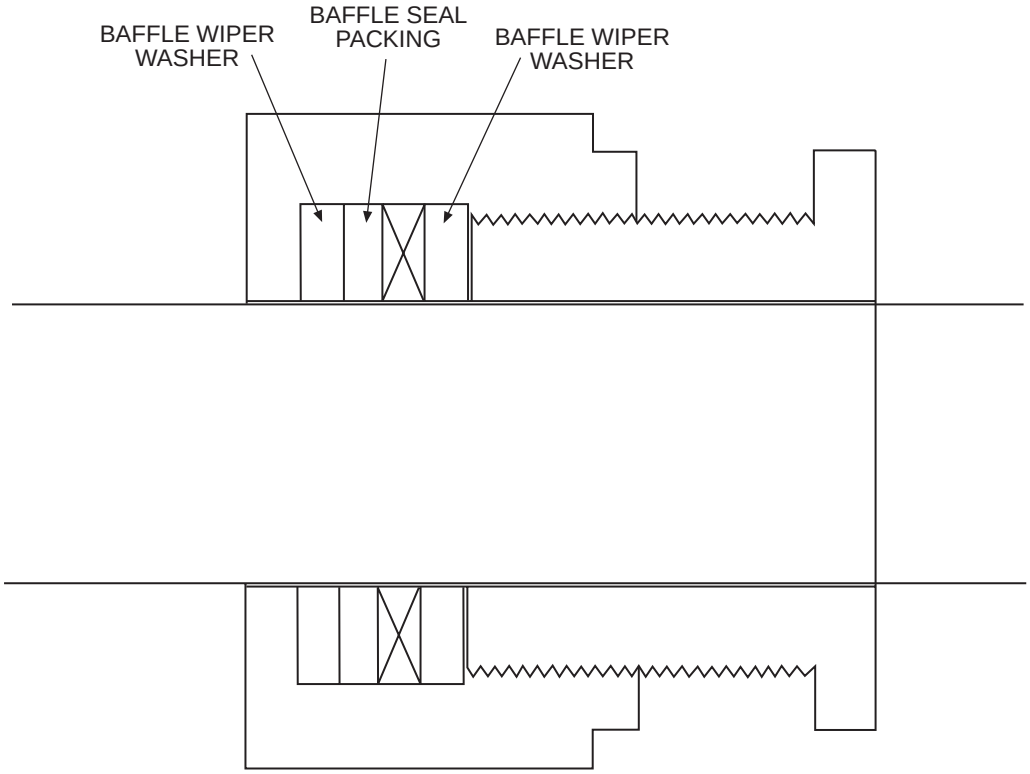


Figure 4-31
Typical Baffle Packing Arrangement

24. Install the new baffle packing in the following order: one Teflon ring, two braided rings, and one Teflon ring.
25. Install the packing adjusting screw, but do **not** tighten it.
26. Connect the plunger adapter to the plunger and tighten it.
27 Perform steps 23 through 26 for the remaining baffles if necessary.
28. Ensure that the oil is at the proper level.
29. Run the charging pump, and adjust the packing as necessary.
30. Install the well cover gasket.
31. Inspect the well, and ensure that it is free of foreign material.
32. Install the well cover and the well cover nuts. Tighten the well cover nuts evenly.

4.4.2.9 Fluid Drive Disassembly

1. Drain the oil.
2. Disconnect the oil cooler piping.
3. Disconnect the driving and driven equipment from the fluid drive.
4. Disconnect the speed controller connections from the fluid drive speed control.
5. Remove the fluid drive from the sub-base.
6. Remove the flexible coupling parts and keys from both shafts.
7. Disassemble the output end.
   - 7.1 Remove the cap screws and the output shaft seal end cover.
   - 7.2 Pull out the dust seal by loosening the set screw.
   - 7.3 Remove the bearing retaining ring, and unscrew the elastic stop nuts from the bearing retainer.

NOTE: To remove the output bearing retainer and output outboard bearing, use the jack screws in the tapped holes provided in the bearing retainer bolting flange. Also support a puller to the bearing retainer shaft seal bolting flange holes. Simultaneously apply a pulling force to the bearing retainer from the jacking screws and puller.

   - 7.4 Remove the output bearing retainer and the output outboard bearing.
   - 7.5 Move the speed control rod to the maximum output speed position.
   - 7.6 Disconnect the control rod bracket and scoop guide by removing the cap screws.
   - 7.7 Remove the scoop guide and the scoop tube by tipping the scoop tube assembly until the scoop tube clears the runner casing.

NOTE: The control rod bracket drops off the control rod when the scoop guide is removed.

8. Disassemble the input end.

   8.1 Loosen the access cover cap screws, and remove the access cover.

   8.2 Remove the connecting link from the pump chain assembly, and remove the chain assembly.

   8.3 Loosen the pump sprocket set screw, and remove the pump sprocket.

   8.4 Loosen the cap screws from the pump housing, and pull the gear pump from the input end bell.

   8.5 Place the fluid drive housing on end by fastening the screw plate clamps to the housing input end**.** Do **not** allow the output shaft to contact the floor**.**

   8.6 Remove the cap screws from the input end bell bolting flange, and place the jacking screws in the tapped holes provided.

   8.7 Tighten the jacking screws to separate the input end bell from the housing.

*NOTE: Before lifting the input end bell and rotating assembly out of the housing, remove the bolts from the end bell that lock the end bell and housing together. A bench-type structure must be provided for supporting the rotating assembly during the remainder of the disassembly. The bench top must have a hole to allow the rotating assembly to rest on the bench***.**

   8.8 Thread an eyebolt to the coupling end of the input shaft.

   8.9 Disconnect the pump-to-housing coupling.

   8.10 Connect the lifting hoist to the eyebolt, and lift the input end and rotating assembly vertically out of the housing. With the shafts in the vertical position, lower the output end onto the bench.

NOTE: To prevent the output rotating assembly from falling onto the inner vane plate during disassembly, block up the output shaft end with wooden blocks.

   8.11 Remove the input shaft seal end cover by removing the cap screws.

   8.12 Loosen the set screw, and remove the shaft dust seal.

   8.13 Remove the bearing retaining ring.

NOTE: Before removing the end bell, remove the drive chain. Failure to do this will cause serious damage to the pump sprocket or drive sprocket.

   8.14 Support a puller from the input outboard shaft end, and secure the other end to the input end bell shaft seal bolting flange.

8.15 Remove the input end bell and the input outboard bearing.

9. Disassemble the rotating members

NOTE: Before disassembling any rotating members, be sure to locate the balancing match marks. The cap screws must be reinserted in their original locations to preserve the dynamic balance of the rotating assembly.

9.1 Remove the impeller pump by removing the cap screws. After the impeller pump is removed, reinsert the cap screws in their original locations.

9.2 Loosen the set screw, and remove the pump drive sprocket.

9.3 Use the bearing puller to remove the input inboard bearing.

9.4 Remove the cap screws from the input shaft flange, and insert the jacking screws in the tapped holes.

9.5 Remove the input shaft from the impeller casing.

9.6 Remove the bearing retainer plate by removing the cap screw.

9.7 Remove the cap screws from the impeller casing.

9.8 Separate the impeller casing from the runner casing by using the jacking screws in the tapped holes.

9.9 Attach the puller to the back wall of the impeller casing, and pull the impeller casing and output inboard bearing. The impeller casing can be removed from the impeller, if necessary, by removing the cap screws.

9.10 Thread an eyebolt to the input end of the output shaft.

9.11 Connect a lifting hoist to the eyebolt, and lift the output rotating assembly vertically.

9.12 Remove the runner from the output shaft, if necessary, by removing the cap screws.

10. Disassemble the control rod

10.1 Remove the speed control rod, lever, or crank assembly.

10.2 Remove one control rod stop by removing the roll pin and unthreading the top.

10.3 Remove the alignment ring roll pin.

10.4 Remove the control rod, making sure to catch the alignment ring, which will drop off the control rod.

10.5 Remove the output end bell from the housing.

10.6 Remove the springless closure.

10.7 Press out the control rod bushings only if a replacement is necessary.

4.4.3 Repair

4.4.3.1 Crosshead Ball and Socket Adjustment

Refer to Figure 4-32 while following this procedure.

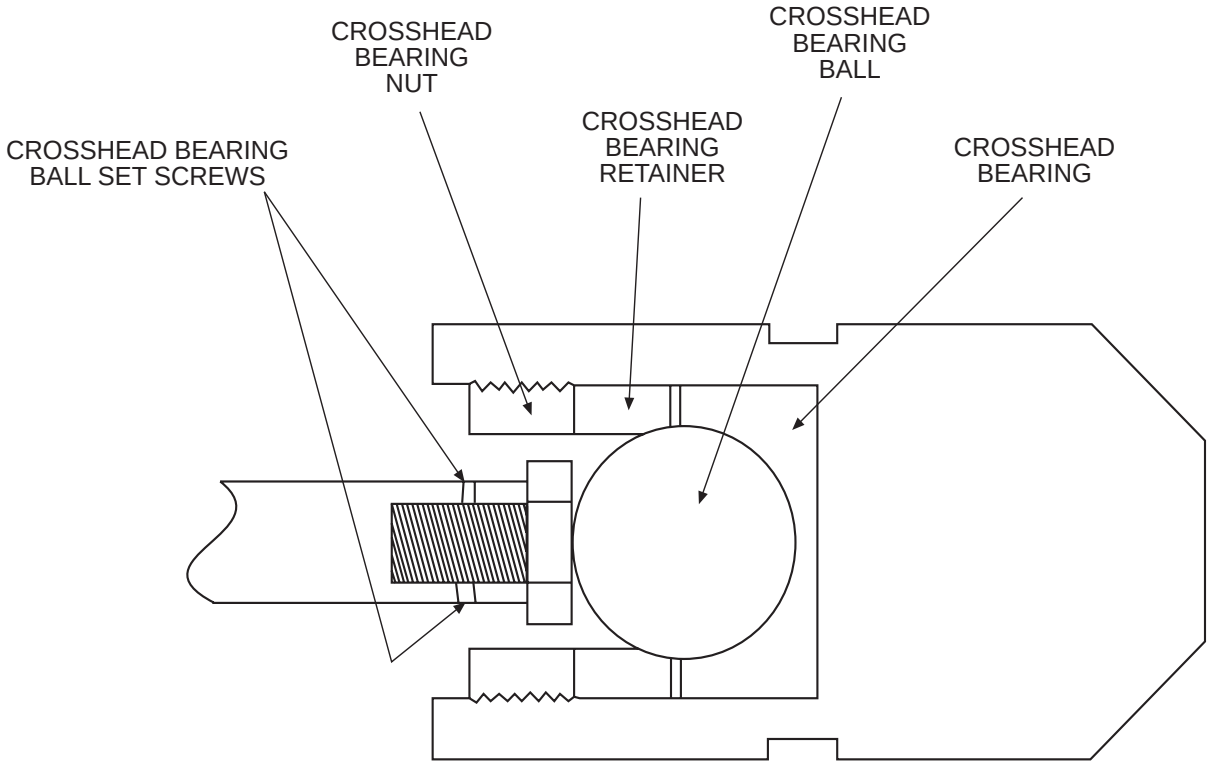


Figure 4-32
Crosshead Ball and Socket

1. Remove the base cover screws.
2. Remove the base cover.
3. Remove the crosshead oil lines.
4. Remove the crosshead cover cap screws and the crosshead guide cover.
5. Rotate the crosshead to expose the crosshead bearing set screw by turning the plunger adapter.
6. Loosen the crosshead bearing set screw.
7. Hold the crosshead bearing retainer nut with a spanner wrench while slowly turning the plunger adapter until a slight drag is noticed.
8. Tighten the crosshead bearing set screw.

9. Perform steps 1 through 8 for the remaining crossheads if necessary.
10. Install the crosshead guide cover and the crosshead cover cap screws.
11. Install the oil lines on the crosshead cover.
12. Install the base cover gasket.
13. Install the base cover.
14. Install the base cover screws and tighten them evenly.
15. Ensure that the oil is at the correct level.

4.4.4 Reassembly

4.4.4.1 Pump End Reassembly

The following tools are required to reassemble the pump end and the power end:

- Hoist, 2-ton capacity
- Lifting straps
- Socket set, 3/4" drive
- Socket drive handles, 1/2" and 3/4"
- Socket drive extension, 1/2"
- Valve fastener gripping tool
- Adjustable wrench
- Packing removal tool
- Crosshead bearing spanner wrench
- Baffle gland wrench
- Valve removal tool assembly
- Valve fastener removing tool
- Allen wrench, set
- Acetylene torch, rosebud tip (#10)
- Gloves, heatproof

1. Clean the inner block, making sure that the valve tapers are clean and dry.
2. Install the suction valves and tap them into place.

NOTE: Using the valve installation tool ensures that the entire seating area has equal force applied to prevent damaging the seat. If this tool is not used, ensure that the force applied covers the entire seating diameter, not just the center of the valve seat.

2.1 Align the matching tapers, and drive the valve seat with the valve installation tool.

2.2 If a new cap screw is being used, trim the thread-lock flush with the top of the thread.

2.3 Install the valve plate, springs, retainer, and hex socket cap screw. Ensure that all parts are concentric with the seat.

2.4 Install the remaining suction valves by repeating steps 2.1 through 2.3.

3. Install the discharge valves, and tap them into place.

NOTE: Using the valve installation tool ensures that the entire seating area has equal force applied to prevent damaging the seat. If this tool is not used, ensure that the force applied covers the entire seating diameter, not just the center of the valve seat.

3.1 Align the matching tapers, and drive the valve seat with the valve installation tool.

3.2 If a new cap screw is being used, trim the thread lock flush with the top of the thread.

3.3 Install the valve plate, springs, retainer, and hex socket cap screw. Ensure that all parts are concentric with the seat.

3.4 Install the remaining discharge valves by repeating steps 3.1 through 3.3.

4. Clean all fits inside the cylinder bore, and ensure that the bore and seating surfaces that come in contact with the adjusting rings and plunger rings are free of burrs and raised areas.

5. Inspect the square shoulders at the threaded end of the plungers and the threaded end of the plunger adapters for raised areas that could cause plunger misalignment.

6. Assemble as a unit the secondary plunger ring, secondary packing, and secondary packing adjusting ring onto the plunger.

7. Slide the front end of the plunger through the front of the cylinder, starting from the packing adjusting ring side.

8. Attach the plunger to the plunger adapter.

9. Use the packing assembly tool to compress the packing.

10. Install the secondary packing spring over the plunger and against the secondary packing assembly.

11. Assemble the primary packing plunger ring, primary packing, and the primary packing adjusting ring onto the plunger.

12. Using the packing assembly tool, compress the primary packing.

13. Install the primary packing spring over the plunger and into the packing bore.

14. Install the front cylinder cap with the gasket over the front cylinder cap studs.
15. Install the front cylinder cap nuts and tighten them evenly.
16. Repeat steps 6 through 15 for the remaining plungers.
17. Attach the upper cylinder cap gaskets onto the upper cylinder caps.
18. Install the upper cylinder caps.
19. Install the upper cylinder cap nuts and tighten them evenly.

4.4.4.2 Pump Power End Reassembly

1. Install the shaft by tipping it to an angle for ease of installation.
2. Apply a thin film of lubricating oil to all bearing surfaces.
3. Center the shaft, and slide the eccentric bearing retainers over the shaft and into the casing. Ensure that the original thrust washers are correctly oriented.
4. Install the bearing retainer cap screws, and torque to the appropriate value.
5. If removed, install the oil pump coupling by tightening the oil pump coupling set screw.
6. Install the oil pump.
   - 6.1 Remove the old gasket, and clean the mating surfaces.
   - 6.2 Position the oil pump, using a new gasket.
   - 6.3 Install the bolts and lockwashers
   - 6.4 Torque the oil pump mounting bolts to the appropriate torque.
7. Install the external oil lines.

NOTE: Typically, the oil filter is a cartridge-type oil filter, and there is an O-ring in an O-ring groove in the pump. This O-ring must be replaced every time the oil filter cartridge is replaced.

8. Replace the oil filter and seals.
9. Rotate the shaft until one rod journal is positioned as far as possible to the rear.
10. Lubricate this rod journal and the associated connecting rod with oil.
11. Pull the connecting rod into place on the shaft.
12. Lubricate the connecting rod cap with oil.
13. Install the connecting rod cap.
14. Install the connecting rod bolts, and snug the nuts.
15. Repeat steps 9 through 14 for the other two connecting rod assemblies.
16. Torque all connecting rod bolts to the appropriate torque.

17. Slide crosshead assemblies toward the crosshead bearing ball bolt, and engage the ball into the crosshead socket.
18. Screw the crosshead bearing retainers into the crossheads.
19. Adjust the crosshead bearings.
    - 19.1 Hold the bearing retainer nut with a spanner wrench while slowly turning the plunger adapter with its wrench. Tighten it until a slight drag is noticed on the plunger adapter.
    - 19.2 Tighten the set screw.
    - 19.3 Repeat steps 19.1 and 19.2 for the other two crossheads.
20. Install the crosshead cover and the cap screw. Torque to the appropriate torque.
21. Install the crosshead cover oil tubing.
22. Install the shaft key.
23. Install the coupling hub, ensuring that it is aligned with the match mark made during disassembly. If a new shaft is being used, ensure that it is the same distance from the end of the shaft as the match mark on the old shaft.
24. Install the gear reducer, ensuring that the previously installed shims are replaced in their original positions.
25. Lubricate the couplings.
    - 25.1 Install a grease fitting or the grease gun hose into the coupling.
    - 25.2 Rotate the coupling until the grease fitting or plug hole is 45° from the top center.
    - 25.3 Pump an approved grease into the coupling until the excess flows from the lower plug hole.
    - 25.4 Remove the grease fitting or the grease gun hose.
    - 25.5 Install the lube plugs, and tighten them snug tight.
    - 25.6 Wipe off the excess grease.
    - 25.7 Repeat steps 25.1 through 25.6 for the remaining coupling.
25. Align the motor, gear reducer, and pump.
26. Connect the couplings.
27. Torque the coupling bolts to the appropriate torque.
28. Lubricate the couplings with an approved lubricant.
29. Place the new front cap gasket over the front cap, and ensure that the gasket is seated all the way.
30. Install the front cap, ensuring contact with the primary packing spring.

31. Install and tighten the front cap nuts, ensuring that the front cap is pulled down evenly.
32. Torque the front cylinder cap nuts to the appropriate torque.
33. Repeat steps 29 through 32 for the remaining two front caps.
34. Fill the pump with oil and install the base cover with a new gasket.
35. Install the base cover screws and tighten them evenly.

4.4.4.3 Fluid Drive Reassembly

Refer to Figure 4-33 while following this procedure.

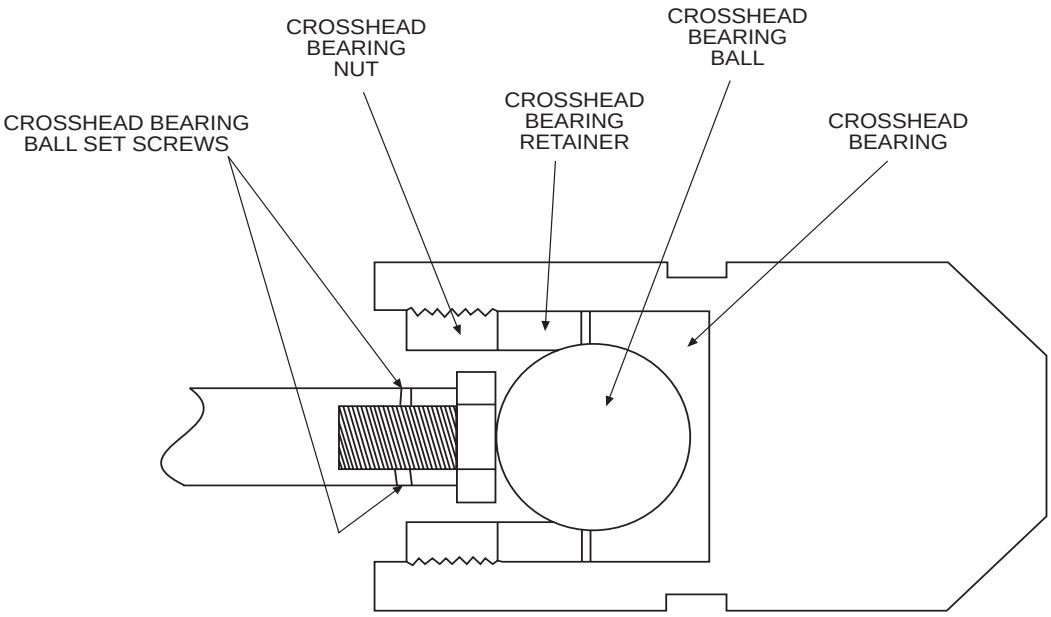


Figure 4-33
Crosshead Ball and Socket

NOTE: It is recommended that new anti-friction bearings be used on reassembly. When anti-friction bearings are assembled, they should be heated in oil to a temperature of 250–300°F. The impeller and bearing backing plate must be on the runner shaft before assembly of the pilot bearing.

NOTE: It is recommended that new gaskets be used on reassembly. Be sure that the gaskets are cut exactly the same as the original gaskets. Gaskets are made of 1/32" thick Armstrong 852 or the equivalent. Seal the gaskets with Permatex No. 3. Do not use an excessive amount of Permatex because the excess can contaminate the oil.

1. Assemble the rotating members.
   1.1 Install the runner on the output shaft, using the cap screws if they were removed.
   1.2 Install the output shaft.
   1.3 Install the impeller, casing, and output inboard bearing.

NOTE: Before installing the impeller casing to the runner casing, be sure to locate the balancing match marks and align them.

   1.4 Install the impeller casing to the runner casing, using the cap screws.
   1.5 Install the bearing retainer plate, using the cap screws.
   1.6 Install the input shaft in the impeller casing, using the cap screws.
   1.7 Install the input inboard bearing.
   1.8 Install the pump drive sprocket and the set screw. Do **not** tighten the set screw.
   1.9 Install the impeller pump, using the cap screws.
2. Assemble the input end.

NOTE: Inspect the chain to be sure that it is free from dirt and grit before installing it. Fit the chain around both sprockets, bringing the free ends together on the pump sprocket. A chain puller might be required to bring the free ends of the chain together so that the connecting link can be inserted and secured in place. Place a straight edge over the chain span, and push on the span center with a steel ruler so that the opposite chain span is taut. The sag is the dimension between the straight edge and the chain span. The maximum recommended chain sag for assembly is 5/8".

   2.1 Install the input outboard bearing, drive chain, and input end bell.
   2.2 Install the bearing retainer ring.
   2.3 Install the shaft dust seal, and tighten the set screw.
   2.4 Using the lifting assembly on the input end shaft, lower the rotating assembly vertically into the housing.
   2.5 Connect the pump to the housing coupling.
   2.6 Install the cap screws in the input end bell.
   2.7 Install the gear pump to the input end bell, using the four cap screws from the pump housing.
   2.8 Install the pump sprocket and the set screw. Do **not** tighten the set screw.
   2.9 Install the access cover and the cap screws.
   2.10 Tighten the pipe coupling to the external pump suction.

NOTE: Accurate alignment of the drive sprocket and the pump sprocket is required to obtain the maximum chain and sprocket life. Assemble both sprockets, but do not tighten the set screw. Accurately align both sprockets axially by placing a straight edge or heavy cord across both sprocket surfaces. After the sprockets are aligned, be sure to tighten the sprocket set screws.

3. Assemble the output end.

NOTE: The scoop tube must not rub on the rotating runner casing when in the minimum output speed position. Assemble the control rod stops exactly as on the original installation. If new control rod stops are installed, set one stop to leave 1/8" clearance between the scoop tube tip and the runner casing in the minimum output speed position. Set the other stop to give the proper stroke travel.

3.1 Install the control rod bracket, scoop tube, and scoop guide.

3.2 Install the output outboard bearing and the bearing retainer, using the elastic stop nuts.

3.3 Install the bearing retaining ring.

3.4 Install the dust seal and tighten the set screw.

3.5 Install the output shaft seal cover, using the cap screws.

4. Install the speed control rod end assembly if applicable. Refer to Figure 4-34.

4.1 Push the control rod away from the side to which the control rod end is to be mounted until the control rod is against the stop.

4.2 Attach the guide plate to the outside of the housing loosely.

4.3 Thread the jam nut on the control rod.

4.4 Thread the adjusting rod on the control rod until it is flush with the outer edge of the guide plate, and tighten the jam nut.

4.5 Thread the jam nut on the rod end, and thread the rod end into the adjusting rod.

4.6 Tighten the jam nut and the guide plate bolts.

5. Install the speed control crank assembly if applicable. Refer to Figure 4-35.

5.1 Position the control rod in the approximate center of travel.

5.2 Thread the adjusting shaft on the end of the control rod until the shoulder on the adjusting shaft is flush with the outside edge of the housing.

5.3 Attach the holding plate to the housing.

5.4 Attach the balance handle by either tightening the set screw into the hole in the adjusting shaft or inserting the roll pin, whichever is provided.

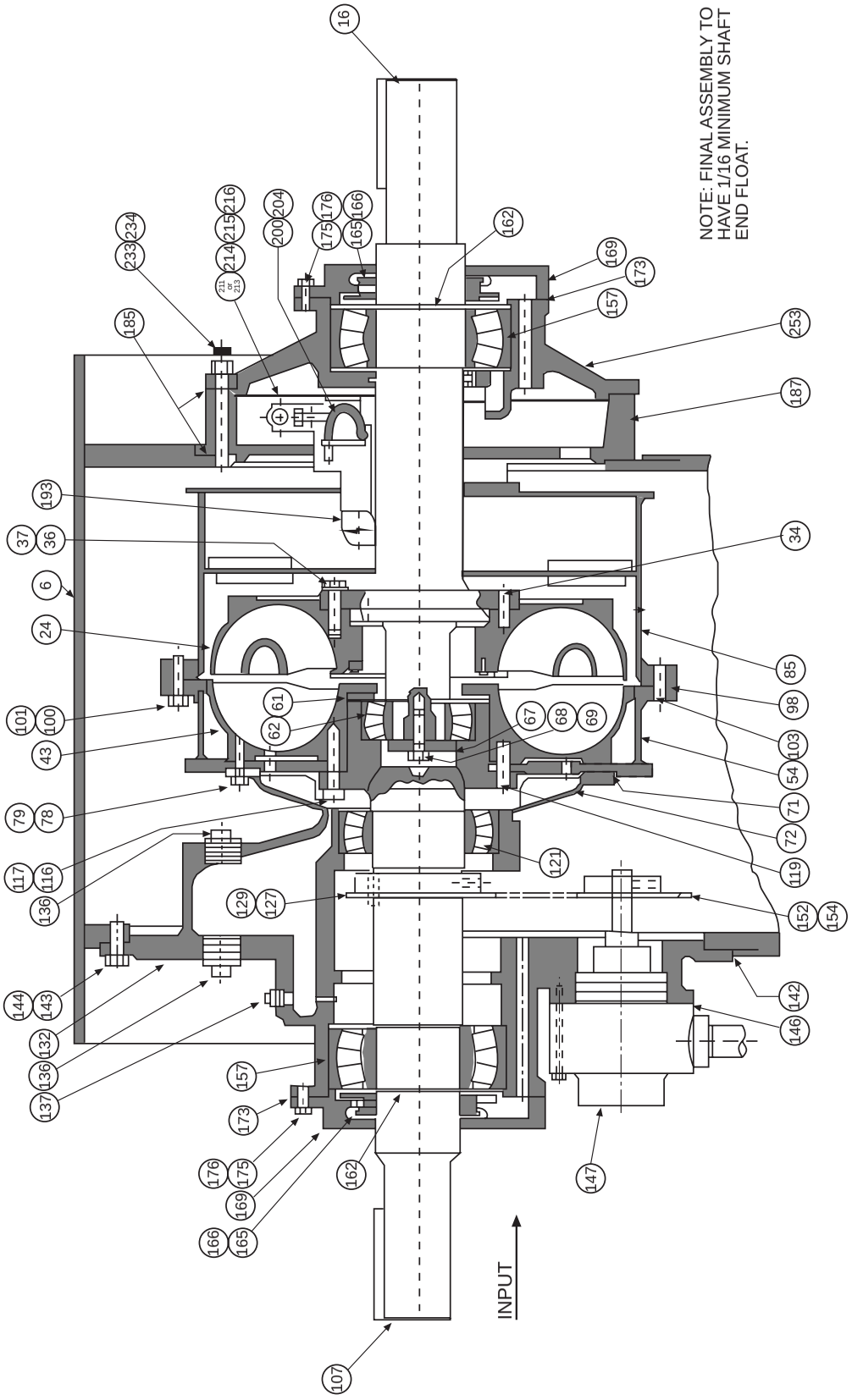


Figure 4-34
Speed Control Rod End Assembly

ITEM NO.	DESCRIPTION
6	Housing
16	Output Shaft
24	Runner
34	Dowel Pin
36	Hex. Hd. Cap Screw
43	Impeller
54	Impeller Casing Assembly
61	Bearing Back Up Plate
62*	Output Inboard Bearing
67	Bearing Retainer Plate
68	Hex. Hd. Cap Screw
69	Flatwasher
71*	Impeller Oil Pump Gasket (1)
72	Impeller Oil Pump
78	Hex. Hd. Cap Screw
79	Flatwasher
85	Runner Casing
98*	Impeller-Runner Casing Gasket (1)
100	Hex. Hd. Cap Screw
101	Flatwasher
103	Dowel Pin
107	Input Shaft
116	Hex. Hd. Cap Screw
117	Flatwasher
119	Dowel Pin
121*	Inboard Bearing
127*	Drive Sprocket
129*	Chain and Connecting Link
132	Input End Bell
136	Sq. Hd. Pipe Plugs
137	Sq. Hd. Pipe Plug
142*	Input End Bell Gasket (1)
143	Hex. Hd. Cap Screw
144	Flatwasher
142*	Input End Bell Gasket (1)
143	Hex. Hd. Cap Screw
144	Flatwasher
146*	Pump Gasket (1)
147*	Pump Assembly
152*	Pump Sprocket
154	Woodruff Key-Sprocket
157*	Outboard Bearing
158*	Outboard Bearing
162*	Retaining Ring (2)
165	Dust Seal Ring
166	Set Screw - Cup Point
169	Shaft Seal End Cover
173*	End Cover Gasket (2)
175	Hex. Hd. Cap Screw
176	Flatwasher
185*	Output End Bell Gasket (2)
187	Output End Bell
193	Scoop Tube Assembly
200	Scoop Tube Guide
204	Soc. Hd. Cap Screw
211	Control Rod Bracket
214	Shim (As Req.)
215	Hex. Hd. Cap Screw
216	Flatwasher
233	Stud
234	Elastic Stop Nut
253	Output Bearing Retainer
262*	Breather Filter (1)

* Recommended Spare Parts

Figure 4-34 (continued)
Speed Control Rod End Assembly

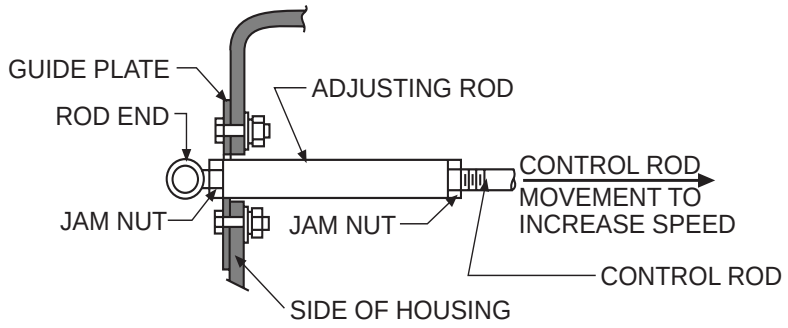


Figure 4-35
Speed Control Crank Assembly

6. Install the speed control lever assembly if applicable. Refer to Figure 4-36.
   - 6.1 Push the control rod away from the side of the housing to which the lever is to be attached until the control rod is against the stop.
   - 6.2 Bolt the guide plate and the bracket on the outside of the housing loosely.
   - 6.3 Thread the jam nut on the control rod.
   - 6.4 Thread the adjusting rod assembly on the control rod until it is flush with the outer edge of the guide plate.
   - 6.5 Tighten the jam nut against the adjusting rod.
   - 6.6 Attach the fork to the end of the adjusting rod.
   - 6.7 Insert the lower pin of the lever assembly through the yoke of the lever. Secure the pin with cotter keys.
   - 6.8 Insert the lower pin into the fork.
   - 6.9 Put the pivot pin through the bracket and the lever assembly. Secure the pin with cotter keys.
   - 6.10 Tighten the bracket and the guide plate bolts.
   - 6.11 Thread the jam nut on the yoke end, thread the yoke end into the lever, and tighten the jam nut.
7. Adjust the speed indicator.
   - 7.1 Attach the speed indicator assembly near the end of the control rod opposite to the speed control rod end, crank, or lever.
   - 7.2 Thread one jam nut on the control rod, add the indicator, and thread the other jam nut on the control rod.
   - 7.3 Attach the indicator strap to the housing and the output bearing retainer.
   - 7.4 Attach the indicator plate and arrow to the indicator strap in the proper position. When viewed from the output end, the speed increase arrow should always point to the right, and the letters should read from left to right.
   - 7.5 Position the control rod in the minimum output speed position. When viewed from the output end, the control rod is pushed to the left against the stop for the minimum output speed position.
   - 7.6 Adjust the indicator to line up with the letter A on the indicator plate with the control rod in the minimum output speed position, and tighten the jam nuts.

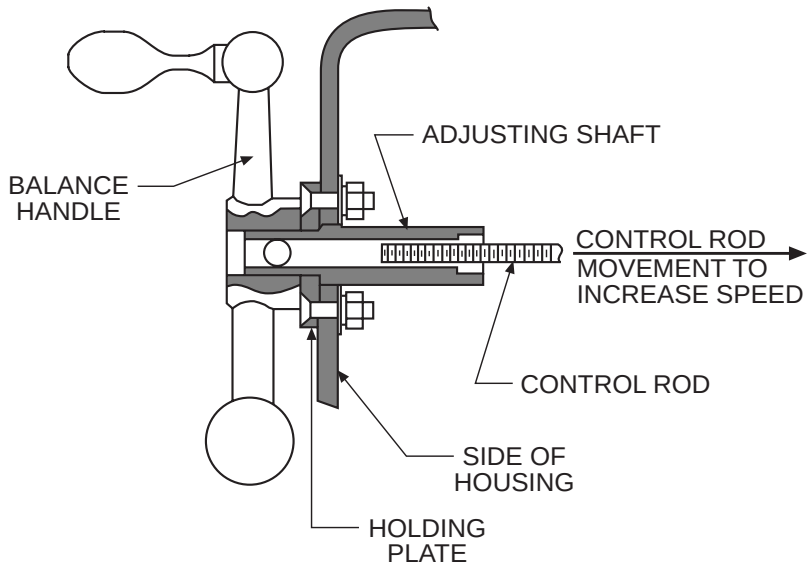


Figure 4-36
Speed Control Lever Assembly

8. Reinstall the fluid drive onto the sub-base.
9. Reconnect any speed controller connections that were previously disconnected.
10. Reconnect the driving and driven equipment to the fluid drive.
11. Reconnect the oil cooler piping.
12. Refill the fluid drive with new oil.

5

POST-MAINTENANCE STARTUP, OPERATION, AND TESTING

5.1 Centrifugal Charging Pump Startup

These general guidelines for centrifugal pump startup are not intended to take the place of approved operating procedures.

5.1.1 Pre-Startup Preparations

5.1.1.1 Flush the Lubricating System

The entire lubricating system, including the reservoir, oil pump, cooler, and piping, should be cleaned before starting the pump after maintenance.

1. Remove the pump sleeve and the Kingsbury bearings.
2. Remove the reservoir cover.
3. Wash the reservoir interior with kerosene and wipe with clean, lint-free rags.
4. Fill the reservoir with warm oil (180–200°F).
5. Start the auxiliary oil pump and circulate oil through the system.
6. Lightly tap or vibrate the entire length of piping to loosen scale or dirt buildup inside the piping. Circulate the oil and vibrate the piping for several hours, observing the oil returning to the reservoir to verify that loose scale and dirt have been flushed.
7. When all loose scale and dirt have been flushed, drain the reservoir and piping.
8. Remove sections of piping where scale and dirt might have accumulated (low or restricted sections). Soak and clean the piping sections with kerosene and a brush.
9. Wipe the reservoir and piping internals with clean, lint-free rags.
10. Reinstall the reservoir cover.

11. Reinstall the piping.
12. Clean and install the Kingsbury bearing.
13. Measure and record the thrust collar runout.
14. Before reinstalling the top half of the bearing housing, pour one pint of clean oil over the entire bearing.
15. Renew the gasket and reinstall the top half of the bearing housing.
16. Clean and reinstall the pump sleeve.
17. Fill the system with cool oil (less than 128°F) to the required level.
18. Run the auxiliary oil pump to fill the oil lines, and add oil to restore the proper oil level.

NOTE: The pump must be completely filled with liquid. The rotating parts within the pump depend upon this liquid for lubrication and can seize if the pump is operated dry.

5.1.2 Pump Startup

1. Ensure that the retainer plates are out of the shaft sleeve groove. Ensure that the shaft rotates by hand. Use the strap wrench if necessary.
2. Open the valves for packing / seal cooling, if supplied.
3. Lift the relief valves by hand to verify freedom of operation. The gland nuts should be finger tight.
4. Open the pump suction valve.
5. Prime the pump by using one of the following methods:

NOTE: If the pump is the "self-priming" type, the pump primes itself automatically by some internal hydraulic device or by recirculation.

   a. For a pump that has a positive head on the suction, open the vent on top of the pump casing. Shut the vent when all entrapped air has escaped.

   b. For a pump that has a suction lift, prime the pump by the means provided.

NOTE: The discharge valve must be opened slowly to avoid abrupt changes in velocity and surging in the inlet line.

6. Start the pump.
7. When the pump reaches full speed and pressure, slowly open the discharge valve fully.
8. Verify that all temperatures and pressures are normal.

5.2 Centrifugal Charging Pump Operation

5.2.1 Normal Operation

After the pump has been started and final adjustments have been made, a periodic check of the following should be made:

NOTE: As the oil temperature rises, the oil level might drop slightly. Ensure that the normal oil level is established after operating temperatures have been reached.

- Bearing temperatures and oil level
- Cooling water system, both for the oil cooler (if installed) and the stuffing box
- Suction and discharge pressure gauges
- Vibration levels
- Satisfactory packing/seal operation

NOTE: It is important that the pump not be operated at less than the design minimum flow. Doing so causes undue vibration, noise, and/or overheating.

NOTE: Always reduce the capacity by throttling the discharge valve. Never reduce the capacity by throttling the suction valve because a sudden demand for more liquid could result in a starved inlet and possibly a "dry" pump before the inlet valve can be opened enough to meet the changed demand. Throttling the inlet could reduce the pressure in the impeller eye below the minimum required, causing cavitation.

5.2.2 Operational Testing

1. Check for unusual vibration by doing the following:
   - 1.1 Feel the bearing caps by hand.
   - 1.2 Listen to various parts of the pump, using a screwdriver, stethoscope, or a portable vibration meter.
2. Using a hand-held mechanical tachometer or a strobe tachometer, check the pump RPM.
3. Ensure that the pump has proper suction and discharge pressures.
4. Using a clamp-on volt and ammeter, ensure that the pump motor is not drawing excess current.

5. Check the following casing and piping joints for leakage:
   - Casing joints
   - Sealing lines
   - Seals
   - Gauge lines
   - Recirculating line, if installed
   - Flange fittings
   - Valves

NOTE: If you cannot hold your hand against the bearing housing, use a thermometer with the bulb imbedded in a mound of clay or putty that is on the external surface of the bearing housing. The maximum allowable temperature for a babbitted journal bearing is 180°F. For general service roller bearings, the maximum allowable temperature is 230°F. Cooling water outlet temperatures of 120–130°F are desirable for both types of bearings.

6. Allow the pump to run for about 20 minutes to reach normal operating temperature, and then check the bearing housing. The temperature of the fluid being pumped determines the temperature of the shaft and pump casing and is a contributing factor in bearing temperature. Normally, you can hold your hand against the housing at temperatures up to about 130°F.

5.3 Centrifugal Charging Pump Testing

5.3.1 Vibration

On-line vibration monitoring is a diagnostic tool used to detect early signs of pump problems. The key to a successful vibration monitoring program is not only the integrity of the data, but also the repeatability of measurement locations and mounting of the sensors. The types of vibration monitoring systems currently in use in the industry are the portable data collection system and the continuous data collection system.

Portable data collection systems typically have from one to four signal inputs and are compatible with a variety of transducer types. Multi-channel units are often able to calculate cross-channel phase. The typical portable data collection system is shown in Figure 5-1. The main disadvantage of portable data collectors is lack of repeatability, usually due to not locating the sensors in exactly the same place every time.

There are at least two important applications for continuous vibration monitoring systems. One application is the continuous examination of journal bearing condition, so that an immediate shutdown signal can be generated in case of impending catastrophic failure. The other application uses sensors and multiplexers for routine monitoring of machines where it would be difficult or impossible to collect the data by hand. These

systems usually do not sense each vibration signal continuously, but are organized to examine each location in a prescribed order and to save spectral or time-domain data according to a predetermined schedule and / or when measured levels exceed predetermined limits. A typical multiplexed system is shown in Figure 5-2.

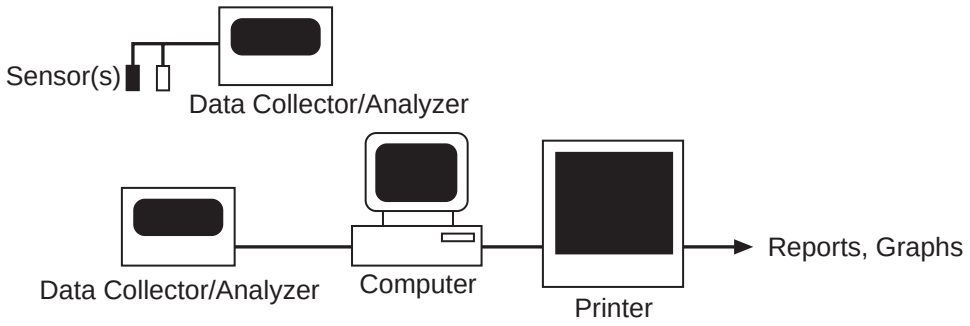


Figure 5-1
Portable Data Collection System

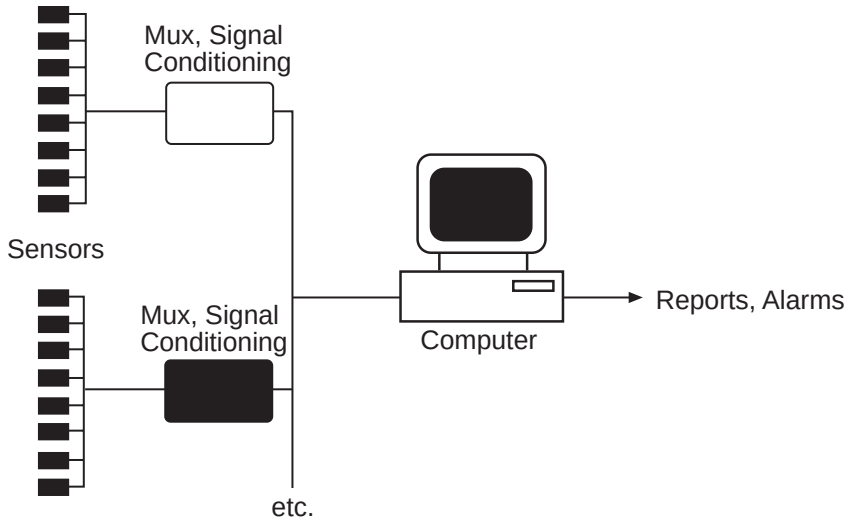


Figure 5-2
Multiplexed Continuous Data Collection System

Most of these systems have no data processing at the sensor locations, and a few of them have minimal processing in the multiplexers. This way, the multiplexer can send data to the host computer in order of severity of the measured RMS levels. This is desirable in that it can analyze the most significant data without waiting for the entire multi-

plexing cycle to repeat. With a system containing several hundred sensors, this becomes an important issue. Some newer continuous monitoring systems use digital signal processing (DSP) at each sensor location. These systems use chips that can be programmed to perform various analysis functions, depending on the condition sensed at the measurement point.

Typical sensor mounting positions for a horizontal pump are shown in Figure 5-3.

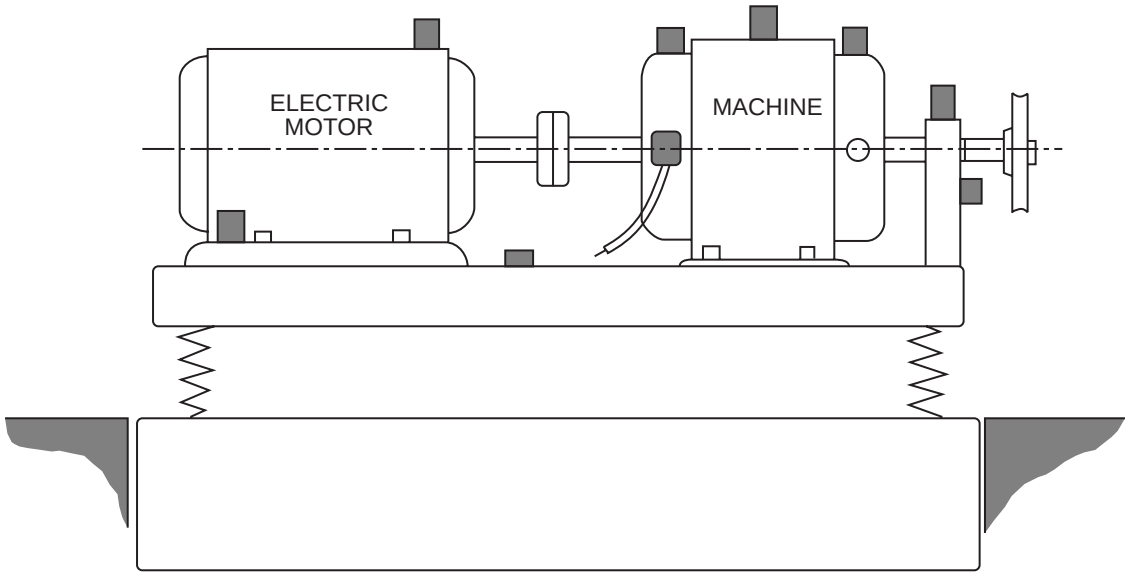


Figure 5-3
Horizontal Pump Sensors

Vibration conversion values are listed in Table 5-1 in order to convert from one reading to another.

Table 5-1
Vibration Conversion Values

	Multiply the following by the numerical values:			
To obtain	Average	RMS	Peak	Peak-to-Peak
Average	1.0	0.900	0.636	0.318
RMS	1.111	1.0	0.707	0.354
Peak	1.571	1.414	1.0	0.500
Peak-to-Peak	3.142	2.828	2.0	1.0

A vibration diagnostic chart is presented in Table 5-2 to aid in the vibration-related troubleshooting of pump problems.

Table 5-2
Vibration Diagnostic Chart

CAUSE	AMPLITUDE	FREQUENCY	PHASE	REMARKS
Unbalance (worn or lost parts, bent shaft)	Proportional to unbalance. Largest in the radial direction	1 x RPM.	Single reference mark.	One of the most common causes of vibration. Appears after maintenance or after long periods of use.
Bent shaft	Steady.	1 x RPM, 2 x RPM if bent at coupling.	180° out of phase axially.	
Thermal bow	Varies during operation.	1 x RPM.	Single reference mark.	Increasing vibration during variations in load or during start-up from a cold condition.
Misalignment of couplings or bearings	Large in the axial direction. 50% or more of radial vibration	1 x RPM usually. Sometimes 2 or 3 x RPM	Single, double, or triple.	One of the most common causes of vibration. Best found by the appearance of large axial vibration. Appears after maintenance or after long periods of use.
Mechanical looseness		1, 2, and 3 x RPM.	Two reference marks. Slightly erratic.	
Soft foot	Variable with the load.	Normally 1 x RPM. Sometimes 2 x RPM.		Check the mountings for variations in vibration amplitude.
Bad bearings, anti-friction type	Unsteady. Use velocity, acceleration, or spike energy measurements if possible	Very high. Several times RPM.	Erratic.	The bearing responsible is most likely the one nearest the point of the largest high-frequency vibration.

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Table 5-2 (continued)
Vibration Diagnostic Chart

CAUSE	AMPLITUDE	FREQUENCY	PHASE	REMARKS
Eccentric journals	Usually not large.	1 x RPM.	Single mark.	If on the gears, the largest vibration is in line with the gear centers. If on the motor, the vibration disappears when the power is turned off. If on the pump, attempt to balance.
Electrical	Disappears when the power is turned off.	1 x RPM or 1 or 2 x line frequency.	Single or rotating double mark.	If the vibration amplitude drops off instantly when the power is turned off, the cause is electrical.
Rubbing		0-50% of 1 x, 1 x, higher harmonics.	Erratic.	Many frequencies over the entire frequency spectrum.
Hydraulic forces		1 x RPM or the number of blades on impeller x RPM and harmonics.		
Bad gears or gear noise	Low. Use velocity or acceleration measurements if possible.	Very high. Number of gear teeth x RPM.	Erratic.	
Cracked shaft	Variable during transients.	1 x, 2 x RPM.	Change in 1 x phase often occurs.	2 x excitation of critical during coast down. Other faults can also cause change in 2 x RPM.
Structural resonance	Steady or variable.	1 x RPM but often odd, non-synchronous frequency.	Erratic.	If the problem starts on an old machine, check for soft feet or cracked supports.

5.3.2 Capacity

If the static heads, pressure differences, and friction-head losses of any system are combined and plotted against the capacity, the resulting curve is called the *system-head curve*. Superimposing a pump head-capacity curve at constant speed on this system-head curve permits us to determine the capacity at which the two curves intersect, and this is the capacity that is delivered into the system by the pump at the given speed. Figure 5-4 shows a system-head curve, while Figure 5-5 shows a pump head-capacity (H-Q) curve superimposed on the system-head curve.

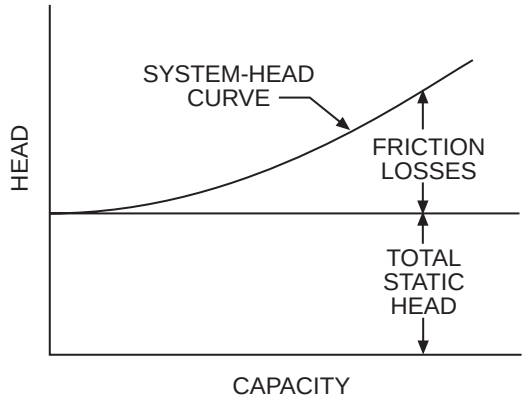


Figure 5-4
System-Head Curve

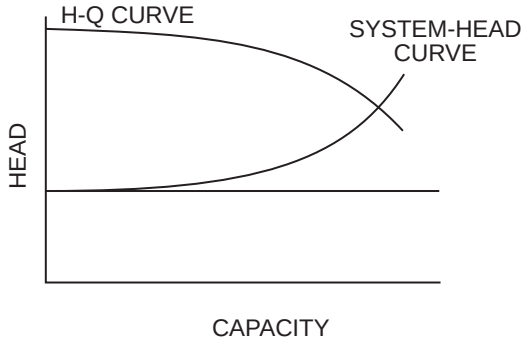


Figure 5-5
H-Q Curve Superimposed on a System-Head Curve

For systems with varying static heads or pressure differences, it is possible to construct curves corresponding to the minimum and maximum conditions, as shown in Figure 5-6. The corresponding intersections with the pump head-capacity curve then determine the minimum and maximum flows that the pump delivers into the system.

The performance of a centrifugal pump is generally described in terms of the following characteristics:

- Rate of flow, or capacity *Q*, expressed in units of volume per unit time, most frequently ft^3/s, gal / min, or m^3/h (1 ft^3/s = 449 gal / min; 1 m^3/h = 4.403 gal / min)
- Increase of energy content in the fluid pumped, or head *H*, expressed in units of energy per unit of mass, usually ft.-lb. / lb. or, more simply, ft, or m
- Input power P, expressed in units of work per unit of time, Bhp
- Efficiency η, the ratio of useful work performed to power input
- Rotative speed N, in rev / min

Because the parameters indicated are all mutually interdependent, it is customary to represent the performance of a centrifugal pump by means of characteristic curves. For any given capacity on such a characteristic curve, the relationship between performance characteristics is expressed by the equation $\eta = \gamma QH/3960P$, where η is efficiency expressed as a decimal, γ is the specific gravity of the fluid pumped, *Q* is in gal / min, *H* is in ft, and *P* is in Bhp.

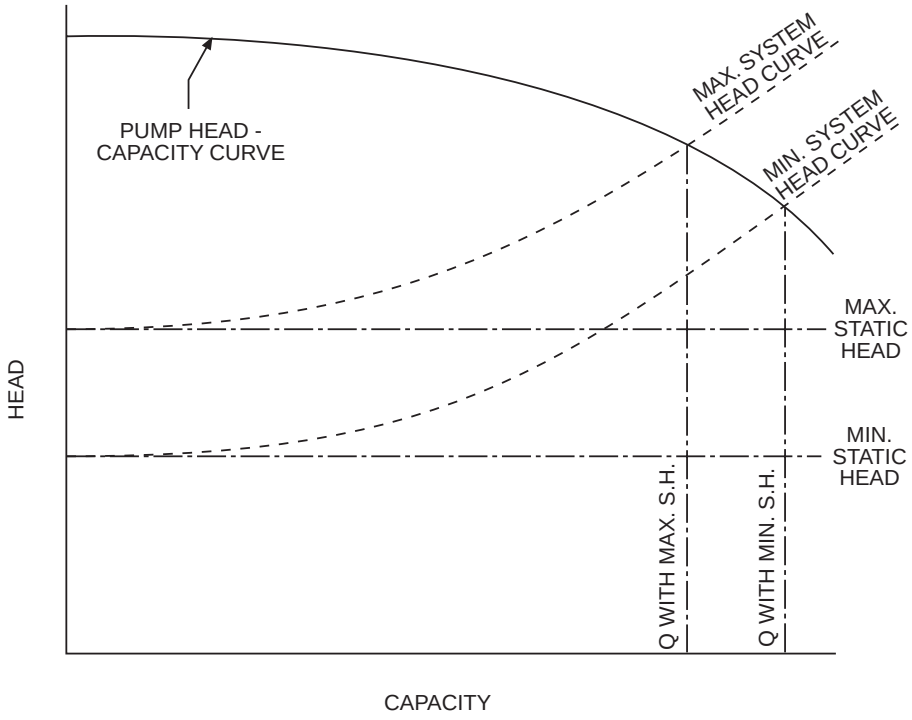


Figure 5-6
Characteristics of a System with a Variable Static Head

Because the capacity at which a pump can operate corresponds to the intersection of the system-head curve with the pump head-capacity curve, any changes in pump capacity can be obtained only by varying one or the other of these two curves. Thus, the capacity of a centrifugal pump operating in a system can be regulated by changing the pump speed or throttling the discharge valve. The pump head-capacity curve can be changed by operating the pump at variable speeds as shown in Figure 5-7. The shape of the system-head curve can be altered by creating artificial losses through throttling a valve in the pump discharge as shown in Figure 5-8.

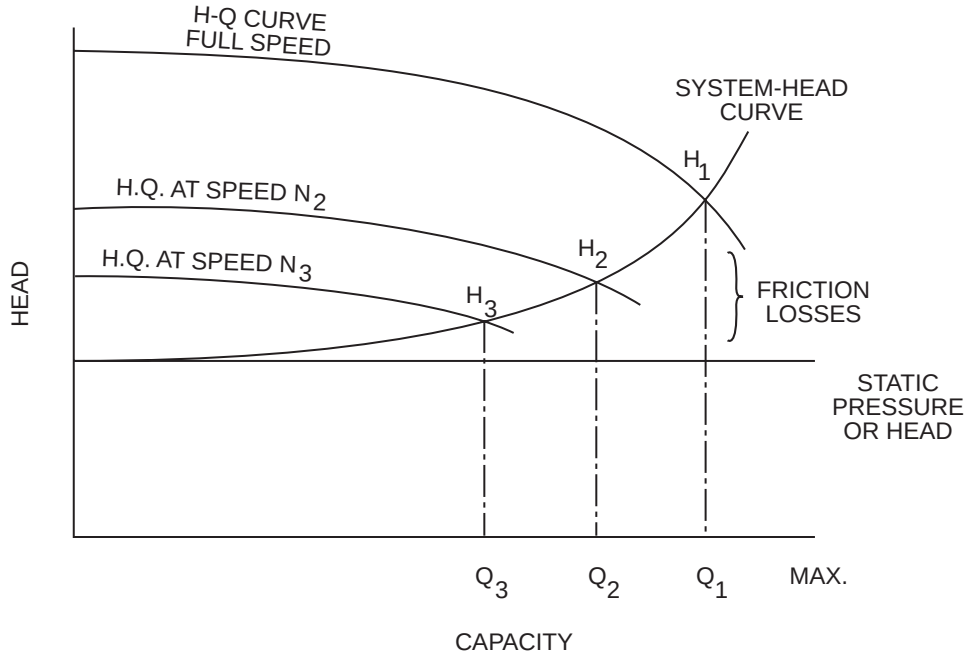


Figure 5-7
Varying Pump Capacity by Varying Speed

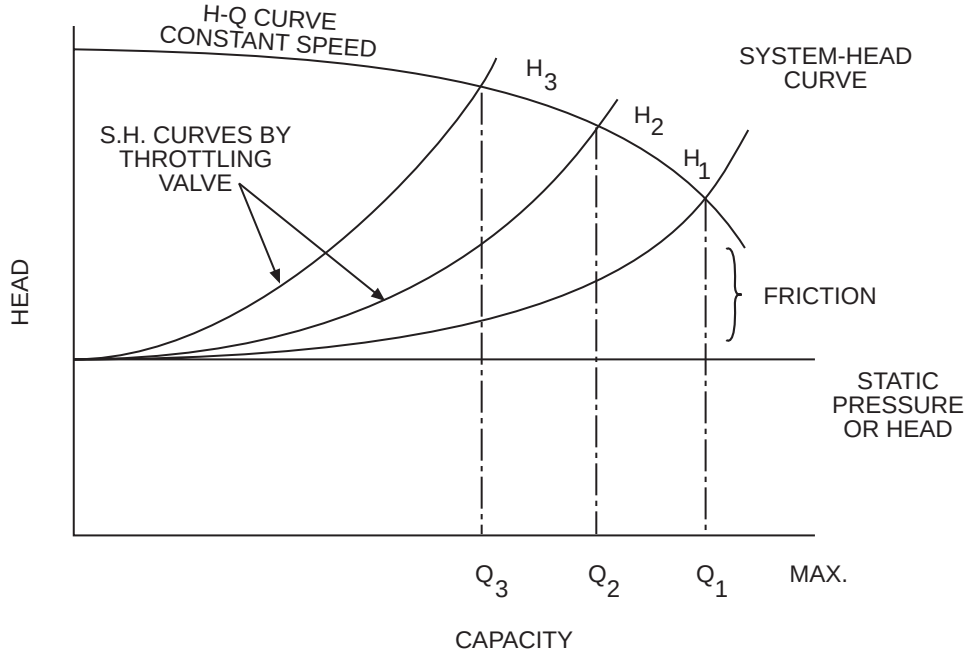


Figure 5-8
Varying Pump Capacity by Throttling

If entrained air or gas is permitted to enter a centrifugal pump along with the liquid it is pumping, the performance of the pump decreases. The amount of air or gas that can be handled is probably in the range of 1/2% by volume. The effect on pump capacity at various levels of air or gas entrainment is shown graphically in Figure 4-6.

In all centrifugal pumps, the affinity laws allow you to project capacity, head, and brake horsepower for small changes in pump speed. For a given pump, the affinity laws are:

- The capacity varies directly as the speed: $\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$
- The total head varies as the square of the speed: $H_1 / H_2 = (N_1 / N_2)^2$
- The brake horsepower varies as the cube of the speed: $BHP_1 / BHP_2 = (N_1 / N_2)^3$

5.3.3 Temperature

The thermodynamic problem that arises from the operation of a centrifugal pump at extremely reduced flows is caused by the heating of the liquid handled by the pump. The difference between the brake horsepower consumed and the water horsepower developed represents the power losses within the pump itself, except for a very small amount lost in the pump bearings. These power losses are converted into heat and transferred to the liquid passing through the pump. The temperature rise can be calculated from the formula:

$$°F\ rise = \frac{(Bhp - whp) \times 2545}{(Flow\ in\ lbs/hr \times C_p)}$$

where: 2545 = BTU equivalent of 1 hp hour

C_p = specific heat of the liquid

A more convenient formula relates the temperature rise to the total head and to the pump efficiency:

$$°F\ rise = \frac{Total\ Head\ in\ Feet \quad (1/e - 1)}{778\ C_p}$$

This formula can be used to plot a temperature rise curve directly superimposed on the performance curve of a centrifugal pump.

The maximum permissible temperature rise varies over a wide range, depending on the type of installation. If a pump were to operate against a completely closed discharge, the power losses would be equal to the shut-off brake horsepower and, because there would be no flow through the pump, all this power would go into heating the small quantity of liquid contained within the pump casing. The pump casing itself would heat up, and a certain amount of heat would be dissipated by radiation and convection to the surrounding atmosphere. However, because the temperature rise in the liquid pumped would be quite rapid, it is safer to ignore the dissipation of heat. On this basis, the temperature rise in the liquid contained in the casing can be determined from the formula:

$$°F\ rise/minute = \frac{42.4 \times Bhp\ at\ shut\text{-}off}{Weight\ of\ liquid\ in\ pump\ in\ lbs} \times \frac{1}{C_p}$$

Because operation of a pump may, on occasion, take place at shut-off or at extremely low flows, a means must be provided to prevent operation below whatever has been established as the minimum capacity. This is typically accomplished by installing a bypass in the discharge line from the pump side of the valves. The bypass must lead to some lower pressure point in the system where the excess heat can be dissipated. Normally, an orifice in this line is used to limit bypass flow to the desired value.

The other temperature of concern is bearing and lubricating oil temperatures. Oil, in addition to acting as a lubricant for the bearing surfaces, cools the bearings by carrying heat away from hot spots. The hot oil leaving the pump bearings is typically returned to an oil cooler, where it gives up its heat to cooling water. The oil is then returned to the oil sump for further use. Controlling the flow of cooling water by throttling valves allows a specific range of oil temperatures to be maintained. Normal oil temperatures run from 120–150°F. The maximum permissible bearing temperature is 180°F, at which point the bearing is termed a “hot bearing.” Some roller bearings are allowed to reach 230°F before they are considered overheated.

Both oil temperature and bearing temperature must be constantly monitored, using installed instrumentation to ensure that the line shaft components are receiving the proper cooling.

5.3.4 Pressure

To deliver a given volume of liquid to the system, a pump must impart to the liquid energy that is made up of the following components:

- Static head
- Difference in pressures existing on the liquid surfaces
- Friction head
- Entrance / exit losses

The static head refers to a difference in elevation. Thus, the total static head of a system is the difference in elevation between the discharge and suction liquid levels. The static discharge head is the difference in elevation between the discharge liquid level and the pump centerline. The static suction head is the difference in elevation between the suction liquid level and the pump centerline. If the static suction head is a negative value because the suction level is below the pump centerline, it is usually called the *static suction lift*. If either the suction or discharge liquid level is under a pressure other than atmospheric, this pressure can be considered either as part of the static head or as a separate item—an addition to the static head.

Friction head loss is the head expressed in feet of the liquid pumped that is necessary to overcome the friction losses caused by the flow of liquid through the piping, valves, and fittings. These losses vary roughly as the square of the flow through the system. They also vary with the size, type, and surface condition of the piping and fittings. These losses typically increase as the piping deteriorates with age. Friction head loss is usually estimated based on data established for average piping 10 or 15 years old.

If the supply of a pump originates in a reservoir or tank, losses occur at the point of connection of the suction piping to the source of supply. These are referred to as *entrance losses*. The magnitude of these losses depends on the design of the pipe entrance. Similarly, on the discharge side of the system where the discharge line terminates at some body of liquid, the velocity head of the liquid is entirely lost and must be considered as part of the total losses of the system. These are referred to as *exit losses*. For the sake of simplicity, entrance and exit losses are generally incorporated into the friction losses of the system.

The following formula can be used to determine the approximate total head that a centrifugal pump will develop:

$$H = f\,(DN / 1840)^2$$

where: H = total head in feet of liquid

f = a speed constant

D = impeller diameter in inches

N = pump revolutions per minute

At shutoff, f varies from 1.0–1.15, and at maximum efficiency, f varies from 1.10–0.75. Normally, an average of 0.95 is sufficiently accurate. To convert H in feet to pressure in psi, divide H by 2.31. To determine H for a multistage pump with equal impeller diameters, determine H for one stage and multiply by the number of stages; as an alternate method, determine H for each stage and add the values of H. Because suction pressure is seldom zero, expected discharge pressure should be computed by adding the positive suction pressure to or by subtracting the suction lift (vacuum) from the total head developed by the pump.

Both the suction pressure and discharge pressure need to be constantly watched to ensure proper pump operation. Typically, the suction side of the pump presents more problems than the discharge side.

Tests are conducted to demonstrate that the hydraulic performance of the pump is satisfactory. To achieve satisfactory results, good instrumentation is needed, as well as a test installation that is free of harmful flow disturbances on the suction side of the pump.

Configuration of the intake, the actual flow path, and the resulting intake velocities and flow patterns are key factors in both the test situation or actual installation. Consideration of end, back, and side wall clearances from the pump suction, along with the distance from the suction to the bottom of the intake are important factors. Intake channel disturbances must be avoided. Channeling devices installed to “improve” flow conditions must be carefully designed if they are to have the desired beneficial effect.

Very few test installations are totally free of hydraulic disturbances. In many instances, such disturbances exist but go undetected because they cannot be seen or easily and conveniently measured. Vortices that entrain air usually induce noise and vibration in the pump but do not necessarily affect the hydraulic performance of the pump. Hydraulic performance, however, is affected when the vortices are of such a magnitude that the entrained air blocks a portion of the water passage. Good practice dictates that intermittent and sustained air entraining vortices be eliminated before testing.

Geometry-induced circulation, as opposed to pre-swirl at reduced capacities, would normally go unnoticed even though it could drastically affect pump performance over the operating range of the pump. For both high and low specific-speed pumps, one possibility for determining the presence of circulation is to run performance tests at two different submergence pressures. If the results are not congruent, circulation could be a reason for the differences.

5.4 Positive Displacement Charging Pump Startup

*NOTE: **Never** start a positive displacement pump with the discharge valve shut because pump damage will occur.*

1. If the pump has a packing cooling/lubrication system, place this system in operation approximately 30 minutes before starting the pump.
2. Open the suction and discharge valves.
3. Verify that the suction pressure is sufficient to prevent pump cavitation.
4. Verify that the suction stabilizer and discharge dampener pressures are correct.
5. Verify that the oil level in the sightglass is at the half-full mark.
6. If the machine has a fluid drive, start the fluid drive as follows:

    6.1 Move the speed controller to the minimum output speed position. This allows the drive equipment to start under virtually no load.

    6.2 Start the driving equipment.

NOTE: When starting a fluid drive that has an air-type oil cooler located in low-temperature air, make sure that the bypass valve on the cooler line opens at the proper pressure.

NOTE: Normal vibration is 1 Mil or less (peak-to-peak) on the housing. If vibration is 3 Mils or greater on the housing, the fluid drive should be shut down for an investigation of the excessive vibrations.

    6.3 Move the speed controller slowly in the direction to increase the speed until the output shaft begins to turn. Allow the machine to run under these condition, and inspect to see that the fluid drive is operating smoothly and with no excessive vibration before increasing the speed.

NOTE: Before the oil temperature reaches 130°F at the oil cooler outlet, the water flow should be started through the oil cooler. If an air cooler is used, start the fan.

    6.4 Verify that cooling is established to the cooler.

NOTE: During operation of the fluid drive, the oil temperature can safely vary a considerable amount. The temperature of the oil leaving the cooler should be 130°F but can safely vary between 60–160°F. The temperature of the oil entering the cooler can go as high as 200°F. Any temperature above 215°F should be investigated immediately.

    6.5 Verify that the fluid drive parameters are normal.

NOTE: If your unit has a low oil pressure cutout switch, the start switch must be held in until the oil pressure reaches the minimum setpoint, typically 10 psi. Normal oil pressure should be in the range of 30–35 psi. Oil pressure can be changed by adjusting the oil system relief valve.

7. Start the machine by pressing the start button, and verify that the oil pressure reaches the normal operating pressure.
8. Verify that the pump is operating normally and that no abnormal noises are present.
9. Verify the proper packing gland leakoff.

5.5 Positive Displacement Charging Pump Operation

In addition to performance testing, vibration analysis, and pressure tightness testing using locally generated procedures, the following break-in procedure should be used if the packing or plungers were changed and the pump/system design allows for a break-in period:

1. 5 minutes at no load
2. 5 minutes at 25% load
3. 10 minutes at 50% load
4. 10 minutes at 75% load
5. 10 minutes, or until the outside temperature of the stuffing boxes has stabilized, at full load.

During this time, occasionally check the plunger well area for leakage. If the temperature of the stuffing box exceeds 140°F, stop the pump until the stuffing box cools.

If any bearings were changed out, the following procedure for bearing break-in should be followed:

1. Run the pump at 0% discharge pressure for 10 minutes.
2. Run the pump at 50% discharge pressure for 10 minutes.

NOTE: When the pump is running at 100% discharge pressure, the oil temperature must stabilize before the completion of the break-in period.

3. Run the pump at 100% discharge pressure for 1 hour.

During break-in and normal operation, the pump should be monitored for excessive vibration noise. If the oil temperature is below 180°F and stable, the pump is ready for normal service. If the oil temperature at any time exceeds 180°F, stop the pump, drain the oil, and inspect the power frame. Allow the pump to cool for 1 to 2 hours before retesting.

5.6 Positive Displacement Charging Pump Testing

5.6.1 Vibration

Vibration testing is typically performed either monthly or quarterly, with the majority of the plants performing it quarterly. It is also performed during surveillance testing, after maintenance has been performed on the pump. Although a few plants have their pumps instrumented for data collection, the majority of the plants use portable vibration data collectors. When using the portable data collectors, the technician must ensure that the vibration readings are taken at exactly the same spot every time for the trend analysis to be effective.

5.6.2 Capacity

Capacity testing on a positive displacement pump is very difficult to perform, because the head curve for these pumps is a straight vertical line. A flow verification is normally done after maintenance during the performance of the surveillance test.

5.6.3 Temperature

The temperatures of main concern in the positive displacement pump are bearing/lubricating oil temperature and stuffing box temperature.

5.6.4 Pressure

The discharge pressure on a positive displacement pump is the same as the system pressure that the pump is pumping to. Therefore, the main pressure of concern for the positive displacement pump is seal water pressure. Air binding of the seal water pump leads to reduced seal water pressure/flow to the stuffing box, causing the stuffing box temperature to rise and the packing wear rate to increase.

6

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Appendix A

COMBUSTION ENGINEERING PLANT CVCS

The chemical and volume control system (CVCS) takes reactor coolant system (RCS) water from the suction of reactor coolant pumps (RCPs).

To maintain RCS water chemistry and the purity of the RCS, the letdown portion of the CVCS is used to divert a small amount of RCS water through filters and ion exchangers where unwanted suspended solids and ionic impurities are removed from the fluid. The water is then returned to the RCS via the charging pumps along with any chemical additions that might be necessary to maintain desired conditions in the RCS. The CVCS consists of three subsystems, as shown in Figure A-1:

- **Letdown**: The portion of CVCS from the RCS to the volume control tank (VCT)
- **Charging**: Commonly considered to extend from the VCT outlet until the water enters the RCS
- **Chemical addition**: The portion of CVCS that provides makeup and chemical addition to the VCT and charging pump suction

NOTE: A general system description is provided in this appendix. Actual plant system and system operation vary from plant to plant. This description is intended to identify system modes of operation that affect charging pump operation and performance.

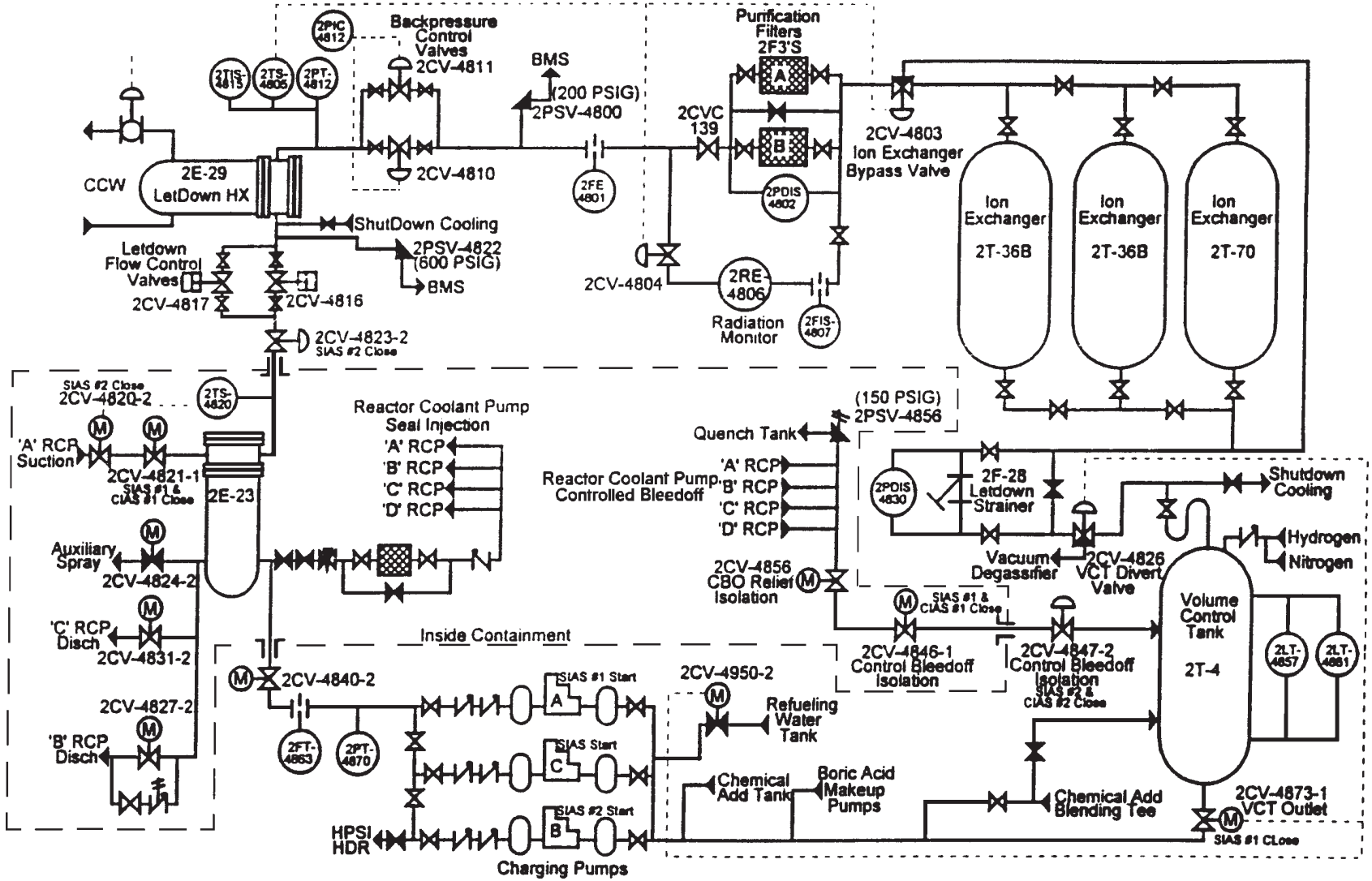


Figure A-1
Chemical and Volume Control System, On-Line Diagram
(Normal Mode of Operation)

Functions of the Chemical and Volume Control System

The primary functions of the chemical and volume control system are to:

- Maintain chemistry and purity of the RCS during normal operation and during shutdowns including crud burst cleaning
- Maintain the desired volume of water in the RCS
- Provide control of the boron concentration in the RCS
- Provide an alternate mechanism to reduce RCS pressure through the use of the auxiliary pressurizer spray
- Provide leak testing of the safety injection system and check valves
- Provide continuous measurement of fission product activity in the RCS
- Collect controlled bleed-off flow from RCP seals
- Detect any leakage that exists in the RCS
- Provide injection of concentrated boric acid during a safety injection actuation signal (SIAS)
- Provide an alternate method of filling the RCS
- Automatically divert letdown when the volume control tank reaches its highest permissible level

Design Criteria of the CVCS

The chemical and volume control system is designed to be capable of performing to the following specifications:

- Accept letdown when heating the RCS at the administrative heatup rate (75°F / hr) or provide makeup using two of the three charging pumps when the RCS is cooled at the administrative cooldown rate (75°F / hr) in order to maintain the desired RCS water volume
- Allow for a design plant power step change (±10%) and a ramp change (±5%) of full power per minute when power is between 15% and 100% power
- With VCT level in its normal operating level band, provide sufficient capacity to accommodate an RCS inventory change resulting from a power decrease from 100% to 0% power with no makeup
- Maintain RCS activity within technical specification limits assuming 1% failed fuel and continuous full power
- Maintain RCS chemistry within technical specification limits
- Allow addition of boric acid to the RCS at a rate capable of counteracting the positive reactivity addition produced during an administrative RCS cooldown rate (75°F per hour) and the maximum xenon decay rate using only one charging pump
- Allow the operator to borate the RCS to the refueling boron concentration required by technical specifications

CVCS Subsystem Matrix and Components

CVCS Process Radiation Monitor

The purpose of the CVCS process radiation monitor is to alert the operator to an increase in RCS radioactivity.

The CVCS process radiation monitor is a trend monitor whose primary purpose is to indicate the possibility of fuel clad failure. Adjustable alarm setpoints are normally set at a value slightly above the background reading. It is expected that gross activity will periodically increase above the alarm setpoints due to normal plant transients. Consequently, the alarm will periodically activate and the operator must determine the cause of the alarm.

In time, the RCS activity should return to the previous, lower, steady state value. If an alarm is received, the setpoints are immediately raised above the higher levels and the alarm circuits are reset. This ensures that any successive increases are brought to the operator's attention.

Volume Control Tank

This 4,200-gallon tank is provided in the CVCS system to perform the following functions:

- Provide a surge volume for the RCS
- Provide a method to maintain H_2 concentration in the RCS. (The VCT is normally blanketed with between 10 and 60 psi of H_2.)
- Provide a reservoir of RCS for charging pumps to ensure adequate net positive suction head (NPSH)
- Provide a readily identifiable method to determine if RCS leakage exists

The VCT is sized to provide sufficient volume so that with the VCT just within the low end of its normal operating band, a power load transient from 100% to 0% full power can still be accommodated without requiring additional makeup. With the VCT just within the high end of its normal operation band, it can still accept up to 500 gallons of additional water during dilution operations.

VCT Level Control

The VCT has a level control system that provides alarms and automatic action (shown in Figure A-2) to alert the operator, actuate alarms, and ensure NPSH to the charging pumps.

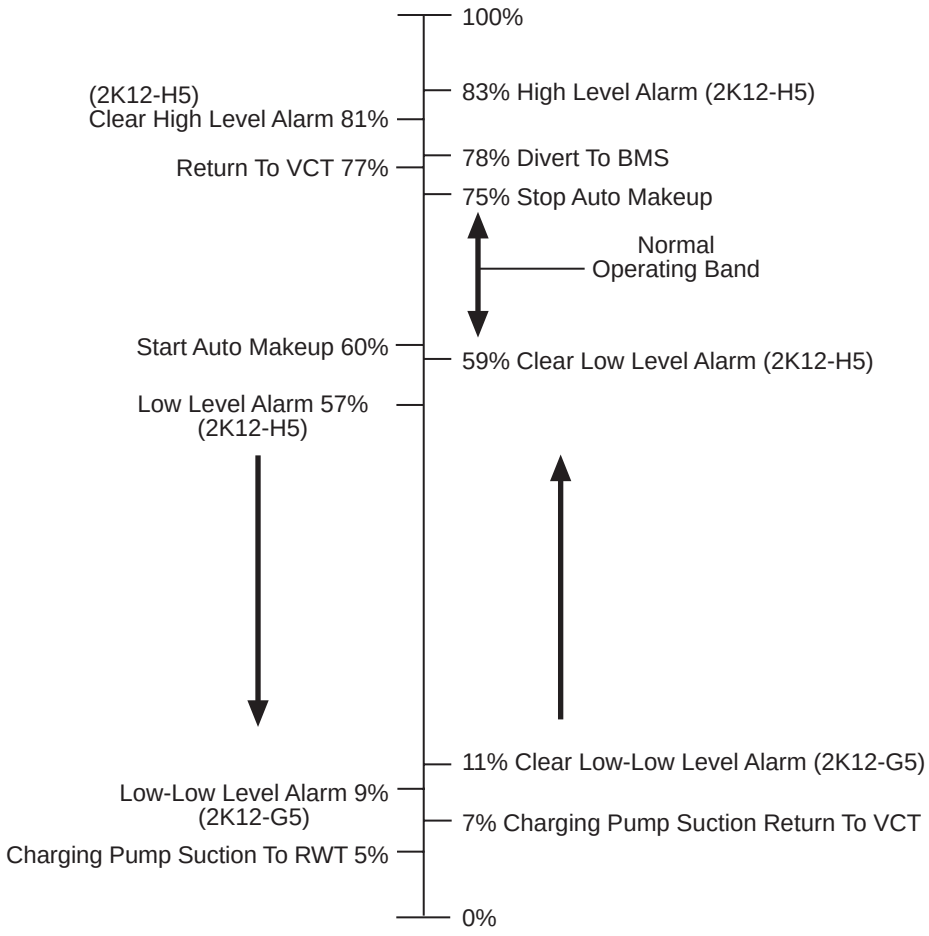


Figure A-2
VCT Level Control System

VCT Gas Control

The VCT is normally blanketed with 10 to 60 psig of hydrogen. This hydrogen overpressure helps maintain the oxygen concentration in the RCS within the technical specification limit of < 0.10 ppm O_2 for steady state operation and < 1.00 ppm for transient operation. Maintaining the oxygen concentration within these limits of the RCS is minimized and reduces the potential for RCS leakage or stress corrosion failure.

The hydrogen blanket also provides adequate NPSH for the charging pumps during normal operation. The pressure in the VCT may be monitored outside the VCT room. An alarm in the control room is provided to alert the operator to abnormal pressures in the VCT.

The pressure in the VCT is usually controlled at > 31 psig. The actual pressure is determined based on the actual and desired total dissolved gas concentration in the RCS.

RCP Controlled Bleed-Off Isolation Control Valve

The controlled bleed-off isolation valves of the RCP are provided to isolate bleed-off flow to the VCT. The controlled bleed-off flow is directed to the VCT to minimize the amount of makeup required for the RCS. Approximately 4 gpm of the RCS flows into the VCT (1 gpm from each RCP).

Controlled bleed-off flow from the RCPs passes through the excess flow check valves, which work on a forced balanced principle. The excess flow check valves are completely closed by 15 gpm.

To prevent damage to the RCP seals, the control bleed-off relief valve is installed on the controlled bleed-off line upstream of the controlled bleed-off flow path if the isolation valves close. The valve is typically set to lift at 150 psig, and it is sized to pass up to 13 gpm (equivalent to one RCP having a failed seal and the three other RCPs having normal controlled bleed-off flow).

A motor-operated isolation control valve is provided to allow isolation of the relief valve. This valve is controlled by a key-operated hand switch located on a panel. The valve hand switch is normally in the “locked open” position.

Two alarms for high pressure are provided on the RCP control bleed-off line, when the indicated pressure is > 120 psig and > 250 psig.

Auxiliary Pressurizer Spray Control Valve

The auxiliary pressurizer spray valve is used during times when normal pressurizer spray is not available because the RCP is secured. This motor-operated valve can be positioned from hand switches on the control room panel or the remote shutdown panel.

Letdown

N-16 Decay Chamber

To minimize radiation levels inside the auxiliary building, an N-16 decay chamber is provided. This is an enlargement in the letdown line just prior to the regenerative heat exchanger. Its purpose is to slow the letdown fluid to allow sufficient time for the radiation decay of N-16. N-16 has a half-life of approximately 7.13 seconds.

Letdown Line Stop Valve

The first component downstream of the RCS is a letdown line stop valve. This valve is located in the containment near the regenerative heat exchanger. The valve is motor-operated and controlled by a hand switch on a control room panel. A valve position indicator (open and closed) is provided on the panel.

The reason for having letdown isolated if the regenerative heat exchanger outlet temperature gets too high is to protect the heat exchanger from excessive thermal stress. When letdown is isolated and charging is still in service, thermal shocking of the charging nozzles at the RCS could result. The SIAS actuation is caused by low RCS pressure and is indicative of either a loss of coolant accident (LOCA) or an over-cooling event such as a main steam line rupture. In these events, the control valve closes to provide possible leak isolation and to minimize the amount of coolant leaving the RCS.

Regenerative Heat Exchanger Inlet Valve

The regenerative heat exchanger inlet valve is physically located in the containment just downstream of the letdown line stop valve. This valve provides physical isolation of the letdown system from the regenerative heat exchanger and also provides inside containment isolation capability of the letdown header. This valve is motor operated and can be controlled from the control room panel. A valve position indication (open and closed) is provided on the panel.

Regenerative Heat Exchanger

The regenerative heat exchanger is physically located in the containment area. This heat exchanger, with charging flow going back to the RCS providing the cooling medium., reduces letdown. The letdown heat exchanger preheats the charging system fluid to approximately 410°F before it enters the RCS. The regenerative heat exchanger also preheats the charging system fluid before it enters the RCS (approximately 410°F), which helps to reduce thermal stress at RCS charging inlet nozzles.

When the regenerative heat exchanger is in service, letdown fluid from the RCS flows through the tube side (at 40 gpm), while charging header fluid from the charging pumps flows through the shell side (at 44 gpm). The 4 gpm difference between these two flows is accounted for by controlled bleed-off flow into the VCT. This flow is normally 1 gpm from each of the four reactor coolant pumps.

Letdown Containment Isolation Valve

The letdown containment isolation valve provides outside containment isolation capability on the letdown header. This valve is an air-operated valve and is controlled from a hand switch on a control room panel. A valve position indication (open and closed) is provided on the panel. The valve will fail closed on loss of instrument air or on loss of dc power.

Letdown Flow Control Valves

The letdown flow control valves are provided to control the letdown flow from the RCS in order to maintain the desired RCS.

During system operation, the control room operator can select one or both of these valves to control flow using a hand switch on a control room panel. Normally only one of these valves is in service and it is controlled automatically by the letdown flow valve controller, located on the panel. The controller controls the position of the selected valves based on input from the pressurizer level control system.

The operator can take manual control of these valves if necessary due to system conditions or automatic control system malfunctions.

Mechanical Description of Control Valves

The letdown flow control valves are globe valves. Letdown flow passes into the bore inlet and then through the sides of the disc stack to the outlet port. The disc stack serves as a protective strainer, preventing particles from entering the system and possibly causing damage to system components. The plug can be positioned within the disc stack bore to expose more or fewer passages to obtain precise control of the flow.

The disc stack permits changes in flow rate while limiting flow velocity. The disc stack consists of a number of discs into which labyrinth flow passages have been etched to allow a fixed flow. Impedance in the passages is developed by a series of right-angle turns with a specific number of turns in each passage to limit velocity to an acceptable level.

Shutdown Purification Inlet

During shutdown cooling operations, the only method to maintain RCS purity is by using a flow path through the letdown system known as shutdown cooling purification. In the shutdown cooling system, a line physically taps off downstream of the shutdown cooling heat exchangers. This line is isolated by a valve and can be used to divert a portion of the shutdown cooling flow (RCS fluid) to the letdown system.

This line from the shutdown cooling system taps into the letdown system just downstream for the letdown flow control valves. While utilizing shutdown cooling purification, a maximum limit (50 psi RCS pressure) must be ensured to prevent challenging the downstream relief valve. Purification flow rate is controlled by using the back pressure control valves and / or controller in manual operation mode.

Letdown Pressure Safety Relief Valve

The letdown pressure safety relief valve is located downstream of the letdown flow control valves. This pressure safety valve (PSV) lifts at 600 psig and discharges to the holdup tanks (HUTs). The PSV is designed to protect the letdown heat exchanger and system piping from being pressurized above the design pressure of 650 psig.

Letdown Heat Exchanger

The letdown heat exchanger is provided to reduce the temperature of the letdown fluid from the outlet of the regenerative heat exchanger to a temperature compatible with the purification ion exchanger resin. A component cooling water (CCW) loop provides cooling water to the letdown heat exchanger.

Backpressure Control Valves

The backpressure control valves are provided to ensure that the pressure in the letdown heat exchanger is maintained above the saturation pressure of the letdown fluid entering the heat exchanger.

During shutdown cooling purification, control valves are used to control the amount of purification flow. The letdown relief valve is designed to protect the low pressure piping, filters, ion exchangers, and strainers from being over-pressurized. This safety relief valve typically relieves at 200 psig and discharges to the holdup tanks.

Boronometer and Process Radiation Monitor Isolation Valve

The process radiation monitor isolation valve serves to isolate the boronometer, which has been abandoned in place by some CE plant designs, and the process radiation monitor if high temperature conditions exist at the outlet of the letdown heat exchanger.

Purification Filters

The purification filters are cartridge-type filters that retain 98% of particles greater than 2 microns. The purpose of the filters is to remove insoluble particulate from the letdown prior to entering the purification ion exchangers. Under maximum letdown flow conditions, the expected differential pressure across the filters is 5 psid (when the inservice filter is clean) to as much as 60 psid (when the filter is clogged). If the differential pressure is > 30 psid, an alarm is activated.

These purification filters are in parallel and only one filter is in service at a time. The filters each pass the maximum letdown flow without exceeding the maximum differential pressure across the filter cartridge during maximum fouled conditions.

Purification Ion Exchangers

The purification ion exchangers are provided to remove ionic impurities from the RCS via the letdown system.[1] The ion exchangers are filled with (32 cubic feet) of usable mixed bed H-OH resin.[2] The ion exchangers are designed with the capability to allow maximum letdown flow.

When restoring an ion exchanger after loading new resin, it is important to realize that the resins will absorb large amounts of boron from the letdown fluid passing through it. This "diluted" fluid will be charged back to the RCS and will add positive reactivity. To preclude this, the standby ion exchanger is borated and lithiated by passing a trickle flow of letdown through the ion exchanger. Placing a new or unborated demineralizer in service is a reactivity manipulation.

Deboration Ion Exchanger

The deboration ion exchanger is loaded with H-OH resin identical to the purification ion exchangers and used as a backup purification ion exchanger. The construction and operation of the tank is identical to the purification ion exchangers.

Ion Exchanger Bypass Control Valve

The purpose of this valve is to divert letdown flow around the purification ion exchangers if the outlet temperature of the letdown heat exchanger increases to 145°F. This serves to protect the ion exchanger resins from damage. The valve automatically repositions to direct flow back through the ion exchangers if the letdown temperature decreases (below 140°F). Additionally, in the event of a dilution accident, the control valve will bypass the ion exchangers.

The air-operated control valve is controlled by a three-position hand switch located on the control room panel. When this switch is on auto, the valve automatically positions to bypass the ion exchangers. The valve fails to the bypass position on loss of instrument air or on loss of dc power supply.

Letdown Strainer

The letdown strainer is provided as a backup to the ion exchanger retention devices to prevent resin from entering the VCT. If the differential pressure across the strainer is in excess of 20 psid, an alarm sounds in the control room. A hard piped line is installed to flush the strainer to the spent resin tank.

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[1]The original design of the system had the ion exchangers filled with an Li form of resin for control of the lithium (Li) and cesium (Cs) concentration in the RCS. One demineralizer was in service for ionic impurities, and the other was in intermittent use for Li and Ca control.

[2]A given resin will remove either cations or anions, not both, from solution. To accomplish simultaneous removal of cations and anions, mixed bed resins are employed. In the mixed bed resins, a cation-exchange resin and an anion-exchange resin are placed in the same column. If the products of the two exchanges are H^+ and OH^-, the net result is to remove ions from solution to yield pure water since H^+ and OH^- produced during ion exchange combine to produce water (H_2O).

VCT Inlet Control Valve

The VCT inlet control valve is a three-way air-operated valve provided to direct letdown flow to the volume control tank (VCT) or the vacuum degassifier in the boron management system (BMS). The valve control circuit will fail to the VCT position on loss of instrument air or on loss of dc power.

Shutdown Purification Return Line

When on shutdown cooling purification, the shutdown cooling pump has a small portion of its discharge diverted to the letdown system in order to help maintain RCS water purity. After passing through the letdown purification filters and ion exchangers, this flow is returned to the suction of the low-pressure safety injection pumps.

Charging Header

The charging header receives water from the volume control tank, the refueling water tank, or the chemical addition system and discharges the water into the RCS.

This portion of the chemical and volume control system works in conjunction with the letdown portion to maintain the pressurizer level in the appropriate range. It provides a means to add highly borated water or pure water for activity control in the RCS. It is also used to perform check valve testing in the safety injection system.

VCT Outlet Control Valve

The VCT outlet valve is provided to isolate the VCT from the suction header of the charging pumps. The valve is a motor-operated valve and is controlled by a three-position hand switch located on the control panel.

RWT to Charging Pump Suction Isolation Valve

The refueling water tank (RWT) to charging pump isolation control valve is provided to align the RWT to charging pump suction. This valve is controlled by a three-position hand switch located on the control panel. When this switch is on auto, the valve automatically positions based on the VCT level.

Charging Pumps

RCS water is diverted to the letdown system to provide filtering and purification to the RCS. The charging pumps serve to send the water that was removed back to the RCS in order to maintain the RCS inventory at desired levels. During steady state operation, total letdown plus controlled bleed-off flows are equal to the total charging flow going back to the RCS.[3]

[3] During power operations, the control board operator can use this concept to identify any leakage that exists in the reactor coolant system. If the charging flow is greater than the total of the letdown PWS controlled bleedoff, then excess leakage can exist. Load transients or any perturbation causing a change in RCS temperature masks any leakage that might be present. Other methods are available to the control room staff to determine actual leakage values.

The charging pumps are positive displacement pumps with three plungers (triplex). Each pump is driven by a 480 volt ac, 100-horsepower motor. Each pump has its own integral leakage collection system. The internals of the pumps are austenitic stainless steel. Each charging pump has its own seal lubrication provided by a separate pump with its own subsystem.

Suction and discharge dampeners are provided to reduce pulsations or vibrations produced by the plungers. The discharge dampener N_2 charge (or overpressure) can be adjusted to lower values than normal when the charging pumps are used during low RCS pressure conditions. The charging pumps suction header can be supplied from any of the following sources:

- VCT
- Blending tee
- Refueling water tank (RWT)
- Boric acid makeup tank (BAMT)

Each charging pump suction piping is equipped with a thermal relief valve that lifts if the suction pressure reaches 150 psig. Charging pump discharge relief valves provide overpressure protection for the positive displacement pumps. These valves all lift at 2,735 psig.

A low oil pressure alarm, set at 10 psig, is provided for each charging pump. These alarms are also actuated by the associated hand switch being in the stop position or by the control circuit being de-energized.

The charging pumps are controlled by hand switches. The swing charging pump can be powered from either the red train or the green train and subsequently has two hand switches. These hand switches are three-position switches.

Normally, only one pump has its hand switch in the start position. The other two pump hand switches are in the auto position, and the pumps cycle as needed depending on the pressurizer / level. Another hand switch, the charging pumps select switch, is used to designate the lead and standby pumps.

NOTE: Several plants have employed runtime meters, located on the charging pump/seal water pump control panel, to record actual charging pump operation. The associated runtime meter is energized when the charging pump receives a start signal. These meters can be used by maintenance to determine the total runtime or by operations to determine the amount of time that a charging pump has run since starting.

In addition to the hand switches, each charging pump has a local hand switch at each pump breaker.

Normally, in remote mode, the following occurs to the associated pump if the switch is in local mode:

- Remote indication and control is lost.
- The automatic start features from the pressurizer level control system and the safety injection activation signal (SIAS) are removed.
- The normal sequencing for loading of this particular charging pump onto the emergency diesel generator is bypassed.
- The runtime meter is not energized on pump start.
- A control room alarm is activated in the control room indicating that the charging pump hand switch is **not** in remote mode.

The charging pumps are powered from vital 480 V ac motor control centers. In addition to an automatic start command from the pressurizer level control system, each charging pump automatically starts if the following signals are received:

- The charging pumps automatically stop if the pressure falls to less than 10 psia with no SIAS signal present.
- If an SIAS signal is present, the charging pumps continue to run, regardless of the suction pressure.

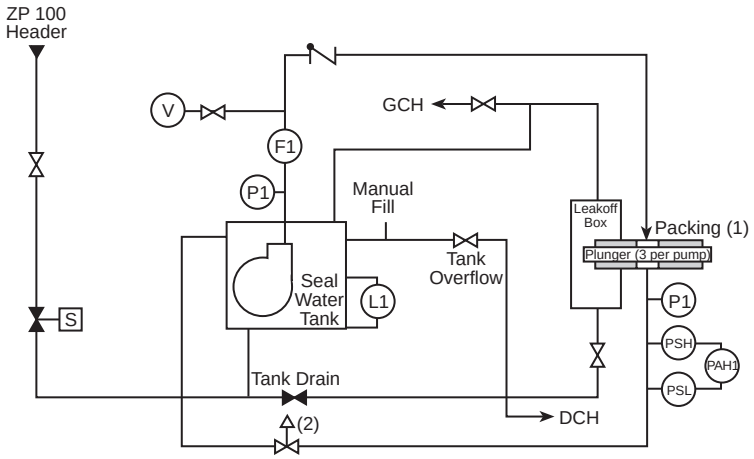
The discharge of the charging pumps has a flow tap and a pressure tap to allow monitoring of the system. The pressure readings can be monitored, and an alarm is generated when the charging pump discharge pressure drops below 2000 psig. The charging pump discharge flow can also be monitored, and an alarm is generated when the discharge flow decreases to less than 25 gpm.

Charging Pump Seal Lubrication System

As shown in Figure A-3, each charging pump has its own charging pump seal lubrication system. The seal lubrication pumps take suction from the seal water tank. Two pressure sensors are provided in each system to check for inner and outer seal failures. These transmitters send a signal to an alarm switch. If the pressure is < 5 psig or > 15 psig, a control room alarm will activate.

The sensors tap in on the seal return side of the charging pump seal. A high-pressure condition indicates inner packing failure and a low-pressure condition indicates an outer packing failure. “Inner” and “outer” refer to the side of the seal water injection ring on which the packing is located.

Makeup to the seal water tank is via a solenoid valve actuated by a level switch mounted on the tank. When the seal water tank level is low, the valve opens to supply makeup water from the RWT. The solenoid valve isolates makeup water just before the water reaches the tank overflow level.



(1) Provided with high-pressure and-low pressure packing. Low-pressure packing is in outer set. Plunger leakoff indication provided by plexiglass cover on top of leakoff box.

(2) Valve is throttled to maintain back pressure.

Figure A-3
Charging Pump Seal Lubrication System

The seal lubrication pumps are controlled by three-position hand switches. When these switches are in the auto position, the seal water pump is interlocked to start when its respective charging pump starts. When the switch is in the hand position, the seal water pump runs continuously. The relay that starts the seal water pump in auto also starts the charging pump runtime meters. The third position on the switches is stop.

Charging to HPSI Header Isolation Control Valve

Should the normal charging header become unavailable for any reason, an alternate charging discharge flow path to high-pressure safety injection (HPSI) header can be used. This connection can also be used for testing of the safety injection check valves into the RCS.

Charging Line Isolation Control Valve

The charging line isolation control valve is provided to isolate the discharge of the charging pumps. During normal operation, the valve is in the locked open position.

After passing through the regenerative heat exchanger, the charging flow can be directed through three different paths. Two of these paths route the charging flow directly back to the RCS and the third allows water to be directed to the top of the pressurizer and can be used as an alternate method to reduce RCS pressure.

RCS Charging Line Isolation Control Valves

Normal charging flow back to the RCS passes through two motor-operated valves. One control valve isolates the charging header for each RCP discharge.

In parallel with these valves is a spring-loaded check valve. The check valve is designed to provide a backpressure of 200 psi on the charging header if both motor-operated isolation valves are closed. This backpressure would force flow through the seal injection during RCS fill. It also provides a charging pump discharge flow path if a charging pump is started with both normal loop flow path isolation valves closed, and provides a thermal relief path if charging line isolation valves are shut with hot letdown flow continuing.

Chemical Addition System

The chemical addition system, shown in Figure A-4, provides makeup and a means to add chemicals to the reactor coolant system, spent fuel system, and the refueling water tank. The ability to minimize corrosion and control reactivity is a function of the system.

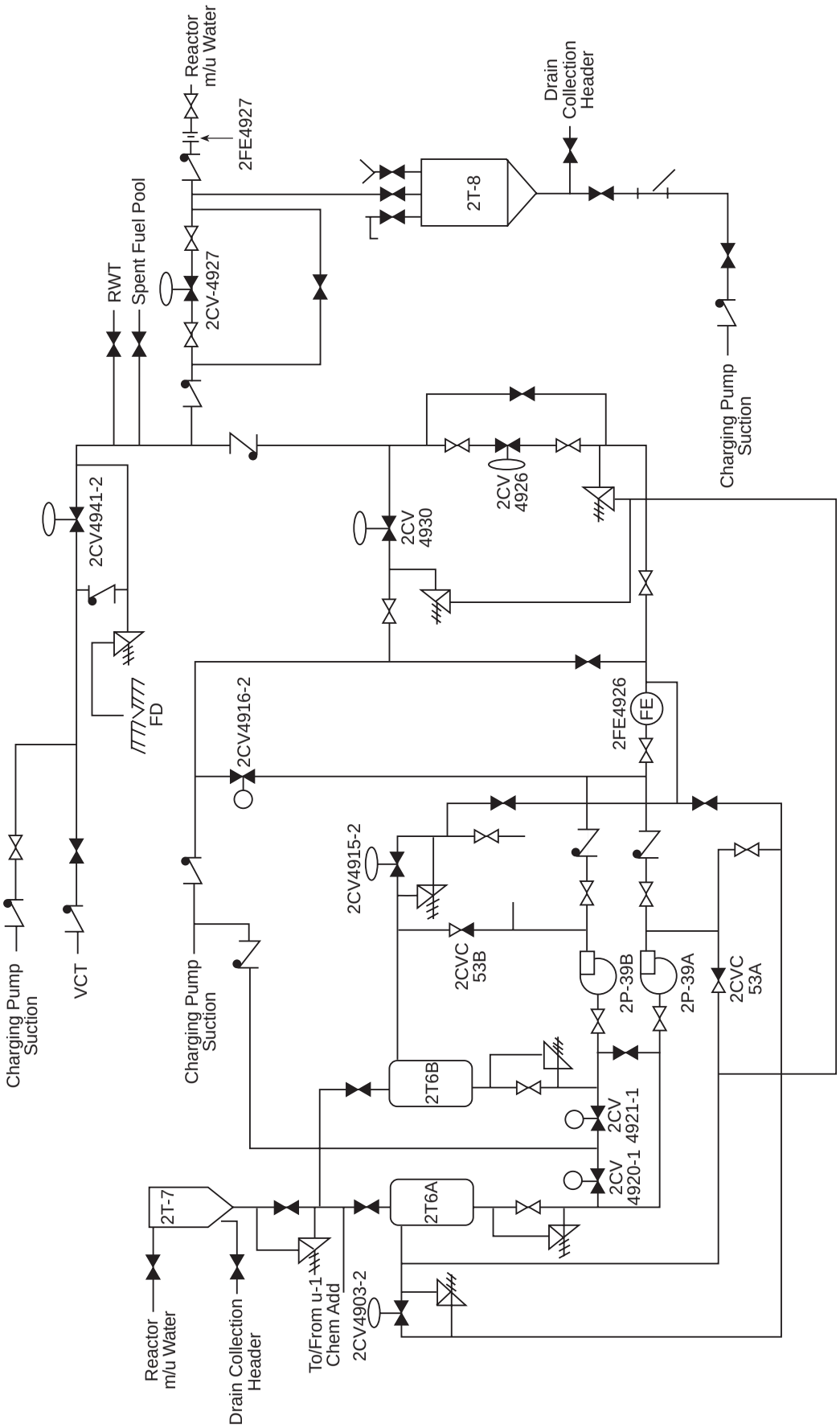


Figure A-4
Chemical Addition System

The chemical addition system works in conjunction with the charging portion of the CVCS. One of the suction sources listed for the charging pumps is the chemical addition system. The chemical addition system provides a means to add chemicals to the RCS and auxiliary systems. The following chemicals are added to the reactor coolant system:

- Boric acid is added to the RCS for reactivity control. Boration is required during a shutdown for refueling, maintenance, cooldown, or other reasons to ensure that the required shutdown margin[4] is maintained. Boration can be required during normal operations to maintain required control element configurations. Changes in boron concentration are required during a fuel cycle to compensate for fuel depletion and fission product poisons buildup in the core. Boric acid is also injected into the RCS by the emergency core cooling system (ECCS) when RCS pressure falls below the allowable emergency safety features actuation system (ESFAS) setpoint on two out of four channels. Boric acid is also added to the following auxiliary systems/components:
  - — Spent fuel system
  - — Refueling water tank (RWT)
- Lithium hydroxide is added for pH control.
- Hydrazine is added for oxygen control at low temperature.
- Hydrogen is added for oxygen control when the plant is at normal operating temperatures. Hydrogen is added to the RCS indirectly by providing a hydrogen overpressure on the VCT.

The chemical addition system also provides a means for mixing and storing chemicals.

Boric Acid Batch Tank

The boric acid batch tank is a 630 gallon tank that is used for the preparation of concentrated boric acid solutions. The batch tank is designed to permit mixing and transferring of up to 12 weight percent (wt%) boric acid solution.

When mixing a batch of boric acid solution, drums of dry boric acid powder are added in the desired proportions to heated demineralized water that has been added to the batch tank from the reactor makeup water tank. Three 15 kW electric immersion heaters are provided to heat the water temperature in the tank to 150°F. Heating the water before adding the boric acid facilitates the mixing of the boric acid crystals and prevents any precipitation of crystals in the tank.

A line is provided at the bottom of the batch tank to the drain collection header for flushing purposes. The tank is provided with an overflow line that directs the flow to a floor drain.

[4] Shutdown margin is defined by technical specifications as the instantaneous amount of reactivity by which the reactor is subcritical or would be subcritical from its present condition assuming all control element assemblies are fully inserted except for the single assembly of highest reactivity worth which is assumed to be fully withdrawn.

After mixing a batch of boric acid solution, the contents of the tank can be sent by gravity drain to the following locations:

- Boric acid makeup tanks via the boric acid batching tank strainer
- Boric acid mixing tank

A strainer removes particles prior to discharge from the tank. The strainer consists of a wire mesh cloth lining in a perforated basket. The strainer is electrically heat traced and insulated to prevent boric acid from crystallizing and obstructing the flow.

Boric Acid Makeup Tanks

The boric acid makeup tanks (BAMT) are used to store the boric solution used to borate the RCS in accordance with the technical specifications.[5] The BAMTs are designed to permit storage of boric acid solutions in concentrations of up to 3.5 weight percent.

Each tank has a capacity of 11,700 gallons, and the level is normally maintained greater than 80% in each tank. The level in each tank is monitored by a level-indicating transmitter.

Each tank is insulated and is provided with two independent, full capacity, strap-on heater banks consisting of three heaters per bank. Redundant heaters are provided to ensure that the borated water source for the RCS is maintained above 55°F.

Each boric acid makeup tank is vented to the gas collection header. Reactor makeup water can be sent to the tanks for flushing purposes. Each tank overflows through a loop seal to the drain system. Although the BAMTs can provide a suction source directly to the charging pumps' suction line, under normal operation the boric acid is routed from the tanks to the suction of the boric acid makeup pumps.

Gravity Feed Control Valves

These motor-operated gate valves provide a boric acid gravity feed flow path from the BAM tanks to the suction of the charging pumps. This path would be utilized if the BAM pumps were incapable of functioning and emergency boration of the RCS was required.

[5] For Modes 1, 2, 3, or 4, technical specification LCO 3.1.2.8 states that the required volume of boric acid solution must be maintained greater than 55°F. The volume of boric acid required in the BAMTs is dependent on the stored BAMT concentration, in weight percent of boric acid, and the RWT boron concentration in PPM. Refer to TS 3.1.2.7 Figure 3.1-1 on page 3/4 1-14 of the technical specifications. In Modes 5 and 6, either the RWT has to be operable or one BAMT with a minimum contained water volume of 3,400 gallons (equivalent to 31% tank level), a boric acid concentration between 2.5 weight percent and 3.5 weight percent, and a minimum solution temperature of 55°F.

Boric Acid Makeup Pumps

Two boric acid makeup (BAM) pumps are provided to send a boric acid solution from the BAMTs to various locations. These pumps receive an automatic start signal from the SIAS. The discharge of the BAM pumps can be routed through the motor-operated emergency boration control valve directly to the suction of the charging pumps. The pump discharge can also be routed to the following locations:

- Directly to the spent fuel pool
- Directly to the refueling water tank
- Through the CVCS blend control valve (CV) to the volume control tank
- Through the control valve (CV) to the charging pump's suction line

Another flow path for concentrated boric acid is directly to the charging pump suction header through a line that taps off just downstream of the boric acid flow element.

The pumps have a 102 psi shutoff head and require a minimum net positive suction head (NPSH) of 2.75 psi. The annunciator is activated when the discharge pressure of either pump falls below approximately 85 psig. The capacity of each pump is greater than the combined capacity of all three charging pumps.

Although both BAM pumps are powered from the same electrical train, redundancy is provided for the system by gravity feed valves that are both powered from the other train.

Hand switches located on the control room panel are used to control BAM pumps. The BAM pump select switch is used to select which pump will start automatically when called upon by system demands. The boric acid makeup pumps and the associated piping are heat traced to maintain boric acid in solution and are insulated for efficiency and personnel protection.

The boric acid makeup pumps can also be used to recirculate the contents of the boric acid makeup tanks and to transfer the contents of one BAMT to the other. A recirculation line is installed from the discharge of each BAM pump back to each BAMT. Each line is equipped with an air-operated control valve (see the discussion on these valves immediately following this section) that closes upon receiving an SIAS signal from the emergency safety feature actuation system (ESFAS). Locked throttle valves are provided as a bypass around each air-operated control valve to provide recirculation when the air-operated valve is closed. A pressure relief valve is provided as a second bypass around the air-operated control valve to allow recirculation if the line with the locked throttle valve does not provide enough flow. The pressure relief valves are set at 150 psig.

Recirculation Control Valves

CVs are used for normal recirculation of boric acid from the discharge of the BAM pumps to the BAM tanks. These air-operated valves are sized to accommodate the excess flow from the BAM pumps that is not being used for normal boric acid addition. These globe valves fail closed on loss of instrument air. Direct valve position indication and control hand switches are located on the control room panel.

Emergency Borate Control Valve

A motor-operated gate control valve provides a path for the boric acid solution from the discharge of the BAM pumps directly to the suction of the charging pumps for emergency boration.

Boration Control Valve

The boration control valve is the normal boration valve that is used to admit boric acid to the suction of the charging pumps and fails closed on loss of air. The valve is controlled by a three-position hand switch. Located upstream from the CV is a relief valve designed to protect the downstream piping from overpressure.

Boric Acid Makeup Flow Control Valve

The boric acid makeup flow control valve is used to control the flow rate of boric acid from the BAM pumps during boration or blending operations. The flow through the CV can be directed to the CV or the blending tee to be mixed with unborated water from the reactor makeup water tank prior to being added to the CVCS, spent fuel pool, or the refueling water tank.

This air-operated globe valve has a solenoid-operated pilot valve. The operation of the valve depends on the four-position makeup mode select switch, as follows:

- A flow recorder is located on the panel.
- A flow totaling integrator, which provides the operator with an indication of the total gallons of borated water added, is located on the panel.
- An preset batch switch, which terminates the addition of borated water in the BORATE mode after a preset amount of borated water has been added, is located on the panel.
- A flow switch, which activates a warning when the flow deviates 2 gpm from the setpoint desired by flow controller, is located on the panel.

Reactor Makeup Water Pumps

The reactor makeup water pumps are used to provide unborated water from the reactor makeup water tank to the RCS via the charging system.

Each of these pumps can be started manually from the control room panel by placing the appropriate control switch in the start position. Either pump can be selected to start automatically by positioning the pump select switch on the panel to the desired pump. If the selected pump hand switch is in the auto position, the pump is started and stopped by the makeup mode selector switch, also located on the panel.

In order to minimize the effects of water hammer, the discharge valve on the pump to be started is throttled prior to starting. For this reason, the automatic start and stop feature is not used; one of the pumps is normally left running in manual mode.

Reactor Makeup Water Flow Transmitter

The reactor makeup water flow transmitter is an orifice-type flow element provided to measure the reactor makeup water pump discharge flow rate.

Reactor Makeup Water Flow Control Valve

This control valve controls the flow of reactor makeup water from the RMW pumps to the blending tee during any dilution or blending makeup of the VCT.

VCT Makeup Control Valve

The VCT makeup control valve is located downstream of the blending tee that receives water from the BAM flow control valve and/or the reactor makeup water flow control valve. When the VCT makeup control valve is open, the operator can provide BAM pump discharge, RMW pump discharge, or the desired combination of both to either the VCT or directly to the suction of the charging pumps by opening the respective manually operated downstream valves.

The manually operated valves are normally positioned to allow makeup flow to go directly to the suction of the charging pumps. The VCT makeup control valve is an air-operated globe valve that fails closed on loss of air or power to the control circuit. The valve is controlled by a hand switch located on the control room panel.

Redundant electrical heat tracing is installed on all piping, valves, pumps, and other components that might contain concentrated boric acid solutions for a significant period of time. Heating these lines is necessary to maintain the boric acid in solution. Thermal insulation is provided for the conservation of heat and protection of personnel. The heat tracing is appropriately sectionalized, taking into account system redundancy requirements and components that might be removed from the system for maintenance.

Thermal relief valves are provided for those portions of the chemical addition system that have the capability to be isolated from the rest of the system. Each relief valve is sized to relieve the maximum thermal expansion rate in the event that maximum duplicate heat tracing power was applied to the isolated line. The heat tracing system has its own local panels for alarm and indicating purposes.

NOTE: Boration/Dilution Concerns: The boric acid addition or dilution required to reach equilibrium conditions depends on the flow path used and the number of charging pumps in operation. With two charging pumps running, approximately 20 minutes is required before the boron concentration in the reactor coolant system stabilizes at the new value. When added to the suction of the charging pumps with two charging pumps running, approximately two minutes is required before the concentration in the RCS is stabilized. Normally, addition of boric acid or dilution is performed via a line up to the charging pump suction with one charging pump running. In this case, approximately four minutes is required to reach equilibrium conditions. If blending at power, the charging alignment would normally be re-aligned to the VCT to minimize the effects of inaccuracies in the blend ratio setup.

Chemical Addition Tank

The chemical addition tank provides a means to inject chemicals into the reactor coolant system via the charging pump suction header. Chemical are added for pH and oxygen control in the RCS. Any chemical additions to the RCS are considered possible reactivity manipulations.

Chemicals from the tank are sent through the chemical addition strainer before being sent to the charging pump suction header.

Reactor Makeup Water System

The reactor makeup water system is provided with water from the reactor makeup water tank. This stainless steel tank has a capacity of 180,000 gallons and supplies water to the suction of the reactor makeup water pumps. These pumps are designed to supply 200 gpm to the RMW header. The RMW header supplies demineralized water to the following:

- Boric acid batch tank
- NaOH tank makeup
- Charging pump seal water tank
- Boric acid concentrator / waste concentrator
- Resin addition water
- Resin discharge header
- Resin flush header
- Resin sluice header
- Postaccident sample system flushing demineralized water stations
- Containment building demineralized water

Makeup for the tank normally comes from the secondary vacuum.

Associated Procedures

Operating Procedure Chemical and Volume Control

The CVCS operating procedure provides instructions for startup, normal operation, shutdown, and testing of components. Testing in accordance with this procedure satisfies the testing requirements of parts of the technical specifications.

Typical Limits and Precautions

- Do not admit hot RCS fluid (> 140°F) to the regenerative heat exchanger until the charging flow is established.
- Do not admit hot letdown fluid (> 140°F) to the letdown heat exchanger until the component cooling water flow is established.
- Do not exceed a letdown flow of 128 gpm.
- Placing an ion exchanger in service can change the RCS boron concentration.
- When possible, the seal water system should be in service (30 minutes) prior to starting the charging pump.
- During normal operations, the letdown pressure should be maintained at < 140°F for ion exchanger resin protection.
- For some plants, piping downstream of the letdown heat exchanger is not qualified for temperatures > 150°F.
- Oscillations in charging pump flows can be caused by any of the following:
  - Improper operation of pulsation dampeners
  - A VCT level indication error with the actual level low
  - Failure of an internal check valve on the charging pump in service
  - Gas binding of charging pumps due to improper filling and venting
- During an event that demands local control of the charging pumps, information regarding RCS pressure is not directly available to the operator who has local control.
- With the local / remote control switch for a charging pump in the local position, any automatic start feature, such as pressurizer level or SIAS, is lost.
- When diverting to the vacuum degassifier, ensure that at least one H_2O_2 analyzer is in service and aligned to the waste gas holdup system or that grab samples are taken and analyzed every four hours per technical specifications.
- Each charging pump is required to have its associated room cooler operable in order for the pump to be declared operable.
- If a charging pump or charging header has been drained, perform the applicable portion of the operating procedure to vent the suction stabilizer.

- When a charging pump is started locally from the breaker, the charging pump could be damaged due to all permissives being bypassed. When a pump is started locally, it cannot be secured from the control room.
- When more than one charging pump is running, short-lived activity radiation levels will increase.

Abnormal Operating Procedures

The chemical and volume control system is used in various abnormal modes of operation. The goal of these procedures is to restore the affected equipment or system to operation or to place the plant in a stable condition. The three operating procedures of concern are excess RCS leakage, moderator dilution, and loss of charging.

Excess RCS Leakage

This procedure stabilizes the plant and attempts to locate and isolate the leak in the RCS. If the leak can be isolated or is within the leak requirements of the technical specification, plant operation can continue. If the leak is greater than these technical specification limits, a plant shutdown is required. Excess RCS leakage is caused by a break in the RCS pressure boundary. The break can be as small as a 0.03 gpm increase in RCS leak rate calculation.

Loss of Charging

This procedure stabilizes the plant, determines the cause of the event, and attempts to restore charging via the normal flow path (if available) or through the high-pressure safety injection system header. If a plant shutdown is required, instructions are provided to perform a controlled shutdown to reach a condition that will allow repair of the charging system. This procedure is entered if any of the following conditions exist:

- The HEADER FLOW LO annunciator is received.
- The HEADER PRESS LO annunciator is received.
- Low charging header flow is indicated.
- The charging header pressure is less than the RCS pressure.

The exit conditions ensure that either normal charging is restored or another procedure is entered that addresses a loss of normal charging. The procedure is exited if any of the following conditions exist:

- The charging system is restored to normal.
- The Standard Posttrip Actions operating procedure is entered.
- The Emergency Boration operating procedure is entered.

Technical Specifications

Reactivity Control

The shutdown margin shall be greater than or equal to that specified in the core operating limits report (COLR in the technical specifications). Technical specification applies in Modes 1–5. If the shutdown margin is less than required, immediately initiate and continue boration at > 40 gpm using 2,500 ppm boric acid solution or equivalent until the required shutdown margin is restored (Emergency Borate). A sufficient shutdown margin ensures that:

- The reactor can be made subcritical from all operating conditions.
- The reactivity transients due to any postulated accident conditions are controllable within acceptable limits.
- The reactor will be maintained sufficiently subcritical to preclude inadvertent criticality in the shutdown condition.

RCS Flow Rate

The flow rate of reactor coolant through the reactor coolant system shall be > 2,000 gpm whenever a reduction in reactor coolant system boron concentration is being made. If the flow rate in the RCS is less than 2,000 gpm, suspend all operations involving a reduction in boron concentration in the RCS.

A flow rate of at least 2,000 gpm ensures adequate mixing, prevents stratification, and ensures that reactivity changes will be slow during boron concentration reductions in the RCS. A flow rate of at least 2,000 gpm will circulate an equivalent RCS volume of 6,650 cubic feet in approximately 25 minutes. The reactivity change rate associated with boron concentration reductions will be within the capability of operator recognition and control.

Boration Systems

The boration injection system ensures that negative reactivity control is available during each mode of plant operation. In Modes 1-4, a minimum of two separate and redundant boron injection systems is provided to ensure single functional capability if an assumed failure renders one of the systems inoperable. Allowable out-of-service periods allow minor component repair, and corrective action can be completed without undue risk to the overall plant from injection system failures during the repair period.

The boration capability of either system is sufficient to provide shutdown margin from the expected operating conditions after xenon decay and cooldown to 200°F. The maximum expected boration capability requirement occurs at end-of-life from full power equilibrium xenon conditions and requires boric acid solution from the boric acid makeup tanks in the allowable concentrations and volumes.

In Modes 5 and 6, one injection system is acceptable without single failure consideration on the basis of the stable reactivity condition of the reactor. Restrictions prohibit core alterations and positive reactivity changes if the single injection system becomes inoperable. In Modes 5 and 6, the boron capability is based upon providing a sufficient shutdown margin after xenon decay and cooldown from 200°F to 140°F.

Boron Injection Flow

As a minimum, one of the following boron injection flow paths shall be operable in Modes 5 and 6:

- A flow path from the boric acid tank via either a boric acid makeup tank or a gravity feed connection and a charging pump to the RCP if only the boric acid makeup tank is operable
- A flow path from the refueling tank via a charging pump or high-pressure safety injection pump to the RCS if only the RWT is operable

If none of the above flow paths are operable, suspend all operations involving core alterations or positive reactivity changes until at least one of the flow paths is operable. The operability of the RWT and BAMTs for Modes 5 and 6 are discussed in the next section.

Borated Water Sources

In Modes 5 and 6, a minimum of one of the following borated water sources shall be operable:

- One boric acid makeup tank with:
  - A minimum contained borated water volume of 3,400 gallons (31%)
  - A boric acid concentration between 2.5 wt% and 3.5 wt%
  - A minimum solution temperature of 40°F
- The refueling water tank with:
  - A minimum contained volume of 61,370 gallons (7.5%)
  - A minimum boron concentration of 2500 ppm
  - A minimum solution temperature of 40°F

If none of the above borated water sources are operable, suspend all operations involving core alterations or positive reactivity changes until at least one of the borated water sources is operable.

Charging Pumps

At least one charging pump in the boron injection path required to be operable by the technical specifications shall be operable and capable of being powered from an operable emergency bus in Modes 5 and 6. If no charging pump is operable, suspend all operations involving core alterations or positive reactivity changes until at least one of the required pumps is restored to operable status.

Boric Acid Makeup Pump

At least one boric acid makeup pump shall be operable and capable of being powered from an operable emergency bus if only the flow path through the boric acid makeup pump is operable. With no boric acid makeup pump operable as required by technical specification, suspend all operations involving core alterations or positive reactivity changes until at least one boric acid pump is restored to operable status.

Boron Injection Paths

The following boron injection paths shall be operable depending on the volume available in the boric acid makeup tanks in Modes 1-4:

- If the contents of one boric acid makeup tank meet the volume requirements of the technical specifications, two of the following flow paths to the RCS shall be operable:
  - One flow path from the appropriate BAMT via a boric acid makeup pump and a charging pump
  - One flow path from the appropriate BAMT via a gravity feed connection and a charging pump
  - One flow path from the RWT via a charging pump
- If the contents of both boric acid makeup tanks meet the volume requirements of the technical specifications, four of the following five flow paths to the RCS shall be operable:
  - One flow path from BAMT A via a boric acid makeup pump and a charging pump
  - One flow path from BAMT B via a boric acid makeup pump and a charging pump
  - One flow path from BAMT A via a gravity feed connection and a charging pump
  - One flow path from BAMT B via a gravity feed connection and a charging pump
  - One flow path from the RWT via a charging pump

With any of the boron injection flow paths to the RCS required above inoperable, restore the inoperable flow path to the RCS to operable within 72 hours, or be in Mode 3 and borated to a shutdown margin equivalent to at least that specified in the Core Operating Limits Report for 200°F within six hours; restore flow paths to operable status within the next seven days, or be in Mode 5 within the next 30 hours.

Borated Water Sources

In Modes 1-4, each of the following borated water sources shall be operable:

- At least one of the following sources with a minimum solution temperature of 55°F:
  - One boric acid makeup tank with the tank contents in accordance with the technical specifications
  - Two boric acid makeup tanks with a combined contents in accordance with the technical specifications
- The refueling water tank with:
  - A contained borated water volume between 91.7–100%
  - Between 2500–3000 ppm boron concentration
  - Solution temperature between 40–110°F

With the above required boric acid makeup tank(s) inoperable, restore the makeup tank(s) to operable within 72 hours or to 200°F within six hours; restore the above required makeup tank(s) to operable status within the next seven days, or be in Mode 5 within the next 30 hours. With the refueling water tank inoperable, restore the tank to operable status within one hour, or be in Mode 3 within the next six hours and Mode 5 within the following 30 hours.

Charging Pumps

In Modes 1-4, at least two charging pumps shall be operable. With only one charging pump operable, restore at least two charging pumps within 72 hours or be in Mode 3 and borated to a shutdown margin equivalent to at least that specified in the COLR for 200°F within six hours; restore at least two charging pumps to operable status within the next seven days, or be in Mode 5 within the next 30 hours.

Boric Acid Makeup Pump

In Modes 1-4, at least the boric acid makeup pumps in the boron injection flow paths, required operable in technical specifications, shall be operable and capable of being powered from an operable emergency bus if the flow path through the boric acid makeup pumps in the technical specifications is operable.

With only one boric acid pump required for the boron flow paths, restore the boric acid makeup pump within 72 hours, or be in Mode 3 and borated to a shutdown margin equivalent to at least that specified in the COLR for 200°F within six hours; restore at least two charging pumps to operable status within the next seven days, or be in Mode 5 within the next 30 hours.

B

WESTINGHOUSE PLANT CVCS

The chemical and volume control system (CVCS) plays an important role in maintaining safe reactor coolant system (RCS) operation under both normal and abnormal plant conditions. Safe operation under normal conditions is accomplished through a variety of functions provided by the CVCS. RCS water chemistry is maintained below the limits stated in the technical specifications. RCS water inventory control is provided to ensure that the reactor core has enough cooling water for adequate heat transfer following any Condition I or II events. Reactivity control is managed by the operator, either manually or automatically, by making changes in the reactor coolant pump operation. Cooling water for the reactor coolant pump's seals is provided by the CVCS.

Under abnormal plant conditions (loss of coolant accidents, main steamline breaks, and so on), a portion of the CVCS is considered part of the plant's emergency core cooling system (ECCS). This portion provides the high head injection of water to the reactor core and reactivity control following certain accident situations, as identified in the Final Safety Analysis Report.

The primary purposes and functions of the CVCS, as shown in Figure B-1, are to:

- Maintain proper water inventory in the pressurizer (that is, the reactor coolant system) from 24.7% level at zero power to 60% level at 100% power
- Supply seal water flow to all four reactor coolant pumps (RCPs)
- Control RCS water chemistry conditions, activity level, soluble chemical neutron absorber concentration (boron), and water makeup
- Process excess reactor coolant to affect recovery and reuse of boric acid and primary makeup water
- Provide the high head injection portion of the emergency core cooling system

Secondary functions of the CVCS are to:

- Process and use excess letdown
- Provide auxiliary spray for pressure control of RCS
- Combine with residual heat removal (RHR) system to provide a flow path and pressure control during solid plant operations

NOTE: A general system description is provided in this appendix. Actual plant systems and system operation vary from plant to plant. This description is intended to identify system modes of operation that affect charging pump operation and performance.

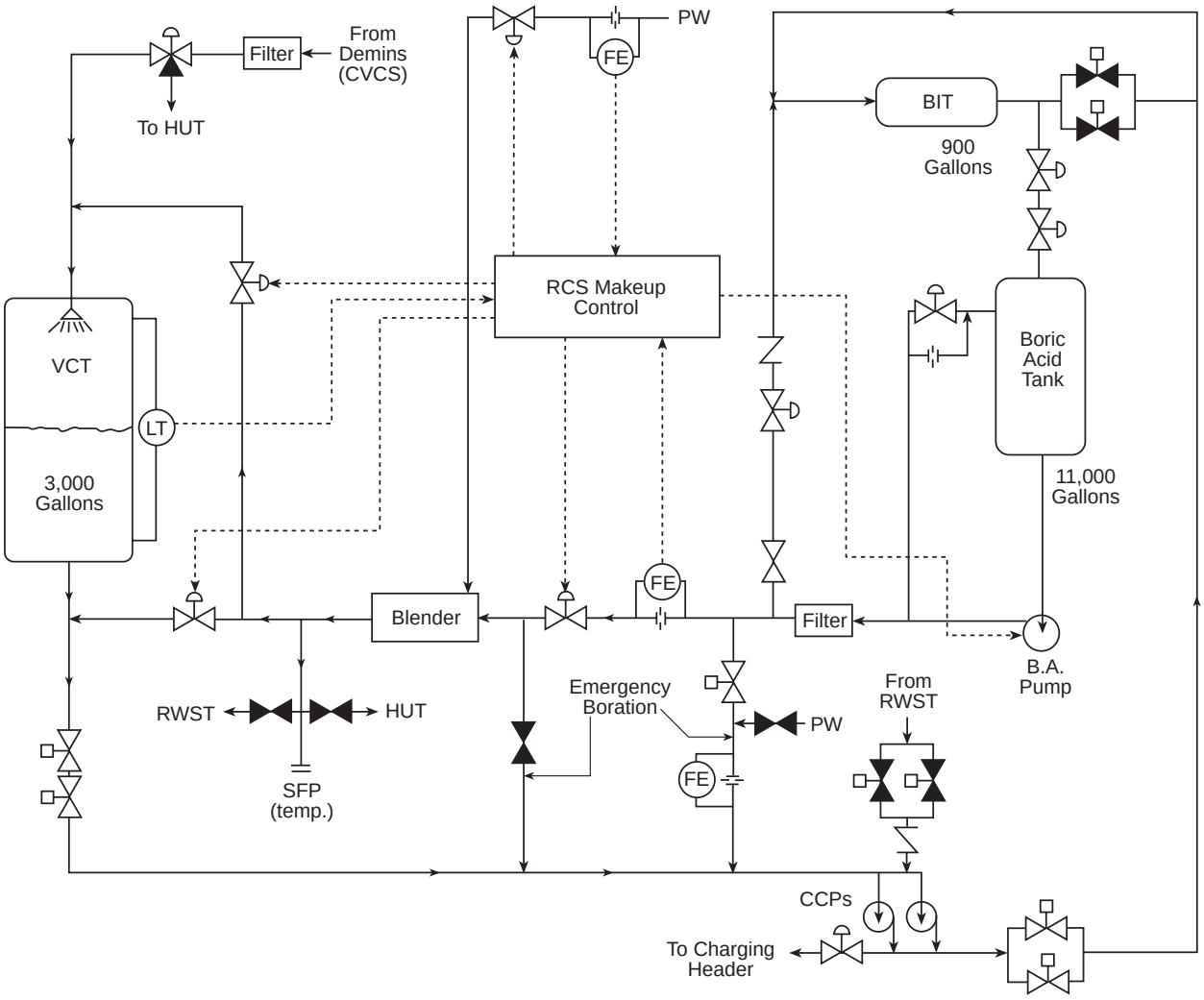


Figure B-1
Chemical Volume and Control System Flow Path

Summary

The CVCS is one of the most important systems at the plant because it has both a normal operation and an emergency dilution purpose. During normal operation, the CVCS functions to control RCS chemistry and volume, as well as to supply seal flow to the reactor coolant pumps. In an emergency situation, the CVCS still supplies seal flow, but it also acts as a key part of the emergency core cooling system to supply high head injection to the RCS.

System Flow Paths

Normal Power Operations (Let Down and Charging)

The system flow paths are discussed below and flow path diagrams are shown in Figures B-2 through B-4.

- Letdown is from the RCS loop-3 crossover leg (3-inch line).
- Regenerative heat exchanger, located in containment. One per unit. Three separate shell-and-tube type exchangers. The cooling medium is charging flow. The letdown is cooled from 545 to 290°F. The heat transfer rate is 10.3×10^6 BTU / hr.
- Letdown orifices. Two orifices are set for 75 gpm and one for 45 gpm at psig pressure drop.
- Letdown orifice isolation valves. These close on no central charging pump (CCP) running, specific letdown orifice isolation valves closed, or low pressurizer level.
- Letdown relief valve. This valve opens at 600 psig. This valve is used to protect the letdown line from the orifices to the pressure control valve (PCV).
- Containment isolation valve.
- Supply line from residual heat removal and flow control valve (FCV). This line is put in service for cleanup of residual heat removal / RCS during cooldowns and refueling and for connecting with CVCS for solid plant operations.
- Letdown heat exchanger: One per unit; cooled by CCS. The heat transfer rate is 14.8×10^6 BTU / hr. The normal outlet temperature is approximately 127°F.
- The letdown heat room itself has a resistance temperature detector installed to detect a high-energy line break in the room. Annunciation in the main control room and a letdown heat room temperature of 144°F alerts the operators that a line break might exist in the letdown heat room.
- Backpressure control valve. This valve is used to maintain a backpressure of normally 300–350 psig to prevent flashing on the downstream side of the letdown orifices.
- Relief valve. Protects demineralizers and piping downstream of pressure control valve. Lifts at 205 psig and relieves to the volume control tank.

- Three-way temperature control valve. At temperatures above 137.5°F, this valve switches position to bypass the mixed bed resins that break down at a high temperature. The valve is normally set so that the mixed beds are in service.
- Demineralizers:
  - Mixed bed demineralizers. Normally, one bed is in service; the other is in standby. They are used to remove fission products, corrosion products, and, as a filter, also lithium.
  - Cation bed demineralizer. One bed, normally **not** in service. It is placed in service when the pH or Cesium-137 activity is too high.
- Reactor coolant filter. One per unit. Usually in service, with a bypass line. Uses micron filter to collect resin fines and particulates.
- Three-way divert valve. The flow control valve is normally set to send flow to the volume control tank. If the volume control tank level gets too high, this valve diverts flow to the holdup tank.
- Makeup flow control valve to change boron concentration in RCS (dilutions).
- Volume control tank (VCT). The volume control tank is the main control point for the CVCS. Its volume is 400 cubic ft. At the entry point is a nozzle that sprays letdown flow through an H_2 blanket over the volume control tank contents. (H_2 is supplied via the pressure control valve.) The volume control tank is vented via the pressure control valve.
- Makeup valve. The flow control valve is another way of CVCS/RCS makeup. The makeup valve is similar to the flow control valve, although flow enters the system downstream of the volume control tank.
- Volume control tank outlet isolation valves. Flow control valves are in series. Normally open, they close on a safety injection (SI) signal or low-low volume control tank level.
- Seal water return line. This is the normal return for the reactor coolant pump seal leakoff; it is also used for a minflow recirculation line for the central charging pumps.
- Chemical feed line. This line is on the discharge of the chemical mixing tank and is used to put chemicals into the RCS.
- Alternate borate line. This line provides a boration path, and flow is through a manual valve.
- Emergency boration line. This line provides an emergency boration path for:
  - — Unexplained reactivity increase
  - — Failure of control rod to fully insert
  - — Uncontrolled cooldown after trip. The emergency boration line is normally used when called for by the technical specifications. Flow is through the motor-operated flow control valve.

- Safety injection tie-in. Two lines tied together, one from the residual heat removal pump and the other from SIS pump suction.
- Refueling water storage tank (RWST) supply line. This line provides water to the central charging pumps on SI actuation or when the volume control tank reaches a low-low level.
- Charging pump. There are three pumps, two are centrifugal and one is positive displacement. The charging pumps (variable speed pumps) are used for pressurizer level control and hydrostatic testing of the RCS. The two central charging pumps have a outlet discharge valve for pressurizer control and serve as high-head emergency core cooling system pumps.

NOTE: The number of pumps varies from site to site. Some plants have more positive displacement pumps than centrifugal pumps.

- Central charging pump (CCP) minflow lines. These lines are used to protect the CCPs from "dead-heading."
- Discharge flow control valve. The flow control valve acts to control charging flow in order to maintain pressurizer level. This valve always maintains at least 55 gpm flow.
- Seal flow line. Part of the central charging pump discharge flow is supplied to the reactor coolant pump seals.
- Seal water flow control valve. This flow control valve can be positioned to maintain proper seal flow to the reactor coolant pump seals.
- Charging isolation valves. Two series isolation valves ensure that charging flow is isolated.
- Auxiliary boration line. This line provides a means to borate the system in the event of flooding.
- Containment penetrations. A check valve provides isolation of the backflow coming out of the containment.
- Regenerative heat exchanger. Heats charging flow from 130 to 495°F. This minimizes the thermal shock to RCS piping and eliminates the reactivity effect due to cold water insertion of the charging flow.
- Normal charging. Normal charging flow is through a flow control valve to RCS cold leg No. 4.
- Alternate charging. Same as normal charging, but the valve is a flow control valve and penetration is to cold leg No. 1. A spring-loaded check valve relieves the volumetric expansion of coolant on the charging side of the regenerative heat exchanger to RCS in the event that the charging side of the heat exchange is isolated while hot letdown flow continues. The valve opens at approximately 75 psig.
- Pressurizer auxiliary spray. This used when reactor coolant pumps or normal paths are not available.

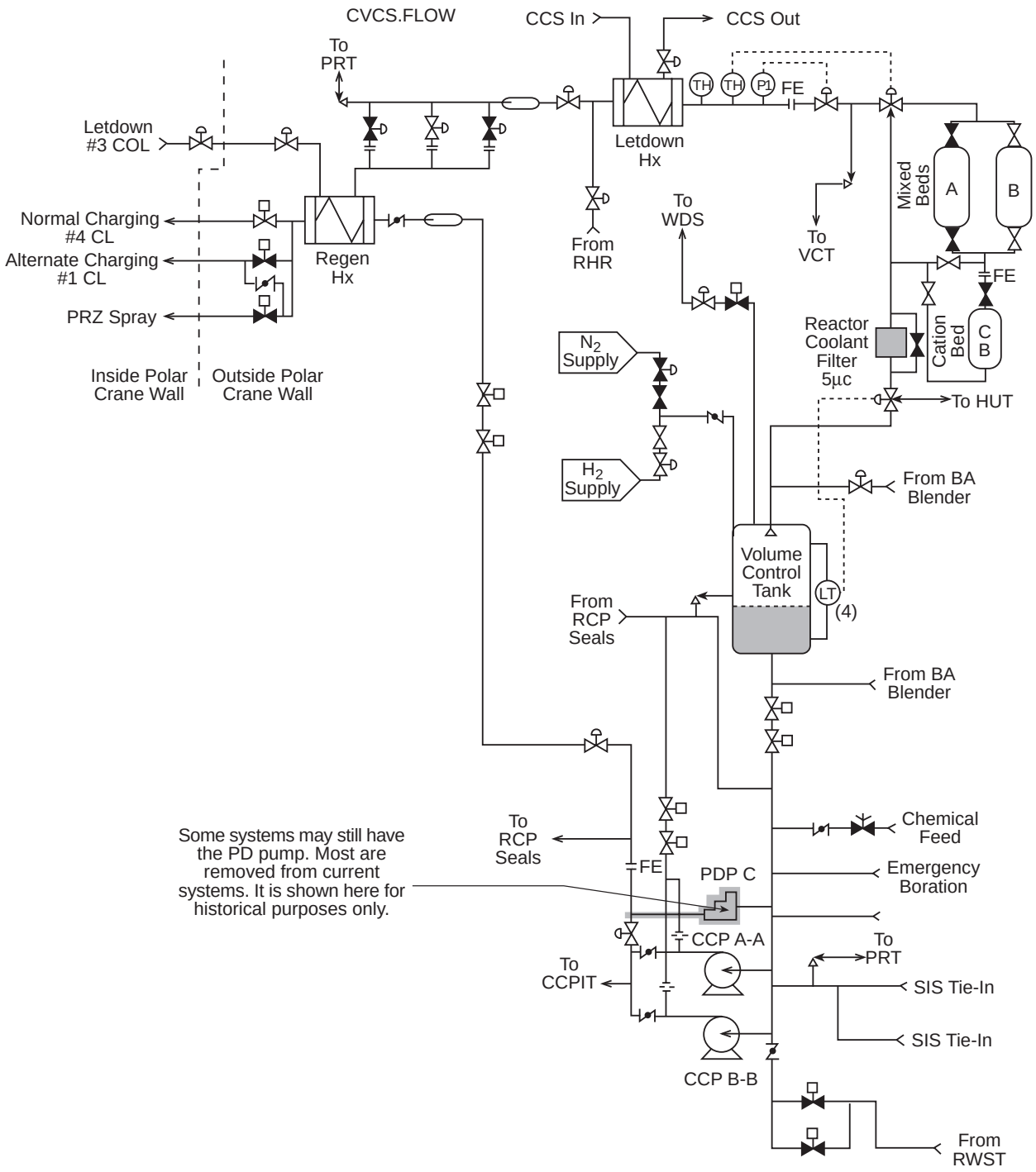


Figure B-2
Chemical Volume and Control System Normal Charging and Letdown Flow Path

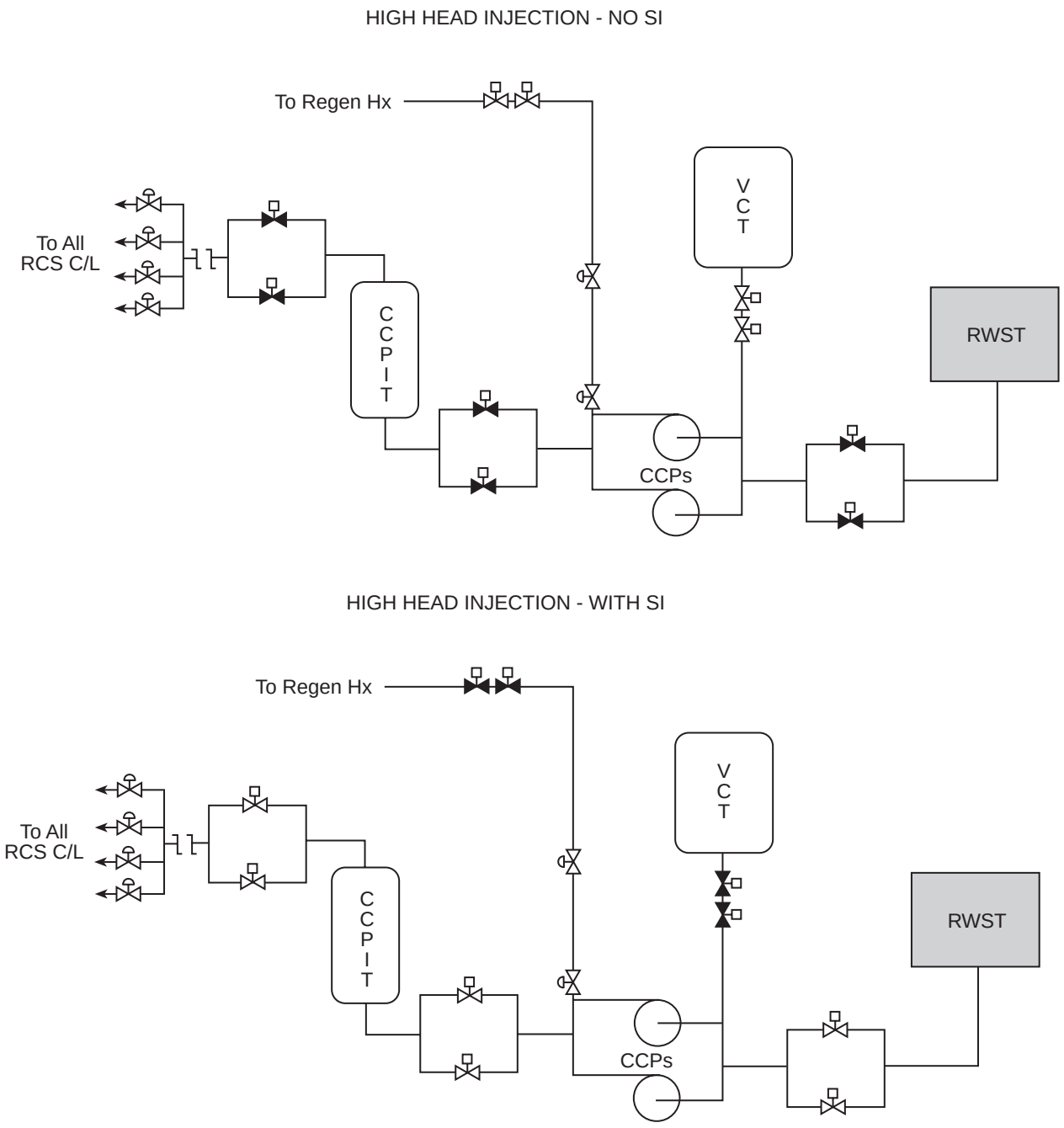
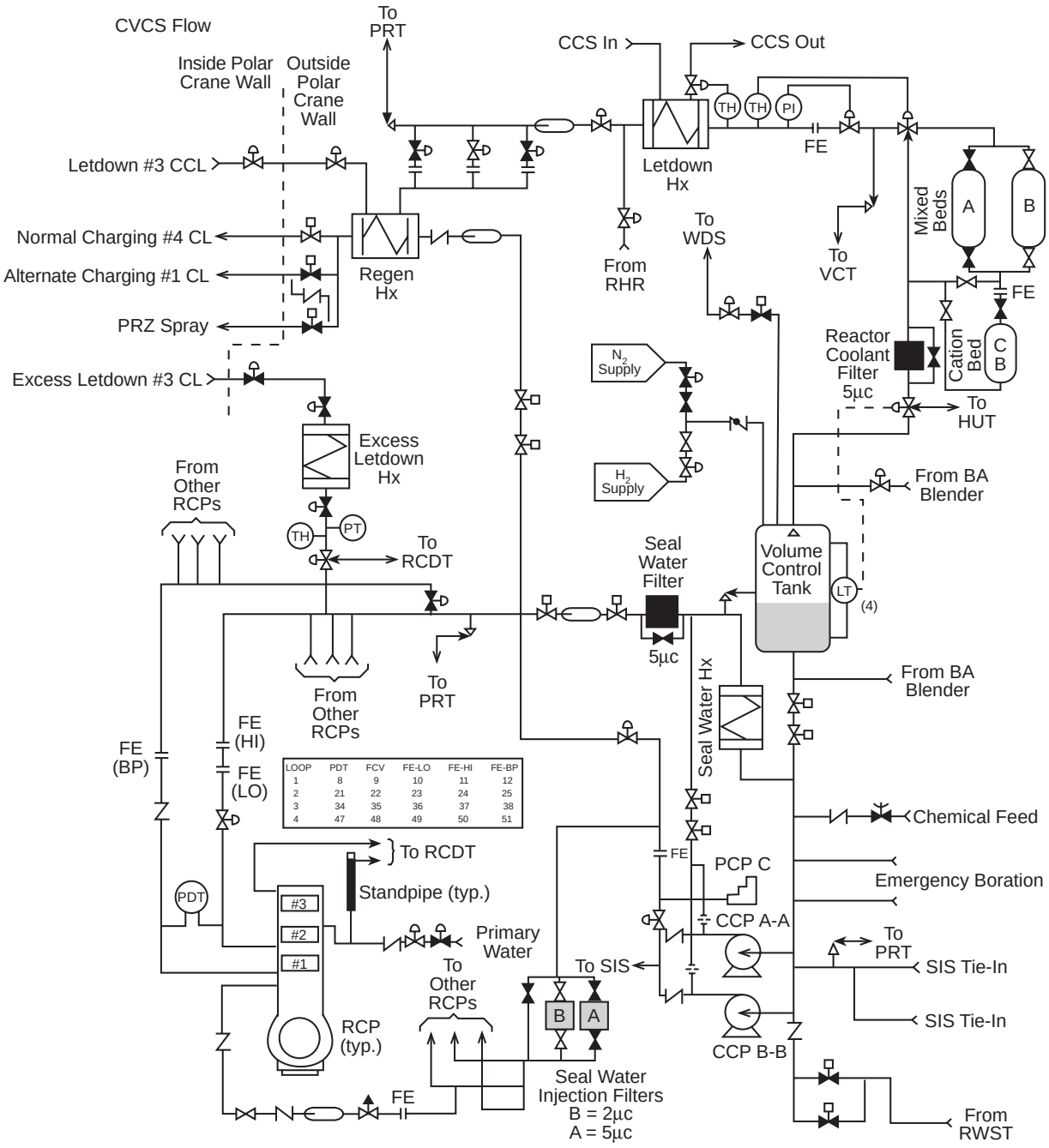


Figure B-3
Chemical Volume and Control High Head Injection Flow Path



LOOP	PDT	FCV	FE-LO	FE-HI	FE-BP
1	8	9	10	11	12
2	21	22	23	24	25
3	34	35	36	37	38
4	47	48	49	50	51

Figure B-4
Simplified Flow Diagram for Makeup Control System

Solid Water Plant Operations

The flow path begins as residual heat removal but immediately after the heat exchangers, the flow moves through the flow control valve to the CVCS. From there, the flow returns to the RCS via the normal CVCS path. All of the letdown orifice valves are open to provide a pressure relief path if RCS pressure should rise rapidly. RCS pressure is 50 to 75 psig with an average temperature of 140°F. At this point, hydrogen peroxide can be added to initiate "crud bursts" of cobalt-58 and cobalt-60. To remove these isotopes, the cation bed is placed in service. RCS pressure control is via flow control valves. The flow control valves are opened and throttled as necessary.

RCP Seal Water System and Excess Letdown System

The RCP seal water system and excess letdown system start downstream of flow element (FE) (3-inch line) and move through one of two 5-micron disposable filters in the seal water injection filters. There is a bypass around the filters. The main line then branches into four separate supply lines (one per reactor coolant pump). The supply lines each have a flow element and a flow balancing line outside the containment. The line then penetrates the containment, and flow moves through two check valves and a 2 to 1-1/2 inch reducer prior to entering the reactor coolant pump.

Inside the RCP

Flow enters the reactor coolant pump below the lower radial bearing but above the reactor coolant pump thermal barrier. Of the 8 gpm flow, 5 gpm flow across the thermal barrier into the RCS and 3 gpm flow across the lower radial bearing to the seals. After the lower radial bearing, there is a No. 1 seal bypass line (normally closed) that is opened when there is less than a 1 gpm flow (in No. 1 seal return line) and the RCS temperature brings in the "high temperature on lower radial bearing" alarm. Each bypass line has a flow-limiting orifice (1 gpm), check valve, and flow switch. The four bypass lines go through a flow control valve prior to tying into the seal water return line. The seal flow normally goes through the No. 1 seal, which has a large pressure drop across it (>2200 psig) to keep the seal faces apart.

Outside the RCP

After passing the No. 1 seal, the majority of the flow enters the No. 1 seal leakoff lines (2.9 gpm). The rest of the seal flow goes through the No. 2 seal and No. 3 seal, which also have appropriate leakoff lines. The No. 2 leakoff line goes to a standpipe, through an orifice, and then to the reactor coolant drain tank (RCDT). The standpipe maintains a backpressure on the downstream side of the No. 2 seal to ensure that the flow reaches the No. 3 seal. The No. 2 seal leakoff is approximately 3 gallons/hour. The No. 3 leadoff goes directly to the RCDT (100 cc/hour). Each No. 1 leakoff line has an isolation valve and two series flow transmitters prior to all four lines tying into one seal water return

line. This allows the No. 2 seal to be placed in service if the No. 1 seal fails. Also tying into the seal water return line is the No. 1 seal bypass return line and the discharge line from the excess letdown heat exchanger.

The Excess Letdown Flow Path

The excess letdown flow path starts at the RCS cold leg loop No. 3 (1-inch line). It passes through two RCS pressure boundary isolation valves (flow control valves, both are air-opened isolation valves). Then it goes through the excess letdown heat exchanger:

- One per unit @ 20 gpm (tube side) RCS flow
- Cooled by CCS from 545 to 195°F
- Heat transfer rate, 4.61×10^6 BTU / hr

The excess letdown flow control valve is normally aligned to the seal water return line (failed position also) and can be diverted to the RCDT.

Seal Water Return Line

The seal water return line has a pressure relief valve that protects the line if a containment isolation occurs. The valve lifts at 200 psig and relieves to the pressurizer relief tank.

Boron Recovery System

The function of this system is to conserve boric acid and makeup water by recycling effluent from the reactor coolant system. The boron recovery system starts at the CVCS holdup tanks and then:

- Passes through one of three gas stripper feed pumps (30 gpm each), one per tank and one spare
- Passes through two series-connected evaporator feed ion exchangers to remove lithium and fission products
- Passes through an ion exchanger filter (25 micron) to collect resin fines
- Passes to the evaporator package if sufficiently cleaned of Li and fission products or to the holdup tanks to be recirculated if not cleaned of Li and fission products

Reactor Makeup Control System

The reactor makeup control system functions are to maintain the desired operating inventory in the volume control tank and to adjust the RCS boron concentration. This system consists of a group of instruments and valves arranged to provide a manually preselected makeup of demineralized water and boric acid to the charging pump suction or the volume control tank.

The reactor makeup control system (RMCS) has five modes of operation in which to provide RCS water inventory control (via the volume control tank) and RCS boron concentration control (via the volume control tank). Boron concentration changes are made during the following plant conditions:

- Reactor startup - cold shutdown and refueling to full power operations
- Load follow - compensation for xenon transients following load changes
- Fuel burnup - decreased to compensate for fuel burnup and increased fission product buildup
- Cold shutdown - increased to bring RCS to cold shutdown conditions (2,000 ppm)

The RMCS consists of the following instruments and equipment:

- Four level transmitters on the volume control tank (used for automatic control)
- Two makeup stop valves (charging pump suction and top of volume control tank)
- Boric acid blender (pipe tee mixer)
- Primary water makeup valve and hand controller
- Boric acid makeup flow transmitter
- Boric acid transfer pump speed switch (one per pump, two total)
- Makeup control switch (start / stop)
- Makeup mode selector switch (six positions)
- Two adjustable batch integrators (one for boric acid and the other for primary water)
- Two flow rate meters (one for boric acid and the other for primary water makeup)
- Two flow controllers with manual and automatic positions (one for the boric acid valve and the other for the primary water makeup valve)
- Two pen flow recorders

The RCMS is normally in the automatic mode of operation to compensate for minor RCS leakage. This mode provides a boric acid solution preset to match the RCS boron concentration, which is fuel-burnup dependent. The other RMCS modes are used when system demands exceed the normal demands, for example, during xenon transients, cooldowns, heatups, and so on.

The RMCS modes of operation (indicated by the makeup mode selector switch) are as follows:

Automatic Mode. The boric acid solution is preset by the operator at the beginning of the shift (if necessary). When in automatic mode, the RCMS opens to a preset position to deliver 70 gpm (FC to 35%). The boric acid valve goes to a valve position as preset by the operator (on the valve hand controller).

Manual Mode. The operator must manually line up the discharge flow path and set the amount of boric acid and primary water on the batch integrator counters. The rate of flow is determined by presetting the flow hand controllers for valves. Procedure:

1. Set the flow rates on the hand controllers.
2. Set the desired quantities of boric acid and primary water on the respective batch counters.
3. Place the mode selector switch on manual.
4. Turn the boric acid to blender flow control switch to start.
5. If borating or blending, open the blender outlet to the charging pump suction valve flow control valve. If diluting, open the valve. If alternate diluting, open both.

At this point, the boric acid transfer pumps go to high speed. The flow control valves open to their respective valve positions and the batch counter starts counting. When the preset quantities have been added, the boric acid transfer pumps go to slow speed and the flow control valves close. The operator must close the flow control valves, reset the batch counters, and place the makeup control switch to the stop position. The operator can choose to flush the discharge path with 10 or 20 gallons of primary water. If either batch integrator finishes before the other integrator, the valve and pump (BA only) associated with the satisfied integrator terminates the flow. The other integrator continues until it is finished.

Borate Mode. Used when a long-term activity reduction is desired.

Procedure:

1. Set the BA flow controller (FC) to the desired flow rate.
2. Set the BA batch integrator (FQ) to the desired quantity.
3. Place the mode selector switch to the borate position.
4. Turn the makeup control switch to start (opens).

At this point, the BA transfer pumps go to high speed. When the batch has been discharged to RCS, flush the lines using the alternate dilute procedure.

NOTE: At this point, the boric acid transfer pumps go to high speed. Realign the modes to automatic mode.

Dilute Mode. Used for long-term reactivity addition. If FLRCS is in automatic mode, control rods should drive in to compensate for dilution.

Procedure:

1. Start a PMW pump, if one is not running already.

2. Set the PW flow controller to the desired flow rate (HS) and FQ (batch integrator) for the desired quantity.
3. Place the mode selector switch in the dilute position.
4. The flow control valve opens.
5. Watch the volume control tank level to ensure that it does not overfill. (The flow control valve should open around 63%.)
6. Re-align the controls to automatic mode when the desired quantity is discharged to the RCS.

Alternate Dilute Mode. Used when rapid dilutions are needed, for example, xenon transients, load changes, and so on. This is the same procedure as dilute mode except that you place the mode selector switch in the alt. dilute position. Ensure that both flow control valves are open. Return the controls to automatic mode.

> ***CAUTION:*** *Alternate dilute reduces H_2 concentration because less dilution water flows through the volume control tank vapor space.*

Emergency Boration. Emergency boration is to be used for an unexplained or uncontrolled reactivity increase due to:

- Increasing count rate while the reactor is shut down
- Abnormal control rod insertion or withdrawal
- Stuck out control rod(s) following a reactor trip
- Uncontrolled RCS cooldown following a reactor trip
- Excessive control rod insertion while at power
- Failure of normal boration equipment when RCS boron concentration increase is required
- Shutdown reactor in the event of an ATWS

Emergency boration is not automatic and requires operator action. It can be accomplished by one of the following methods:

- Normal flowpath through flow control valve
- Alternate flowpath through manual valve
- From reactor water storage tank through LCVs
- Emergency boration through BIT

RCP Seal Water System Operation

The seal water system is always in service even if all reactor coolant pumps are not running. This is so that if the need arises, the reactor coolant pumps can be started without having to wait for flow path re-alignment or lineup. The reactor coolant pumps generally are shut off when the average temperature is less than 160°F. Normal seal injection flow is between 6–11 gpm. Normal seal return flow is 3 gpm / reactor coolant pump. The No. 1 seal return bypass valve is opened when: RCS pressure is more than 100 psig, is less than 1000 psig, and the No. 1 seal leakoff valve is open, and the No. 1 seal leakoff flow rate is more than 1 gpm. Seal injection water flow is greater than 6 gpm / reactor coolant pump. Volume control tank pressure is maintained at greater than or equal to 17 psig to ensure that the No. 2 seal flow is adequate.

ECCS Mode of Operation

Upon receipt of an SI signal, the CVCS normal charging and letdown is isolated from the central charging pumps. CVCS isolation involves the closure of flow control valves. Flow realignment is such that both central charging pumps discharge through the RCS while taking suction from the RWST. LCVs open flow control valves (outlet valves) and inlet valves open. Seal water flow is still being maintained since the flow control valve does not close.

Component Description

Heat Exchangers

Regenerative Heat Exchanger

One per unit. One pass, three shell heat exchanger.

Shell side: 2,485 psig, 650°F, austenitic stainless steel.

Tube side: 2,735 psig, 650°F, stainless steel.

Flow rates: tube side-27, 170 lbm / hr; shell side-37, 050 lbm / hr.

ΔTs: shell side 545 to 290°F; tube side 130 to 495°F.

Heat transfer rate: 10.3×10^6 BTU / hr.

Letdown Heat Exchanger

One per unit.

Shell side: 150 psig, 250°F, carbon steel.

Tube side: 600 psig, 400°F; stainless steel.

Flow rates: shell side CCS flow, 203,000 lbm / hr (normal), 492,000 lbm / hr (design); tube side RCS flow, 37,050 lbm / hr (normal), lbm / hr (design).

ΔTs: shell side 95°F to 125°F; tube side 290 to 127°F (normal), 380 to 127°F (design).

Heat transfer rate (design): 14.8×10^6 BTU / hr.

Excess Letdown Heat Exchanger

One per unit.

Shell side: 150 psig, 250°F, carbon steel.

Tube side: 2,485 psig, 650°F, stainless steel.

Flow rates: shell side CCS flow, 115,000 lbm / hr; tube side RCS flow, 12,380 lbm / hr.

ΔTs: shell side 95 to 135°F; tube side 545 to 195°F.

Heat transfer rate: 4.61×10^6 BTU / hr.

Seal Water Heat Exchanger

One per unit.

Shell side: 150 psig, 250°F, carbon steel.

Tube side: 150 psig, 250°F, austenitic stainless steel.

Flow rates: shell side CCS flow, 105,000 lbm / hr; tube side seal water return flow, 160,500 lbm / hr.

ΔTs: shell side 95 to 120°F; tube side 144 to 127°F.

Heat transfer rate: 2.49×10^6 BTU / hr.

Pumps

Centrifugal Charging Pumps

Two per unit. 100% capacity each, 2,800 psig, 300°F, 600 HP, 6900 V shutdown bds., 150 gpm, design (w / 60 gpm recirculation), 5,800 ft. head, design 550 gpm at 600 psig (w / 0.0 gpm recirc.). Stainless steel, multistage diffuser design, self-contained lubrication system, mechanical seal cooling system, cooled by CCS. Single speed centrifugal pumps (induction motors).

Reciprocating Charging Pump

One per unit. Positive displacement pumps, variable speed, 3,200 psig, 300°F, 200 HP, 480 V shutdown boards, 98 gpm, design, 5,800 ft. head, design. Maximum operating pressure, 3,125 psig. Stainless steel construction, minimum charging flow is 32 gpm for reactor coolant pump seal water supply through a mechanical setting on the hydraulic speed changer.

Boric Acid Transfer Pumps

Two per unit. Two-speed pumps: 75 gpm (low), 150 gpm (high). 75 gpm design, 235 ft. head, design, 150 psig, 250°F. Supply boric acid makeup and fill BATs from boric acid batching tank.

Primary Water Makeup Pumps

Two per unit. 150 gpm, design. 15 ft. head, design.

Tanks

Volume Control Tank (VCT)

One per unit, 400 cubic ft volume (3,000 gallons), 75 psig, design, stainless steel, A nitrogen and hydrogen supply line provided to either reduce H_2 concentration in the volume control tank (cooldowns) or to provide oxygen scavenging. The hydrogen blanket is normally maintained at 17 psig to scavenge O_2.

Holdup Tanks

Two (shared), stainless steel, 127,000 gallon capacity each. Basis for size is one cold shutdown and restart at 80% of core life followed by a hot shutdown and restart on both units. Approximately four system volumes of one unit, 15 psig, 200°F, vacuum relief valve (1/8 psiv). One recirculation pump for both tanks (500 gpm @ 100 ft. head).

Boric Acid Tanks

Three tanks, 1 per unit and 1 shared, capacity 11,000 gallons each. Sufficient boric acid (20,000 ppm) for refueling, plus on cold shutdown at BOL with most reactive control rod withdrawn. Stainless steel. Atmospheric design pressure at 250°F. Contains two 100% capacity heaters to prevent boric acid from coming out of solution (165°F normal).

Central Charging Pump Injection Tank

One per unit, 900 gallon capacity. Provides reactivity control to ensure that the reactor does not return to an appreciable power level following a main steamline break.

Primary Water Storage Tanks

One per unit. Capacity 187,000 gallons each. Aluminum with neoprene coating and a diaphragm to keep O_2 within limits. 32 ft. high and 32.5 ft. in diameter.

Monitor Tank

Can be shared if a multi-unit site. Capacity 18,000 gallons based on BA evaporators operating continuously for periods in excess of 12 hours.

Chemical Mixing Tank

Used for maintaining RCS chemistry. Hydrazine for O_2 control when RCS 250°F and heating up RCS. LiOH for pH control as required. Hydrogen peroxide added to oxygenate RCS to rapidly promote and accelerate the release of cobalt-58 and cobalt-60 so that it can be removed prior to cavity flooding. RCS temp 140°F, one reactor coolant pump running and going to open up RCS.

Demineralizers

Mixed Bed

Two per unit, one in service, on standby. 39 cubic ft. 120 gpm, design (normal 20–120 gpm), 200 psig, 250°F, stainless steel construction. Removes corrosion products (filter action), fission products (ion exchange), and can be used for lithium removal. Contains both anion (removes positively charged ions) and cation (removes negatively charged ions) resins.

Evaporator Feed Ion Exchangers

Two per unit. Removes lithium and fission products; hydrogen from cation resin - 27 cubic ft. Capacity; one core cycle w / load follow operations and 1% fuel defectors. Connected in series to ensure no breakthrough of cesium (with 1% fuel defects), 75 gpm design, and 72 gpm normal, 200 psig, 250°F design.

Evaporator Condensate Demineralizers

One per unit, anion resin, 20 cubic ft, 72 gpm (normal). Used as a polishing demineralizer. Capacity based on 10 ppm boron carryover. Regenerative. 200 psig, 250°F design.

Filters

All filters are of stainless steel construction. Design pressures, temperatures, and flow rates are dictated by the system's requirements at the point in the system in which the filters are located. Generally, all filters have a bypass line and manual isolation valves to ensure flow while filters are being replaced. The exception is the seal water injection filters, which are 1-micron disposable filters.

Letdown Orifices

Provides rated flow only when RCS pressure is at 2,235 psig and downstream pressure is 335 psig (ΔP = 1,900 psig). As RCS pressure drops, so does the letdown flow according to the relationship: diameter of 75 gpm orifices is .246 inches. Diameter of 45 gpm orifice is .19 inches

Electrical Power Supplies

- Centrifugal charging pumps - 6900 V shutdown boards
- Reciprocating charging pumps - 480 V shutdown boards
- Central charging pump auxiliary oil pumps - 480 V reactor motor-operated valve (MOV) boards
- Boric acid transfer pumps - 480 V reactor MOV boards. Heat Tracing-480 V reactor MOV boards
- Air-operated valves, control circuits, and solenoids are powered from 125 V dc battery boards
- MOVs - 480 V reactor MOV boards

Appendix C

RELIABILITY-CENTERED MAINTENANCE

A study conducted at the Sequoyah Nuclear Plant entitled "Reliability-Centered Maintenance Qualitative Analysis" was presented at the 1992 ANS/ASME Nuclear Energy Conference. Figure C-1 shows the maintenance hours saved due to reliability-centered maintenance (RCM) analysis at Sequoyah.

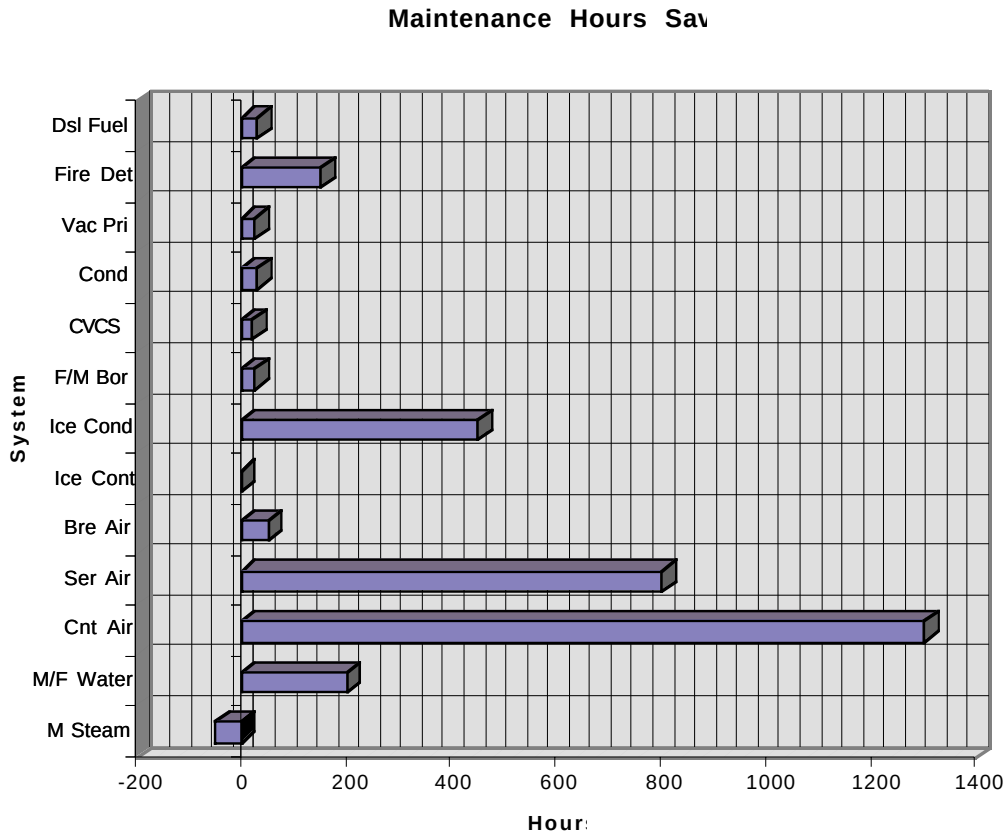


Figure C-1
Maintenance Hours Saved by RCM Analysis

This study concluded that RCM implementation on 13 systems resulted in a total maintenance reduction of over 3000 hours per year at a cost of about $20,000 per system. The study also showed that applying RCM techniques to the CVCS resulted in little savings, principally because the RCM analysis showed that the CVCS was already being optimally maintained at Sequoyah. Even so, Sequoyah's CVCS had experienced a high number of failures, second only to the Main Steam System at that plant. The number of CVCS pump failures reported by Sequoyah Plants 1 and 2 (between 1990 and 1994) is on a par with all plants reporting failure data to the Nuclear Plant Reliability Data System (NPRDS) during that period for the same model pump.

The fact that maintenance cost savings might not be realized for any one system or component is not a reason to discount the benefits of RCM analysis, because unless and until the analysis is done, the savings cannot be calculated. Sources of data available for the RCM analyst include plant-specific equipment history, Nuclear Plant Reliability Data System (NPRDS), probabilistic risk assessment (PRA), and others. *Nuclear Plant Reliability: Data Collection and Usage Guide*, TR-100381, developed by EPRI, is a useful reference for determining the applicability and availability of maintenance-related data.

There are many important issues that are fundamental to how the RCM process is conducted. A paper by L. W. Owsenek and W. D. Midgett, of PRC Engineering Systems, Inc., entitled "Important Decisions in RCM Program Planning: Inception to Implementation," highlighted these issues and was presented at the 1991 International Atomic Energy Agency meeting in Vienna, Austria. All of the issues raised can have significant effects on the ultimate success of the RCM program and need to be addressed by RCM program planners. The major issues presented by Owsenek and Midgett are paraphrased in the following sections, not to discourage, but to forewarn planners and managers and to encourage debate and informed decision making before the process starts.

Issue # 1. Justifying the Program

The basic question of cost effectiveness is always raised, hopefully in the planning stages of the program. Lessons learned include:

- Clearly identifiable cost savings cannot be obtained from other utility RCM programs.
- Near-term cost savings will not be realized from PM task reductions unless accompanied by maintenance staff reductions.
- Long-term savings from avoidance of equipment failure are probabilistic.
- Near-term savings will be achieved in plants that currently have excessive or nonapplicable PM tasks.
- Long-term savings will be achieved in plants with weak or inadequate PM programs.
- The many by-products of RCM analysis (design improvements, inventory upgrades, procedure improvements, person-rem reduction, and so on) are tangible.

Issue # 2. Goals and Objectives

Broad goals and objectives that are necessary for policy reasons have little value in guiding performance. The process of setting RCM program goals and objectives should consider:

- Elimination of ineffective or nonapplicable tasks.
- Identification of tasks to reduce contracted labor.
- Identification of tasks to reduce critical path time during outages, start-up, or testing.
- Substitution of predictive maintenance tasks for time-directed tasks can outpace the plant's technical capability.

Issue # 3. System Selection

System selection must be linked to the goals and objectives. Lessons learned include:

- Selecting systems for RCM analysis based on "importance" leads to interminable debate unless a universally accepted definition of "important" is used by everyone involved.
- The importance of a system might not be relevant if not linked to goals and objectives of the program.

Issue # 4. Critical Component Definition

Critical components must be identified to focus attention on only those components whose failure has the most significant effect on safety, power generation, or cost of corrective maintenance. Lessons learned include:

- A consistent selection process must be followed.
- Inconsistent selection can result in costly RCM evaluation of all equipment.
- Selection of critical components should include determining the cost of component failure.

Issue # 5. Generic Failure Modes, Causes, and Mechanisms

While the consequences of component failure might differ based on function and operating environment, the number and types of failure modes (and underlying causes and mechanisms) will generally be the same. Lessons learned include:

- Without consistency in the terminology of failure modes, causes, and mechanisms, the data used in the RCM analysis is tainted.
- Consistency can be achieved by using the EPRI RCM Workstation software.
- Data entry in the industry-wide equipment reporting databases is personalized and inconsistent, and must be interpreted and validated prior to use in analysis.

Issue # 6. Vendor Manual Recommendations

Vendor equipment manuals do not recommend maintenance tasks that optimize equipment reliability, achieve reliable operation at the lowest cost, consider the nuclear plant operating environment, or address the precise function of the equipment. When determining the relevancy of vendor recommendations, consider the following:

- Is the preventive maintenance according to vendor recommendations a specific technical specification or license requirement?
- Will vendor warranties be affected by deviating from vendor-recommended maintenance?
- Are vendor manuals up-to-date?
- How experienced are the RCM Project Team members?
- What methods should be used to optimize the frequency of time-directed PM tasks?

The RCM process is a systematic approach for selecting and documenting maintenance tasks on critical components. An authoritative reference and valuable source of information on nuclear industry-specific RCM methods is the EPRI *Reliability-Centered Maintenance (RCM) Technical Handbook*, TR-100320, that accompanies the EPRI RCM Workstation software. Although the handbook does not provide instruction in RCM methods, it does provide technical information and guidance that can be applied effectively across a variety of RCM methods practiced in the electric power industry.

The “Westinghouse Approach to Reliability Centered Maintenance” was presented by J. F. Kolonay and H. T. Maguire, Jr., at the April 1991 American Nuclear Society international meeting on Nuclear Power Plant and Facility Maintenance. The Westinghouse approach follows the traditional RCM model. It is included in this manual in its entirety because it provides a philosophical, yet clear, discussion of the traditional RCM elements as they relate specifically to the nuclear power industry:

THE WESTINGHOUSE APPROACH TO RELIABILITY-CENTERED MAINTENANCE

J. F. Kolonay and H. T. Maguire, Jr.

Westinghouse Nuclear and Advanced Technology Division

P.O. Box 355, Pittsburgh PA 15146 USA

ABSTRACT

Reliability-centered maintenance (RCM) is a technique which is gaining wide application in the nuclear power industry. RCM programs have now been initiated at a number of utilities. The status of these programs ranges from pilot system analyses to identify critical components, to multi-system programs which have resulted in fundamental changes in maintenance plans and even equipment changes. The reported experiences vary in the level of perceived success and enthusiasm. In particular there is a consensus that one of the most difficult tasks is the development and follow through for recommended changes to the plant's existing maintenance program. This paper suggests an integrated approach to address these issues. Details of the difficulties encountered are described as well as suggestions for improving the process.

THE INTEGRATED MAINTENANCE PROGRAM

The overall maintenance program must ensure that the set of maintenance activities reflect and are integrated with all the varied needs and requirements for the selected components. The program should be broad enough to support such diverse but critical aspects as corrective maintenance, optimal spare parts stocking, and maintenance training programs. Further, to be effective, the program must provide that any recommended new maintenance program actions be implemented and documented all the way through to new procedures.

Recommended actions that are not reconciled with activities such as utility commitments to regulators or Plant Life Extension will have difficulty being implemented at the plant, or if implemented, will not be fully effective. One aspect not accounted for in some programs is the fundamental need to plan for corrective maintenance. In addition, there is a need for a program which is engineered and documented. It is essential that an integrated and comprehensive maintenance program be developed

The following lists key aspects of an integrated program:

1. Critical System Identification
2. Critical Component Identification
3. Component Configuration Definition
4. Location Specific Requirements
5. Component Specific Requirements
6. Component Specific Activity Development
7. Activity To Task Breakdown
8. Subtask Development
9. Procedure Layout And Production
10. Implementation Assessment
11. Feedback Cycle Implementation

A comprehensive and integrated approach is required to address adequately the varied needs and interfaces which drive a plant maintenance program. The functional performance designed into the equipment must be understood and combined with all the requirements external to the equipment in order to provide an effective and justifiable maintenance program. A systematic process of development and documentation of the engineering basis for the overall maintenance plan can satisfy all the necessary requirements while providing a very efficient means to build and maintain the necessary documentation.

This program is depicted in Figure C-2 and is outlined below.

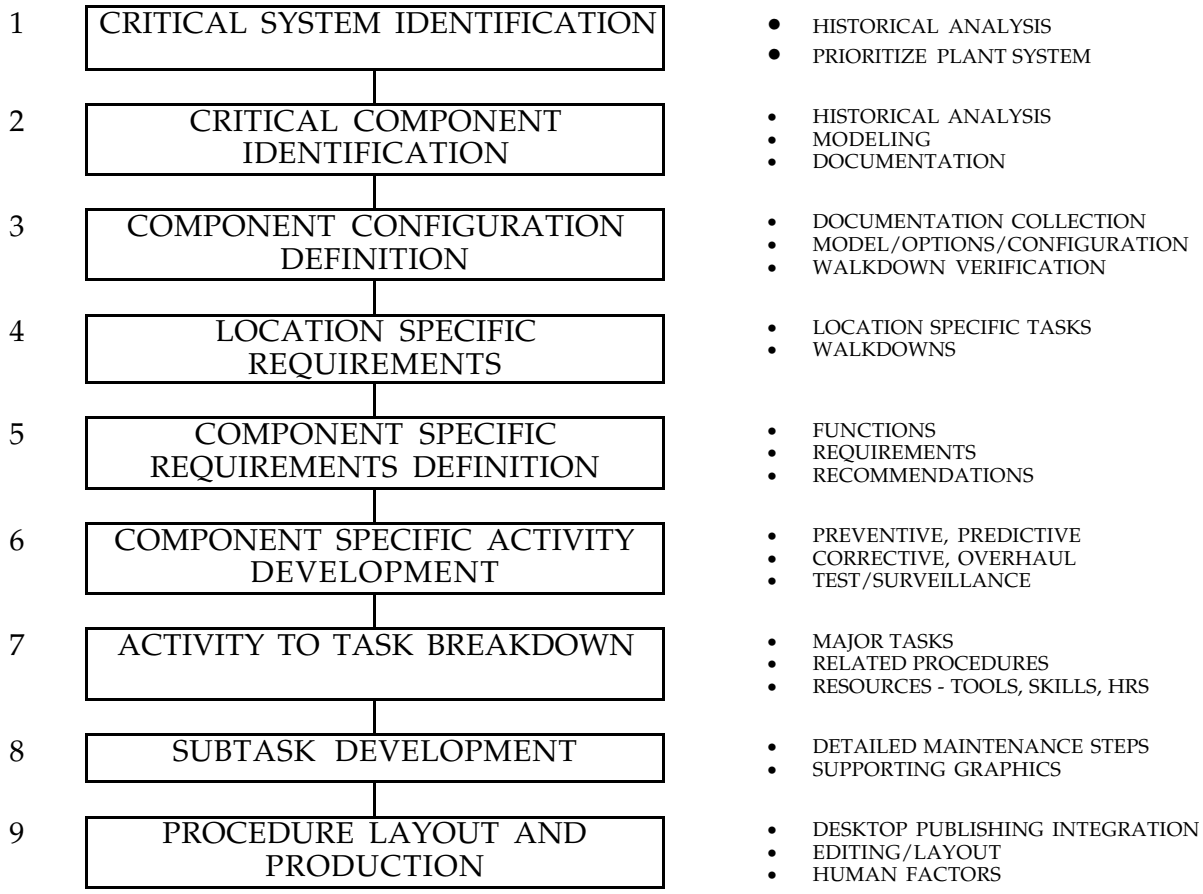


Figure C-2
Integrated Maintenance Program

1. IDENTIFICATION OF CRITICAL SYSTEMS

The selection of systems provides an important focusing mechanism, allowing for progress on specific plant goals to be as rapid as possible. The basis for selection and prioritization of systems is varied but is usually some combination of the following criteria:

– Plant Safety

– Plant Availability / Operations

– Maintenance Cost Reduction

– Current Issues at the Site

Using the above criteria, the selection / prioritization of systems can usually be quickly and easily accomplished.

2. IDENTIFICATION OF CRITICAL COMPONENTS

The RCM philosophy requires that resources be concentrated on those components which are critical to plant operations and safety. The prioritization process is based upon the impact of the equipment on plant availability, plant safety, economic, and other plant specific factors.

The result of this process should be a prioritized list of the plant's critical components and the documentation necessary to support the logic for selection. The following stages in the process will then only address the identified critical components. The process for identification and prioritization of those components is straightforward, and if the criteria are clearly defined, can be a fairly automated process that is easily delegated and should not require extensive resources.

3. COMPONENT CONFIGURATION DEFINITION

The determination of the exact component configuration is a crucial step but one which might be taken for granted. The manufacturer / model number for equipment is commonly recorded in plant databases, but there are cases where the simple use of this information is insufficient. First, the component may have been changed out since the database was created without a subsequent change in the database. In addition, complicated components are made up of subcomponents of many different manufacturers and these need to be identified and verified. For example, a pump might have an optional roller bearing and be coupled to the motor with a coupling from another manufacturer. It is important to know exactly what bearing is in the pump now, not what came with it originally. Also, is the oil system the same? Has the impeller been upgraded? Has the lubrication been revised in favor of another type? Is the seal a stuffing box or has it been replaced with a mechanical type?

The depth of knowledge described above is difficult to capture in a plant equipment database and often is known only to the maintenance personnel who work on the equipment, but is vital for the proper understanding of the component and the correct specification of a maintenance plan.

4. LOCATION SPECIFIC REQUIREMENTS

In the specification of maintenance programs, it is useful to separate information which is specific to the application of a component from that which is generic to that equipment type. Thus, maintenance preparation for a given manufacturer / model number may be different for each application. Here are some of the considerations:

– A component may have a more rigorous application (higher temperatures, pressures, radiation, longer cycle times, etc.) than other uses of the same equipment and, therefore, require more maintenance.

– A component may be in a difficult location, requiring special consideration. Such things as temporary scaffolding, access to pneumatic or electrical power, cranes, etc., may have to be supplied.
– A component may be in a safety application requiring special QA holdpoints.
– A component may be in a hazardous environment requiring special permits, e.g., radiation, hot work, confined area, etc.

5. COMPONENT SPECIFIC REQUIREMENTS DEFINITION

This is one of the most critical stages in the program because it identifies all the various recommendations and requirements that are related to the component and provides the link necessary to formulate the basis for the overall maintenance program. For each critical component, this includes identification and documentation of:

– Related industry cases and standards
– Utility commitments to regulators
– Related documents issued by regulators (e.g. NUREGs or SOERs)
– Plant Technical Specifications related to the component
– Vendor documentation such as manuals, technical bulletins, and supporting vendor analyses
– Vendor recommendations for inspection and maintenance
– Vendor warranty requirements
– Plant Life Extension requirements

These items are catalogued and filed along with a database of cross-references to all identical components for which they apply. This database should also enable the most efficient identification of the impact of changes or updates to particular references.

6. COMPONENT SPECIFIC ACTIVITY DEVELOPMENT

Next, an evaluation should be performed based upon the requirements and equipment performance needs identified above, combined with the specific equipment's historical performance in the field, as well as with detailed knowledge of the equipment's capabilities. The new activity list should reflect any new techniques such as diagnostic & monitoring which can support more effective and lower cost performance. The result of this step is detailed identification of all the preventive, corrective, predictive, and test maintenance activities necessary to support the component's performance as well as the frequency of performance.

The resulting list of activities is reconciled with any and all existing activities wherever they may be performed within the plant's organization. This then forms the overall maintenance plan for the component and provides the detailed documentation necessary to justify the program as well as to support ongoing changes to the program. The plan identifies where separate procedures are needed or where generic procedures apply.

7. ACTIVITY TO TASK BREAKDOWN

The activities identified above for the maintenance plan should then be broken down into the major tasks necessary to carry out the activities. Each activity is made up of major tasks and each task is one or more steps or conditions. These tasks encompass all the actions normally required to execute them within the plant's policies. In addition to the specific maintenance actions, this includes such things as pre and post maintenance system alignments, health physics requirements, as well as such administrative tasks as notification of impacted plant personnel.

The result of this step is a list of all the tasks necessary to carry out the specific maintenance activities. Also included should be a cross-reference to identical or similar tasks to identify where existing information may be used to describe the task. This provides an integration point for all related plant procedures.

8. SUBTASK DEVELOPMENT

The tasks identified above now need to be described in enough detail that the implementing personnel can execute the activities. In addition, the language used in the descriptions should be sufficiently detailed and formatted to support the plants Procedure Writers Guide. This is because these descriptions will be catalogued and electronically imported for use in the creation of procedures. Some of the tasks are addressed by programs that are separate from the specific maintenance activity and can be addressed by a reference to the plant procedures or policies in which the task is described. For the maintenance activities, the major steps of each task need to be described in detail and include detailed step-by-step instructions and include lists of spare parts, tools, and other resources that are needed. A human factors approach is taken if the task definitions are executed effectively. The Major / Minor step procedure writing technique is recommended to make the procedures usable for both experienced and less experienced maintenance procedures.

The result of this step is detailed step-by-step task descriptions that will form the basis for development of plant maintenance procedures and can also provide support for maintenance personnel training programs. A computerized database of these task descriptions will facilitate the use of individual tasks for multiple procedures.

9. PROCEDURE LAYOUT AND PRODUCTION

The task definitions must now be incorporated into specific procedures. These procedures are targeted to support all aspects of the maintenance process; corrective maintenance; preventive maintenance; and testing.

This stage in the overall process should emphasize the human factors approach to presentation of the tasks. This includes incorporation of high quality integrated text and graphics, presentation of cautions and hold points, and references to all related procedures and documentation. The computerization within a system that provides state-of-the-art desktop publishing capabilities will enable efficient and high quality output as well as enabling the cross-references that link all related documentation.

The result of this stage is a set of procedures that will provide maintenance personnel with the documentation necessary to effectively carry out their activities. The documentation can also be used as a maintenance training tool.

PROGRAM INTERFACES

Once a maintenance program has been established, an assessment of the ability of the organization to implement that program is necessary. This is a systematic assessment of all of the elements necessary to implement the plan. Some of the questions that need to be answered include:

- Has the PM scheduler been updated for new PMs and advised frequencies?
- Has a decision been made on the basis for the first performance of a new PM?
- Are all necessary spare parts for the maintenance program (both preventive and corrective) available? Is their stocking level appropriate? Do all existing parts meet the proper quality level and have the required documentation?
- Do all new procedures have the proper approval?
- Is maintenance training and staffing sufficient to support the revised maintenance program?

The above list is, of course, not exhaustive, but if the answer to many of the above answers is “NO”; then a substantial effort may be required before a successful implementation of the program can be completed.

ONGOING FEEDBACK CYCLE

One of the benefits of implementing a structured maintenance program is its ability to learn from past experience and improve over time. Unfortunately, this is not automatic or even particularly easy. Some of the important factors for being successful are:

– Create and implement procedures for collecting maintenance historical data in such a way that it can be utilized for assessing the performance of the maintenance plan.
– Ensure that the computer support for the maintenance data collection is adequate for the job.
– Assign the responsibility for a constant reassessment of the maintenance plan and monitor its progress.

The maintenance program should slowly evolve after its initial implementation to constantly improve its effectiveness. Maintenance activities should be added, deleted, and modified as necessary to maintain or improve performance. New predictive techniques should replace old ones and new products (improved lubricants, packing, bearing designs, materials) will gradually require changes in the plan.

CONCLUSION

A comprehensive and integrated approach is required to address adequately the varied needs and interfaces which drive a plant maintenance program. The functional performance designed into the equipment must be understood and combined with all the requirements external to the equipment systematic process of development and documentation of the engineering basis for the overall maintenance plan can satisfy all the necessary requirements while providing a very efficient means to build and maintain the necessary documentation. The coordination and cooperation of many groups within the plant are necessary to gain full benefit from the effort.

Appendix D

SHAFT ALIGNMENT

Misalignment can be defined as "Any condition in which the driver and driven shaft centerlines are not coplanar vertically and horizontally at the machine's normal operating temperature." It is one of the most common sources of excessive, disabling vibration in rotating machinery.

Machinery alignment is concerned with the practice of positioning machines so that the rotating elements rotate as one continuous unit. This eliminates all undue mechanical stresses on the shafts, couplings, bearings, and so on. To accomplish this, it is imperative that the centerlines of the two shafts form one continuous line; therefore, the concern is with shaft alignment. Although the term coupling alignment is often used, the true intent is to make sure that the shafts are aligned, not the couplings. If all couplings were bored straight and true through their exact center and were machined perfectly about their rim and face, it might be possible to align the two coupling halves and obtain correct machinery alignment. But irregularities in the machining process and even the forces imposed on the coupling halves during installation (especially with couplings that must be heated during installation) leave a large margin for error. This is not to say that the dial indicators should not be placed on the coupling halves to obtain measurements, but rather that the two shafts should be rotated simultaneously to obtain the desired readings.

Before proceeding with a description of the procedures for the various shaft alignment strategies, some basic concepts that apply to the alignment process must be understood.

Indicator sag is the term used to describe the bending of the dial indicator mounting hardware as the dial indicator is rotated from the top position to the bottom position. This bending always takes place during the rotation of the dial indicator. The degree to which the mounting hardware bends, however, is dependent upon the length and strength of the hardware. As the bending occurs, significant errors can be present in the indicator readings that are used to determine vertical misalignment, especially in the rim readings. To ensure that all readings are correct, it is always best to determine the amount of indicator sag present in the equipment before starting the alignment process. Indicator sag is visually displayed in Figure D-1.

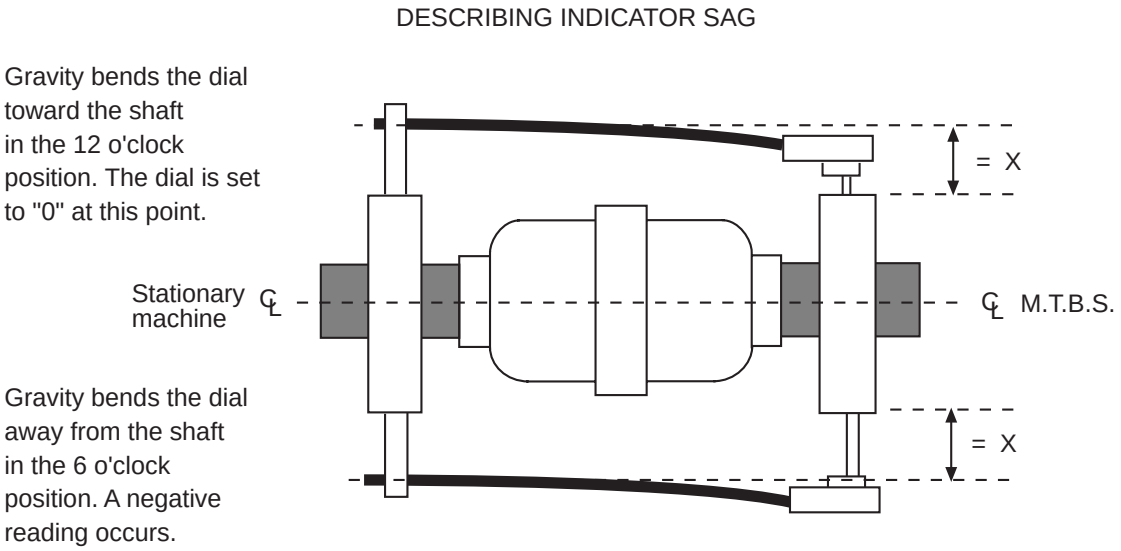


Figure D-1
Indicator Sag

Soft foot is the term commonly applied to the condition that exists when all four of the machine feet are not supporting the weight of the machine. When this condition exists, the shaft position changes every time the machinery foundation bolts are loosened and retightened. This condition makes performing an alignment extremely difficult.

Soft foot is a relatively new discovery. Machines that have had excessive vibration levels since installation have had these levels significantly reduced by correcting soft foot. The technical community has developed a general approach to checking soft foot with the machine running, reducing vibration levels even more. The procedure for performing this running soft foot check is included later in the soft foot test procedure. Soft foot is shown in Figure D-2.

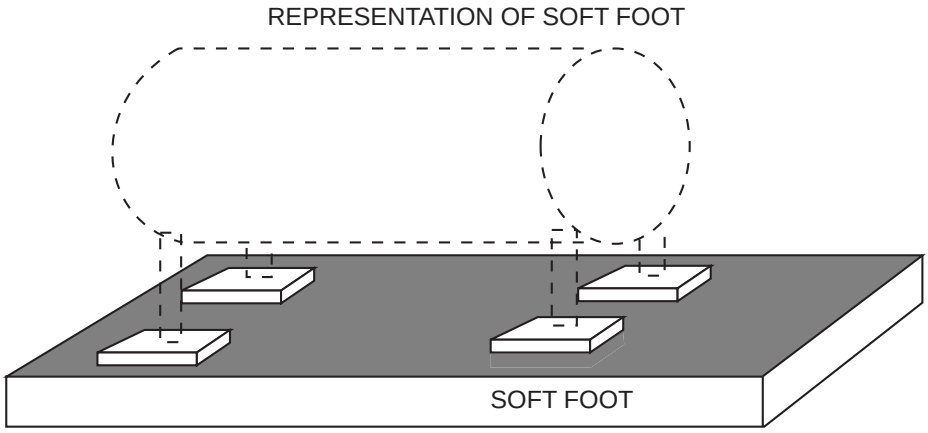


Figure D-2
Representation of Soft Foot

Because metal expands when it is heated, the change in the metal temperature that often occurs when a machine goes from a non-operating to an operating condition can greatly influence the alignment. As the machine heats up, the expansion of the metal causes the vertical position of the bearings to rise.

Many manufacturers include information relating to thermal rise in their installation manuals in the section dealing with alignment. If this is not included in your manual, the only way to determine the exact amount of compensation necessary to correct for thermal rise is a hot alignment check.

The accuracy of a hot alignment check is dependent on how soon after shutdown dial indicator readings can be taken. When thermal rise is known, the necessary compensation can be made during the initial alignment procedure. If the cold alignment setting is not known for your pump, it can be determined using the procedure in Section D.2 below.

All physical descriptions of alignment conditions must be stated in terms of angularity and offset. Angularity is the slope that is formed by the intersection of two shaft centerlines, measured in thousandths of an inch per inch. Positive and negative values are used to describe the direction of the slope.

Offset is the distance between the centerline of two shafts as measured at the midpoint of the coupling in either the horizontal or vertical plane, measured in thousandths of an inch.

The aligned condition of the shafts almost always consists of a combination of angularity and offset. Angularity and offset are shown in Figures D-3 and D-4, respectively.

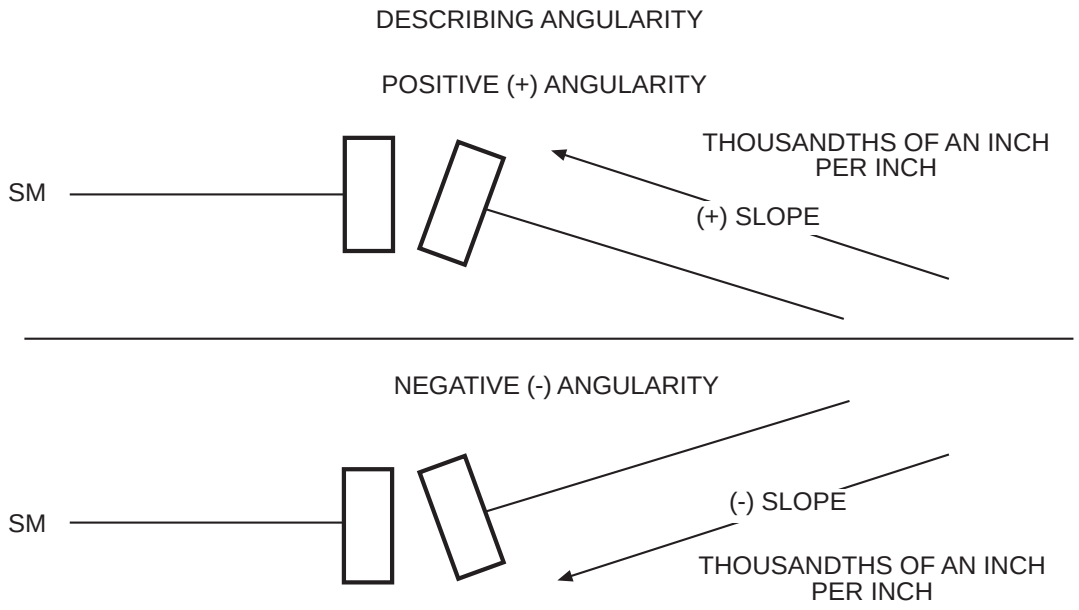


Figure D-3
Angularity

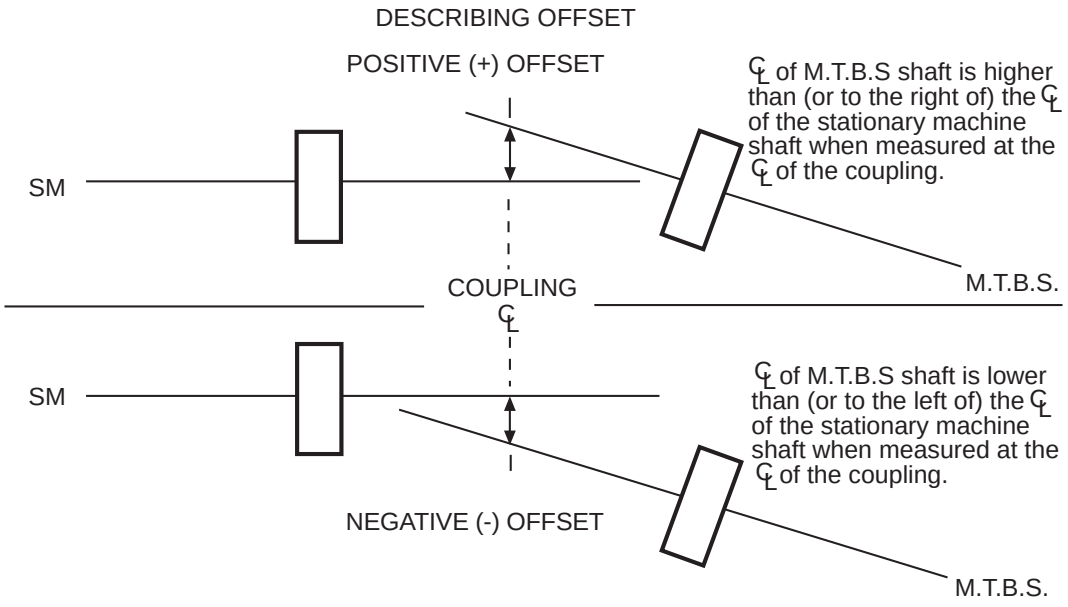


Figure D-4
Offset

There are essentially two techniques in current use for obtaining or verifying proper shaft alignment. These techniques are the dial indicator method and the laser alignment method. A description of these methods is given below.

There are three methods of aligning machinery with dial indicators:

- Two indicator method, dial indicator readings taken at the stationary machine. This method requires taking one rim reading and one face reading at the stationary machine (the dial indicator mounting brackets and posts are attached to the machine to be shimmed).
- Two indicator method, dial indicator readings taken at the machine to be shimmed. This method is the same as the previous method except the rim and face readings are taken at the machine to be shimmed (the dial indicator mounting brackets and posts are attached to the stationary machine).
- Indicator reverse method. This method requires the taking of two rim readings, one set at the machine to be shimmed and one set at the stationary machine.

The most common alignment mistake made is attempting to align, or checking the alignment of, two pieces of machinery by taking only rim readings on only one of the machines. The above three alignment methods make it obvious that this procedure supplies only half of the information needed. It is possible to get a set of rim readings that appear to be perfect on a machine that is considerably misaligned.

If a dial indicator alignment method is to be used, the factors that determine the method to use are end play and distance versus radius.

End Play - Nearly all machines with journal or sleeve bearings have some end play, but it can be manageable. Sufficient pressure can be applied to the end of the shaft during rotation to keep it firmly seated against the thrust bearing or plate. When end play is unmanageable, the indicator reverse method should be used because axial movement of the shaft causes large errors in face readings. However, the effect on rim readings, which are the only readings needed with the indicator reverse method, is negligible.

Distance versus Radius - When a choice exists, the best dial indicator method is determined by the following rule: If the distance between the points of contact of the two dial indicators that are set up to take rim readings for the indicator reverse method is larger than one-half the diameter of travel of the dial indicator that is set up to take face readings for the two indicator methods, the indicator reverse method should be used. The basis for this rule is that misalignment is more apparent (the dial indicator reading is larger) under these circumstances and, therefore, corrections are more accurate. See Figure D-5 for a visual representation of this determination.

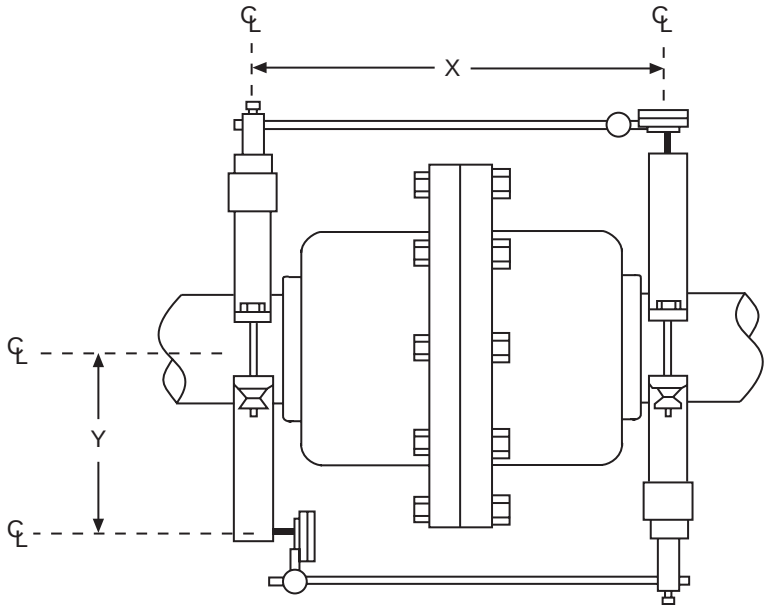


Figure D-5
Distance versus Radius

The most common dial indicator method used is the reverse indicator method. The procedure for this method is described below.

D.1 Alignment Procedure for Reverse Dial Indicator Method

WARNING: *To prevent injury to personnel, all shims shall be constructed in such a manner that a portion of each shim protrudes from beneath the support foot after it is fully inserted so that it can be grasped without placing one's fingers beneath the support foot. All burrs and sharp corners should be removed.*

1. Assemble the coupling using the manufacturer's instructions.
2. Check for and correct soft foot.

    2.1 Ensure that all hold-down bolts on both the pump and the driver are tightened and that all dowel pins are removed.

NOTE: When installing the dial indicator, it should be as close as possible to the hold-down bolt while allowing enough room to get a wrench on the bolt to loosen and tighten the bolt, as shown in Figure D-6.

2.2 Install a dial indicator on one foot of the machine.

2.3 Zero the dial indicator.

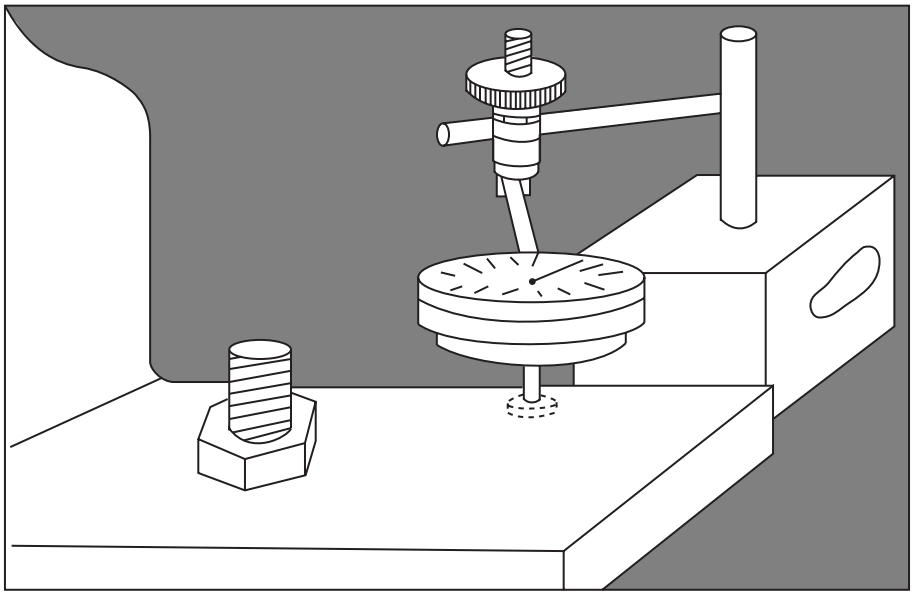


Figure D-6
Installing the Dial Indicator

NOTE: If the dial indicator shows more than 0.003" movement, soft foot exists on that support foot. Correct it using steps 2.5 through 2.7 before proceeding to measure any other feet.

2.4 Loosen the hold-down bolt, and note the amount of indicator movement.

2.5 If the measurement in step 2.4 exceeded 0.003", remove, measure, and clean the existing shims.

2.6 Replace the shim under the foot with a shim that has a thickness equal to the shim that was removed plus the amount of indicator movement measured in step 2.4.

2.7 Retighten the hold-down bolt, and repeat the soft foot check.

NOTE: If the soft foot condition has not improved after two correction attempts, a bent foot condition exists. Correct a bent foot using steps 2.8 through 2.11.

2.8 Loosen the hold-down bolt, and remove all shims that were added to correct for soft foot.

2.9 With the hold-down bolt loose, measure the gap under each corner of the support foot with a set of feeler gauges.

2.10 Manufacture a tapered shim to the dimensions of the gaps measured at the four corners of the support foot.

2.11 Install the tapered shim, and repeat the soft foot check.

2.12 Repeat steps 2.1 through 2.11 for the other feet.

3. Check the piping alignment.

3.1 Disconnect the mating flanges, and check for the following four possible problems:

NOTE: It might first be necessary to adjust the concentricity of the flanges by applying some pressure by hand to the piping in order to insert the first bolt. This bolt should be tightened enough to hold the flanges in position while you attempt to slide all other bolts through the flange bolt holes.

a. Concentricity. The outer diameter of the mating flanges should be concentric within 1/8".

b. Gap. The maximum separation between the flanges should be 1/4".

c. Parallelism. Measure the gap at four quadrants. The maximum deviation should be no more than 1/16".

d. Bolt hole orientation. Ensure that the bolt holes are similarly oriented by sliding a flange bolt through each bolt hole. No binding should occur.

3.2 Mount the dial indicator on the machinery base.

3.3 Position the indicator spindles to read the pump inlet and outlet flanges in both the horizontal and vertical planes.

NOTE: If the indicators move more than 0.004", excessive piping strain exists and should be corrected before proceeding.

3.4 Tighten the piping, and observe the indicator movement.

4. Measure the indicator sag.

NOTE: Before dial indicators on machinery shafts are mounted, indicator sag must be measured. Indicator sag is the bending of the dial indicator mounting hardware that occurs when the dial indicator is rotated from the 12 o'clock position to the 6 o'clock position when taking alignment readings on horizontally mounted machinery. Indicator sag must be compensated for only in the vertical plane before alignment readings are taken. Figure D-1 illustrates that the centerline of the indicator support rod remains equidistant from the centerline of the shaft, but the rod deflection causes the indicator spindle to extend (giving a negative error) after the shafts have been rotated 180°. Steps 4.1 through 4.5 measure indicator sag.

4.1 Attach the indicator bracket assembly to a stiff length of pipe or rod. Use the same bracket configuration and spacing that will be used when taking indicator readings on the pump.

4.2 Adjust the indicator so that the indicator spindle is at 50% travel and perpendicular to the reading surface.

4.3 Hold the test piece so that the indicator is at the 12 o'clock position, and zero the dial indicator.

4.4 Rotate the test pipe until the indicator is in the 6 o'clock position, and record the amount of indicator movement. The indicator always reads a negative value, and this is the amount of indicator sag.

4.5 Repeat steps 4.1 through 4.4 to verify the repeatability of the readings.

5. Mount the dial indicators on the shafts.

5.1 Mount the dial indicators on the shaft using the same configuration used when determining indicator sag.

5.2 Ensure that the indicator spindle is set to 50% travel and is perpendicular to the shaft centerline.

5.3 Compensate for indicator sag as follows:

5.3.1 Rotate the shafts so that the indicator is in the 12 o'clock position.

5.3.2 Set the dial indicator to the positive value of measured indicator sag (for example, if indicator sag was measured to be -0.007, the indicator should be set to +0.007).

5.4 Rotate the shafts an additional 360° to ensure the repeatability of the indicator settings.

5.5 Rotate the shaft so that the indicator nearest the machine to be shimmed (MTBS) is in the 12 o'clock position.

6. Perform the shaft alignment.

NOTE: Dial indicator readings in the vertical plane are referred to as being at the 12 o'clock or 6 o'clock position. The 12 o'clock position corresponds to the top, and the 6 o'clock position corresponds to the bottom.

NOTE: The shafts should always be rotated an additional 360° degrees after taking a dial indicator reading to ensure the repeatability of the readings. Steps 6.1 through 6.10 are for vertical plane alignment.

6.1 Rotate the shaft 180° so that the indicator closest to the MTBS is in the 6 o'clock position, making sure to follow the travel of the dial on the indicator.

6.2 Read the indicator value, and record the value as MB (MTBS indicator bottom reading).

6.3 Set the indicator closest to the stationary machine (SM) in the 12 o'clock position to compensate for indicator sag.

6.4 Rotate the shafts 180° so that the indicator closest to the SM is in the 6 o'clock position, making sure to follow the travel of the dial on the indicator.

6.5 Read the indicator value, and record the value as SB (SM indicator bottom reading).

6.6 Using a tape measure, measure and record the following values to the nearest 1/8":

6.6.1 The distance between indicator stems. Record this distance as A.

6.6.2 The distance from the coupling midpoint to the indicator nearest the MTBS. Record this distance as B.

6.6.3 The distance from the indicator nearest the MTBS to the front foot of the MTBS. Record this distance as CV.

6.6.4 The distance from the front foot to the back foot of the MTBS. Record this distance as DV.

6.7 Using the following formulas, calculate the values of vertical angularity and vertical offset.

Vertical angularity: (AngV) = (MB + SB) / (2 x A)

Vertical offset: (OSV) = (AngV x B) - (MB / 2)

NOTE: If the angularity and/or offset are outside tolerances, the machinery must be re-aligned in the vertical plane.

6.8 Using the following formulas, calculate the values for front foot shimming and back foot shimming.

Front foot shimming: FV = (AngV x CV) + (MB / 2)

Back foot shimming: BV = (AngV x DV) + FV

NOTE: When an alignment is performed, moves for the vertical plane correction must be performed first. After correcting vertical plane misalignment, a separate set of readings must be taken for the horizontal plane before a horizontal alignment correction is attempted. When making corrections in either the vertical or horizontal plane, snug the jacking bolts in the axial position to preserve the hub spacing. If axial jacking bolts are not available to secure the machinery in the axial position, hub spacing must be rechecked after finalizing the alignment.

6.9 Perform the shim adjustments, which were calculated to satisfy the vertical plane alignment.

NOTE: When re-tightening the hold-down bolts, use the tightening sequence shown in Figure D -7.

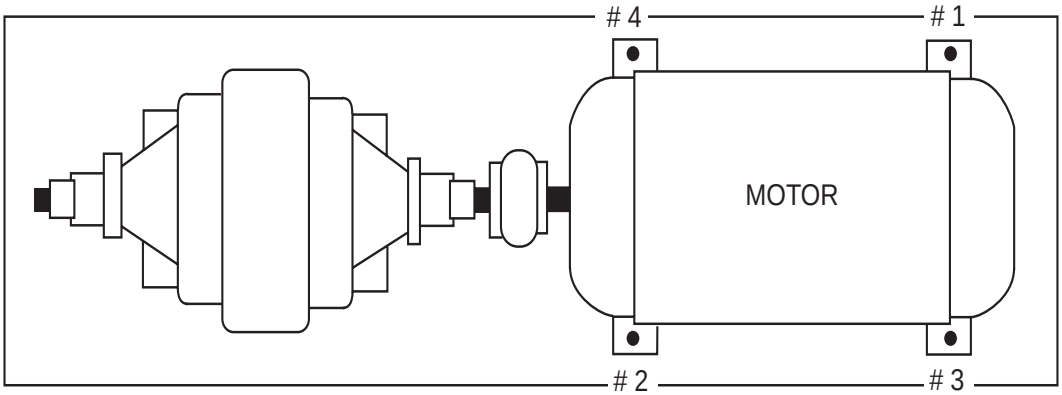


Figure D-7
Hold-Down Bolt Tightening Sequence

6.10 Tighten the bolts, and take a new set of vertical readings to verify the alignment.

NOTE: Dial indicator readings in the horizontal plane are referred to as being at the 3 o'clock or 9 o'clock position. All such references consider the clockwise direction as though the user were standing behind the outboard end of the MTBS and looking toward the coupling (the 3 o'clock position corresponds to the right side of the MTBS and the 9 o'clock position corresponds to the left side of the MTBS). Required adjustments in the horizontal plane are stated as being to the left or right. The above direction of view is used for referencing these moves also. Steps 6.11 through 6.17 perform the horizontal plane alignment.

6.11 Rotate the shafts so that the dial indicator closest to the MTBS is in the 3 o'clock position in the horizontal plane. Zero the indicator.

6.12 Rotate the shafts 180° so that the indicator closest to the MTBS is in the 9 o'clock position, making sure to follow the travel of the dial on the indicator.

6.13 Read the indicator value, and record the value as ML (MTBS indicator left reading).

6.14 Zero the indicator closest to the SM in the 3 o'clock position in the horizontal plane.

6.15 Rotate the shafts 180° so that the indicator closest to the SM is in the 9 o'clock position, making sure to follow the travel of the dial on the indicator.

6.16 Read the indicator value, and record the value as SL (SM indicator left reading).

6.17 Using the following formulas, calculate the horizontal angularity and horizontal offset. Use the same values for A and B as measured for vertical calculations.

Horizontal Angularity: (AngH) = (ML + SL) / (2 x A)

Horizontal Offset: (OSH) = (AngH x B) - (ML / 2)

NOTE: If the angularity or offset is outside the accepted tolerances, the machinery must be realigned in the horizontal plane. Steps 6.18 through 6.20 perform a horizontal realignment.

6.18 Using the following formulas, calculate the values for front foot movement and back foot movement. Use the same values for A and B as measured for vertical calculations. If indicators can be placed in line with the hold-down bolt centerlines, CH and DH measurements are also identical to CV and DV.

Front foot movement: (FH) = (AngH x CH) + (ML / 2)

Back foot movement: (BH) = (AngH x DH) + FH

6.19 Perform the required front and back foot movements. A positive value requires the feet to be moved to the right. A negative value requires the feet to be moved to the left.

6.20 Tighten the bolts and take a new set of horizontal readings (ML and SL).

6.21 Recalculate the horizontal angularity and horizontal offset.

6.22 After achieving acceptable horizontal alignment, take an additional set of vertical readings, and reverify the values for vertical angularity and offset. Record all values. Typical alignment specifications in Table D-1 should be used if no other tolerances are specified.

6.23 After satisfactory alignment has been achieved in both the vertical and horizontal planes, stake the feet using dowels.

6.24 Perform a final alignment check to ensure that doweling has not disturbed the alignment.

Table D-1
Typical Alignment Tolerances

RPM	Allowable Tolerances
0 - 1800	Less than 1.0 mil/inch
1801 - 4000	Less than 0.5 mil/inch
Above 4000	Less than 0.25 mil/inch

D.2 Determining the Cold Alignment Setting

Thermal growth occurs on nearly all rotating machinery. The only true measurement of thermal growth is obtained by taking a hot alignment check and a cold alignment check for each pump. The cold alignment setting can then be established for the pump by comparing the results of the two alignment checks.

Thermal growth is caused by metals expanding when machinery has a change in temperature from room temperature to operating temperature. The casing, bearings, and foundation (the shaft support system) change dimensions with a change in temperature. This change in shaft position is usually enough to push the shaft alignment far outside allowable tolerances.

The cold alignment setting (the desired angularity and offset in the cold state) must be established for each piece of machinery to attain an accurate alignment in the hot operating condition. A hot alignment check and a cold alignment check must be performed on the machinery, and the results obtained from these checks can be compared to establish an accurate cold alignment setting. These two checks should be taken within at least a few weeks of each other to avoid errors in the comparison that might result from such things as component wear. The following values recorded during hot and cold alignment checks are required:

Vertical angularity:	AngV
Vertical offset:	OSV
Vertical front foot shimming:	FFV
Vertical back foot shimming:	BFV
Horizontal angularity:	AngH
Horizontal offset:	OSH
Horizontal front foot adjustment:	FFH
Horizontal back foot adjustment:	BFH

Cold alignment settings can then be established by subtracting the results of the hot alignment check from the results of the cold alignment check using the following formulas:

NOTE: The subscripts s, c, and t are defined as follows: s = cold alignment setting, c = cold check, and t = hot check.

(Vertical angularity, cold alignment setting) $AngV_s = AngV_c - AngV_t$

(Vertical offset, cold alignment setting) $OSV_s = OSV_c - OSV_t$

(Horizontal angularity, cold alignment setting) $AngH_s = AngH_c - AngH_t$

(Horizontal offset, cold alignment setting) $OSH_s = OSH_c - OSH_t$

D.3 Common Problems When Using the Reverse Indicator Method

The following common errors are procedural errors that can result in the inability to achieve a satisfactory alignment:

- Misreading the dial indicators. This is a common occurrence when taking readings in either plane and usually is caused by the operator's failure to observe the movement of the dial pointer during the entire 180° rotation of the shaft. Careful and continuous observance of the pointer is required because it can make more than one complete revolution about the dial or start its rotation in one direction and reverse direction before the completion of the 180° rotation of the shaft. Extreme care is required when it is necessary to observe the pointer with a mirror. The shafts should always be rotated slowly, and it is recommended that the operator observing the dial indicator verbally announce the indicator reading several times during shaft rotation.
- Moving the machine in the opposite direction of that which is required. This is a common occurrence when adjustments are being made in the horizontal plane and is easily determined by the fact that the machine ends up further misaligned than it was before an adjustment was made (angularity, offset, and required horizontal moves will approximately double). The two major causes of this error are:
  - Dial indicator readings were taken on the wrong side of the coupling. Check that the indicators are being set to zero in the 3 o'clock position and read in the 9 o'clock position, looking from the outboard end of the MTBS toward the coupling.
  - Disorientation while making adjustments. Check that adjustments made at the support foot of the MTBS are to the left when the calculated answer is negative and to the right when it is positive, looking from the outboard end of the MTBS toward the coupling.
- Using inaccurate dial indicator readings. Inaccurate dial indicator readings are normally the result of improper installation of the indicator mounting hardware or positioning of the dial indicator. To ensure proper installation and positioning:
  - Check the bracketing and mounting hardware and dial indicator clamps for looseness.

  - Check to see if the dial indicator spindle exceeds its length of travel (by either bottoming out or losing contact with its measuring surface) at any time during the rotation of the shafts.
  - Check that the indicators and bracketing system do not hit any obstructions through the full 360° of shaft rotation.
  - Ensure that the dial indicator spindle is positioned perpendicular to the shaft in both the radial and axial planes.

- Using dirty or damaged dial indicators. Ensure that the dial indicator spindle moves freely throughout its entire length of travel.
- Using inaccurate dimensions. Ensure that the dimensions measured with a tape measure are correct. Remember that dimensions CH and DH are not the same as CV and DV unless the dial indicators used to monitor the horizontal adjustments can be placed at the front and back hold-down bolt centerlines (slight differences of up to 1/2" do not result in errors that cause more than two readjustments and can, therefore, be ignored).

If no procedural errors as described above are found, the machinery system must contain a problem. The most common problems and recommended methods for finding them are as follows:

- Soft foot. The existence of soft foot is the main reason that machines remain out of alignment after calculated adjustments have been made in either plane. Even if soft foot has been initially checked and corrected, it is advisable to repeat the procedures.
- Piping strain. Check to ensure that piping strain is not forcing the machinery to spring back while making adjustments in the horizontal plane or while tightening the piping flange bolts. Monitoring indicators should always be used when tightening flange bolts, even if piping strain has been checked.
- Binding of hold-down bolts. Check to ensure that the MTBS is not bolt-bound and being forced to spring back while making alignment moves in the horizontal plane or while tightening the hold-down bolts.
- Defective shims. Check that the machinery shims are free of all rust, dirt, and burrs. Additionally, do not rely upon sizes stamped on shims. Always verify the shim thickness with a micrometer.
- Improper tightening of hold-down bolts. Ensure that the hold-down bolts have been tightened sufficiently and in the proper sequence.
- Damaged bearings. Check the bearings by performing a thrust and bump check. Anti-friction bearings should be checked for damage by performing an indicator reading balance check. This check, described below, should be repeated at least three times. Unbalanced readings are indicative of anti-friction bearing damage.
  1. Set the dial indicator on the MTBS side of the coupling to zero in the 12 o'clock position.

2. Rotate the shafts until the indicator is in the 3 o'clock position, and record the reading.
3. Rotate the shafts until the indicator is in the 6 o'clock position, and record the reading.
4. Rotate the shafts until the indicator is in the 9 o'clock position, and record the reading.
5. Add the readings recorded at the 3 o'clock and 9 o'clock positions.
6. Subtract the sum of the two readings from the 6 o'clock position reading.
7. Potential bearing damage or bent-shaft condition exists if the answer is not within ± 0.002".
8. Repeat steps 1 through 8 using the indicator located on the SM side of the coupling.

- Bent shafts. Check for possible bending of the internal shafts by performing the indicator reading balance check, described above. Unbalanced readings are indicative of a shaft that is bent between the bearings (shafts that are bent outside the bearings can be properly aligned and do not create unbalanced readings).
- Loose coupling hubs. Check that the coupling hubs are securely fastened to the shaft.

D.4 Checking for Soft Foot with the Machine Running

Below is a generic procedure for checking a machine for soft foot with the machine running. Although soft foot corrections were probably already made during the initial alignment process, this is one of the better ways to pinpoint problems, by actually checking the effects of hold-down bolts on vibration levels.

NOTE: You must be extremely careful when performing this procedure because you are loosening hold-down bolts with the machine running. It is recommended that at least two persons be present when this procedure is performed.

1. Make a simple top view sketch of both the driving and driven machines, clearly numbering and marking the bolt locations.
2. Place the vibration instrument pickup against the point of highest vibration, usually at a bearing cap or the housing.
3. Read and record the vibration data.
4. Slowly loosen one bolt, watching for vibration changes. If vibration remains the same or increases, retighten the bolt. If, as the bolt is loosened, the vibration level decreases, loosen the bolt all the way and then retighten to finger tight. Usually bringing the bolt to a finger-tight condition does not increase the vibration, compared to when it was completely loose. However, sometimes an ever-so-slightly finger-tightened bolt increases the vibration considerably. If that happens, keep that bolt completely loose.

5. Continue the process with the other bolts, recording which bolts end up being tight and which are finger-tight or loose. Electric motors, generators, and other machines that have no pipe strain forces can run with almost all of their bolts loose. Try to keep at least two tight, sometimes only one. Experiment by loosening first the outboard (back) bolts and retightening, then loosening the inboard bolts. In other situations, the best results are obtained when working across the diagonal (for example, front left and rear right or vice versa).
6. Sometimes, the procedure is very simple and direct; at other times, it is more involved. If your adjustments are producing good results, you might improve them further by more experimentation. A bolt that gave the best results when tight in the beginning might give entirely different results after others have been adjusted. You might end up with loose bolts that earlier gave the best results when they were tight, and tight bolts that originally gave good results when loose.
7. Use thickness gauges and/or the previous dial indicator method for correcting soft foot under loose bolts. Do only one foot at a time. When you can tighten a bolt with no increase in vibration, you have corrected that soft foot. Go on to the next until all soft feet have been corrected.

D.5 Laser Alignment System

A standard laser alignment system consists of some optics, a laser, a sensor, a computer, and mounting fixtures.

The quality of measurement depends greatly on the quality of the optics. Most shaft alignment systems have only two or three optical components. These optics are coated to reduce glare, reduce incident light, filter light that is both higher and lower in frequency than the laser beam, and provide scratch resistance. The surfaces of a typical prism are ground to a tolerance of better than one arc-second (0.000278°). The optics are used to reflect or focus the light from a laser. Shaft alignment systems use very low-power (typically less than 1.0 milliwatt), solid-state-type lasers. The support electronics pulse the laser on and off several times a second.

The laser is used primarily as a straight reference line that does not sag or deflect. It attaches to one of the shafts so that the beam is parallel with the shaft centerline. When the shaft is rotated, the laser rotates with it. The beam will cut a cone in space when the shaft turns.

Shaft alignment systems must use a sensor to measure the position of the laser beam. The most common sensor uses the bi-axial position-sensitive detector. The detector emits a signal that is proportional to the position of the beam on the detector surface; it measures relative beam position.

Detectors can be from 6 mm diameter to over 10x10 mm square. They have the quality of giving a repeatable signal, but one that is nonlinear. There are several different approaches to turn a nonlinear detector into a linear sensor. The easiest approach is finding a linear spot on a detector and limiting the measurement range to that area. This is cheap but only yields a sensor range of about 0.030–0.050".

Another approach is to linearize only a single axis of the detector. This is also relatively cheap, but it requires more than one sensor (and laser) to measure misalignment.

A more global approach is inspecting each detector, mapping the surface non-linearity, and generating a set of equations that describe the linearity of the detector. These equations must then be burned into a chip on the sensor so that for any raw (nonlinear) reading, the corresponding correct (linear) reading is generated. This approach yields the highest accuracy and best long-term stability of any known method. However, it requires considerable know-how and manufacturing expertise from the vendor to produce.

It is the quality, range, and linearity of the sensors that differentiate laser shaft alignment systems. Everything else in the system depends on the sensor for data. If the sensor is inaccurate, the alignments will be slow and the results questionable. If the sensor is good, the alignments will be fast. In the typical system, the sensor is mounted to the machine shaft (the one to which the laser is not attached). It rotates with the shaft while the laser is rotated with its shaft. At various angles, usually 90° intervals, rotation is stopped and measurements are taken.

The sensor feeds its data to a computer for display. Sometimes the computer is a PC-compatible laptop, but this is the exception rather than the norm. Often the most difficult job the software has to do is read the sensor in real time.

Appendix E

PUMP DESIGN IMPROVEMENTS

Charging pump operational failures have resulted in recent design changes to improve the reliability and availability of charging pumps. Two areas that are being investigated and where design improvements are being tried are pump shafts and plunger / packing combinations. Shaft cracking has become a more visible problem recently, while packing wear has been a continuous maintenance and cost burden. Investigation results and design changes to improve these two areas are discussed in the following sections.

E.1 Pump Shaft Cracking

Shaft cracking has been a problem over the last several years, specifically in the 11-stage centrifugal barrel-type pump with an in-line rotor and pressure balancing drum. The pump shaft is fabricated from an A-276 type 414 stainless steel material.

The pump shaft cracking issue was first recognized when 9 out of 40 in-service pumps suffered from shaft failures before 1977. The pump shafts used in these pumps were fabricated from A-276 type 414 stainless steel with a 1000°F temper. The evaluations conducted on the failed shafts included surface examinations, metallographic and fractographic examinations, mechanical property tests, and chemical evaluations. The mechanical property tests consisted of hardness measurements, tensile tests, and Charpy toughness tests.

The crack locations corresponded to either the locknut thread roots or split ring grooves, where the stresses were likely to be higher due to the effect of stress concentration. The fracture planes were transverse to the shaft. The surface examinations of the failed shafts generally showed the presence of circumferential wear patterns in the balance drum seating area. These wear patterns appeared bright with no evidence of oxide deposits, suggesting that the wear might have been active until the shaft failed. The presence of wear damage suggested that the shafts had been subjected to vibratory loads.

The results of the fractographic examinations showed that the cracking initiated at one or more closely situated sites on the OD surface of the shaft at the thread roots. The cracks quickly joined to form a single crack and progressed across the diameter of the shaft. The fracture surface appeared smooth with numerous marks, suggesting changes in load, environment, or both during the life of the crack. Higher magnification showed

that the fractures were consistent with crack progression under cyclic loading with evidence of fatigue.

The results of the metallographic examinations did not reveal any evidence of abnormalities in the material microstructure. Chemical analysis showed the material met the requirements of type 414 stainless steel. Hardness measurements of the polished sections showed average Rockwell "C" hardness values in the range of 30–35, corresponding to an approximate tensile strength value of 136–157 ksi. The results of the Charpy impact testing showed that the material corresponded to a fracture appearance transition temperature (FATT) in the range of 350–400°F.

Based on the overall results of the evaluation, it was concluded that the cracking in the pump shafts was induced by a high cyclic fatigue mechanism. The circumferential wear marks suggested the presence of vibratory loads on the shaft under operating conditions. Although review of the shaft analysis under normal operating conditions demonstrated that the shaft stresses were within acceptable limits, the shaft failures demonstrated the need for increased design margin. The following steps were taken in an attempt to alleviate the occurrence of shaft failures:

1. Change in the Shaft Material Heat Treatment

The first action that was taken was to change the heat treatment in order to improve the mechanical properties of the shaft material. The steel processing specification was revised to austenitize the shaft material at 1814°F, oil quench the material, temper it at 1150°F, and air-cool it. Environmental fatigue tests indicated an improvement of 12% for material treated to the revised specification. Charpy impact tests showed a 300% improvement in impact strength. This improvement in the impact strength greatly improved the shaft fracture toughness and reduced notch sensitivity.

After incorporation of the changes to the shaft manufacturing process, additional shaft failures occurred. All failures occurred at the second-, fourth-, or eleventh-stage split ring grooves or the balance drum locknut thread. As a result, a detailed analysis was performed for both the locknut thread and the split ring groove. Failures could not be predicted in either location.

2. Redesign of the Balance Drum Locknut and Shaft Threads

Suspecting that the shafts were being subjected to loadings that were beyond the original design basis, the pump vendor initiated fully instrumented pump testing that focused on shaft loading. Results indicated that the shaft end tended to lift relative to the balance drum due to the hydrodynamic bearing film in the radial bearing. This shaft lift created an additional bending load in the locknut area. The balance drum locknut and shaft threads were redesigned. A new two-piece tapered nut design was shown to result in a more uniform stress distribution across the shaft threads. Additionally, the shaft threads were rolled instead of cut, which further reduced the corresponding stress concentration factor. Successful operating experience with these upgrades demonstrates

the adequacy of the new design. Although charging pump failures have occurred after implementation of these upgrades, in each case it was suspected that the pump encountered conditions that were beyond that of the design basis.

E.2 Plunger/Packing Failure

Packing wear and subsequent replacement has been a huge maintenance burden on positive displacement pumps from the beginning, with significant amounts of money being poured into frequent replacements. It is not unusual to replace the packing on a pump with less than 500 run hours, and it is considered a success if a pump manages to exceed 1000 hours without having the packing replaced.

The replacement of the pump packing takes about a full day for three to four people, not including support personnel from health physics and radiological waste. The cost can be anywhere from $5,000–15,000 per pump repack, depending on whether the plungers and other components need replacing. The majority of plants were averaging approximately one repack per month.

In the early 1990s, Jack Wade at Waterford 3 began experimenting with different plunger / packing combinations in an attempt to increase pump availability and run time between failures. As of December 1994, the A charging pump had run for over a year and 4,100 hours without a repack. The B pump had run for 9 months and 2,700 hours, and the A / B pump had run for 11 months and 2,000 hours.

From 1989 to 1994, many things changed that affected packing life. The new pump blocks had larger seal water ports, which allowed more cooling for the packing. The Operations Department began running seal water 30 minutes before starting the pump and continued to run the seal water for 30 minutes after the pump was secured. This resulted in better packing lubrication and more cooling. The run cycle was also extended from one day to seven days, reducing the number of times the pump was started and stopped. The pump plungers originally had a Colmonony 72 surface coating. They were replaced with plungers coated with a titanium nitride material. Although these plungers cost approximately $1,000 more, they extend the life of the plunger about four times what it had been.

The most significant change was the change in the packing itself. In 1989, Waterford 3 was using UTEX EDPM and cotton material packing. It was a chevron-type packing that lasted about 1,000–1,300 hours of pump operation. A new Nomax slug-type packing was recommended by the manufacturer of the new block. This packing lasted only about 900 hours. The packing currently being used is made by Garlock and is a poly cotton-SBR rubber-nitral reinforced material. Arkansas Nuclear One is currently using this same plunger / packing combination on one pump and experiencing success. The pump currently has about 2,000 hours of run time.

This combination might not be the cure-all for all plants, but it is very apparent that different plunger / packing combinations produce significantly different results. Packing cooling and the surface finish on the plunger itself also play a very important role.

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Electric Power Research Institute
3412 Hillview Avenue
Palo Alto, CA 94304
(P.O. Box 10412, Palo Alto, CA 94303)
800-313-EPRI
www.epri.com

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