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Large Butterfly Valve Maintenance Guide

Technical Report

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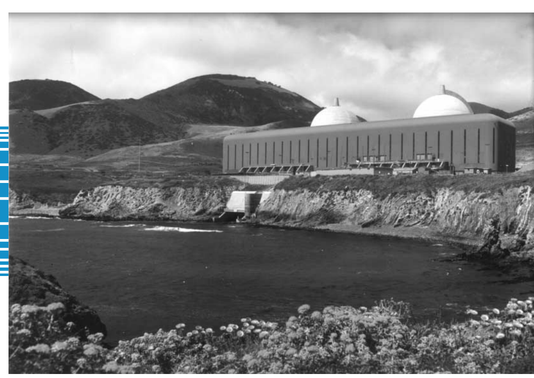

NMAC

Large Butterfly Valve Maintenance Guide

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Final Report, December 2003

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REPORT SUMMARY

Background
In 2002, EPRI NMAC conducted a survey of site representatives to determine which maintenance guides would be most useful to plant personnel. The survey identified large butterfly valve maintenance as the sixth most useful issue with respect to adding moderate to high value to the plant. To address this issue, an additional survey was completed for areas that had not been previously addressed by earlier EPRI documents. It was determined that while there were several guides addressing application, particularly for motor-operated butterfly valves, there were no guides that addressed butterfly valve maintenance in any detail.

Objectives
- To provide detailed information to personnel involved with large butterfly valve maintenance
- To provide insights to experienced personnel
- To provide information, guidance, and instructions to less experienced personnel assigned to the maintenance and operation of large butterfly valves

Approach
A detailed review of industry literature, product information, and standards was conducted to identify the various designs, applications, and maintenance practices associated with large butterfly valves. Utility and industry failure databases were surveyed to determine specific problems and commonly encountered failure mechanisms. Based on these reviews, a draft was prepared that identified condition monitoring, preventive maintenance, and troubleshooting methods. The draft was presented to a Task Advisory Group (TAG) consisting of NMAC-member utility engineers for review and feedback.

Results
This guide provides information to help power plant maintenance personnel understand the basic principles of large butterfly valve design and operation. It identifies problems with large butterfly valve operation and maintenance (O&M) and uses that information to provide troubleshooting strategies. Finally, it provides inspection and monitoring techniques for implementing a preventive maintenance program for butterfly valves, including those used for raw water system valves and containment isolation.

EPRI Perspective

Large butterfly valve O&M, particularly for raw water systems such as service and circulating water, continues to be a challenge to reliable plant operations. This guide provides the tools necessary to understand the special problems of large butterfly valve maintenance. It also provides information about effectively troubleshooting and correcting those problems as well as steps for establishing a condition monitoring and preventive maintenance program.

Keywords

Butterfly valve
Condition monitoring
Preventive maintenance
Raw water
Reliability
Troubleshooting

ABSTRACT

This guide provides information to help power plant maintenance personnel understand the basic principles of large butterfly valve design and operation. It identifies problems with large butterfly valve operation and maintenance (O&M) and uses that information to provide troubleshooting strategies. Finally, it provides inspection and monitoring techniques to implement a preventive maintenance program for butterfly valves, including those used for raw water system valves and containment isolation.

This guide was prepared based on a detailed review of industry literature, product information, and standards to identify the various designs, applications, and maintenance practices associated with large butterfly valves. Utility and industry failure databases were surveyed to determine specific problems and commonly encountered failure mechanisms. Plants were contacted directly to determine their experience and solutions. Condition monitoring, preventive maintenance, repair, and troubleshooting methods were identified based on these reviews and discussions. A draft was prepared and presented to a Task Advisory Group (TAG) consisting of NMAC-member utility engineers for review and feedback.

ACKNOWLEDGMENTS

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Curt A. Helmers	Duke Energy
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Russell L. Lake	Calvert Cliffs
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John F. Lund	TVA
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1 INTRODUCTION

1.1 Background

In 2002, EPRI NMAC conducted a survey of site representatives to determine which maintenance guides would be most useful to plant personnel. The survey revealed that preparing a guide that addressed large butterfly valve maintenance problems ranked sixth with respect to adding high to medium value to the plant. To address this issue, an additional survey was completed for areas that had not been previously addressed by earlier EPRI documents. It was determined that while there were several guides addressing application, particularly for motor-operated butterfly valves, there were no guides that addressed butterfly valve maintenance in any detail.

1.2 Guideline Approach

The approach included the following process:

- Review of EPRI and industry literature, product information, and standards to identify various designs, applications, and maintenance practices associated with large butterfly valves
- Survey of utility and industry failure databases to determine specific problems and commonly encountered failure mechanisms
- Identification of and recommendations for condition monitoring, preventive maintenance, and troubleshooting methods that met the task objectives

1.3 Highlighting of Key Points

Throughout this report, key information is summarized in Key Points. Key Points are bold lettered boxes that succinctly restate information covered in detail in the surrounding text. The primary intent of a Key Point is to emphasize information that will allow individuals to take action for the benefit of their plants.

The Key Points are organized according to the three categories:

- O&M Cost
- Technical
- Human Performance

Each category has an identifying icon, as shown below, to draw attention to the Key Point when quickly reviewing the guide.

Key O&M Cost Point

Emphasizes information that will result in reduced purchase, operating, or maintenance costs.

Key Technical Point

Targets information that will lead to improved equipment reliability.

Key Human Performance Point

Denotes information that requires personnel action or consideration in order to prevent injury or damage or to ease completion of the task.

Section 6 contains a listing of all key information in each category. The listing restates each Key Point and provides reference to its location in the body of the report. By reviewing this listing, users of this guide can determine if they have taken advantage of key information that the writers believe would be of substantial benefit.

1.4 Glossary

A glossary of terms used in this guideline is contained in Appendix A.

2
BASIC PRINCIPLES OF LARGE BUTTERFLY DESIGN AND OPERATION

2.1 Introduction

This section provides descriptive information about large butterfly valves to help maintenance personnel understand the basic principles of large butterfly valve design and operation. For the purpose of this guide, a large butterfly valve is considered to be 10 in. (DN 250) and larger.

This chapter begins with a general discussion of butterfly valves in typical power plant isolation valve applications. Specific part discussions follow in subsequent sections. Special considerations related to butterfly valves in modulating/throttling service are provided in this section. EPRI's *Application Guide for Motor-Operated Valves in Nuclear Power Plants* [1] provides detailed discussions for the design, installation, operation, and torque requirements for motor operated butterfly valves in nuclear power plants. That EPRI report should be consulted for additional details not covered here.

Figure 2-1 shows an overall assembly of a butterfly valve with a manual gear actuator with the principal parts identified. The body is usually carbon steel or stainless steel. The disc, disc pin, and shaft are stainless steel or some other high alloy material. The bearings are normally an elastomer such as polytetrafluoroethylene (PTFE). For nuclear power plants, the seat is normally rubber or some other elastomer. Some designs (for example, triple-offset) are commonly metal-to-metal seated.

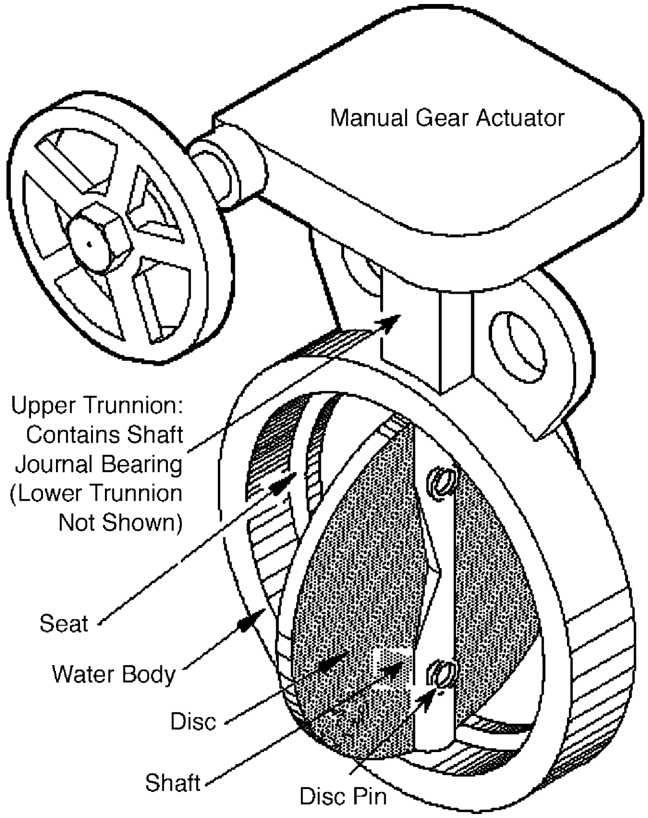


Figure 2-1
Typical Butterfly Valve with Manual Gear Actuator

Butterfly valves are most commonly used in nuclear power plant low-pressure and low-temperature water systems such as service water and component cooling water. They are also found in ventilation systems such as containment purge/makeup. Although they are most often used in the pressure service of the American National Standards Institute (ANSI) Class 300 (PN 50) or less, higher pressure designs are available up to ANSI Class 1500 (PN 250) in smaller sizes using metal-to-metal seats. Rubber seats are generally limited to Class 600 (PN 100).

Butterfly valves offer several advantages over other types of valves, especially where soft seats are acceptable. These advantages include:

- Reduced installation cost, weight, and space requirements, particularly in large sizes.
- Reduced operating energy cost due to high flow capacity (C_v) and low pressure drop in the full open position.
- Reduced maintenance costs even when handling dirty fluids and fluids with suspended solids (for example, in service water applications).
- Improved sealing capability with seat tightness up to Class VI [2] particularly with high-performance valve designs (see Section 2.2.1).

- Versatility in material selection, which extends butterfly valve applications to higher operating pressures (typically up to Class 600/PN 100) and temperatures (typically up to 400°F/200°C), and lower leakage (typically up to Class VI requirements).
- Generally self-closing hydrodynamic torque characteristics that make certain butterfly valves good choices for fail-close operation.
- Flow characteristics that make butterfly valves well-suited for throttling service, but with limitations as discussed later in this report. With appropriate disc designs, butterfly valves can be used for a control function.

Butterfly valves are one of a group of valves labeled quarterturn. The ball valve and plug valve are also quarterturn valves. The term quarterturn comes from the fact that the disc is rotated approximately one-quarter of a full rotation from the fully closed to fully open positions. The disc rotation of some valves is limited to 60° to 70°. Some angle seated butterfly valve designs close at angles other than 0°.

The disc is attached to a shaft that extends outside the body and is connected to an actuator. These valves can be compact, lightweight (resulting in reduced piping supports), and relatively inexpensive. They are available in sizes as large as 120 inches (DN 3000).

2.2 Design Codes

The overall population of butterfly valves in a U.S. nuclear power plant is divided into two broad categories: ANSI design and AWWAdesign.

2.2.1 ANSI Design Butterfly Valves

ANSI design butterfly valves are built to the requirements of ANSI B16.5 [3], ANSI B16.34 [4], or ANSI/ASME Section III [5]. They are identified as Class 150, 300, and 600 (PN 20, 50, and 100). Many of these valves are also designed to MSS SP-67 *Butterfly Valves* [6] and MSS SP-68 *High Pressure-Offset Seat Butterfly Valve*s [7].

2.2.2 AWWA Design Butterfly Valves

American Water Works Association (AWWA) design butterfly valves meet the requirements of *AWWA Standard for Rubber-Seated Butterfly Valves* [8]. They are identified as Class 25A, 25B, 75A, 75B, 150A, 150B, and 250B. They are specifically designed for fresh water with a temperature of 33–125°F (0.6–52°C) and a maximum steady-state pressure of 250 psig (1723 MPa). The class designation (for example, 250) is the maximum working pressure. The suffix letter A or B defines the flow rate capability for the valve in the fully open position. "A" designates a maximum velocity of 8 ft/s (2.4 m/s) and "B" designates a maximum velocity of 16 ft/s (4.9 m/sec).

2.3 Liner

In the symmetric design shown in Figure 2-2(a), valve seat leakage is prevented by using a sealing system between disc and body referred to as the liner. The liner also prevents the fluid from coming into contact with the body. This permits the use of carbon steel in most low-temperature fluid systems and helps keep the price of the valve low. The liner most commonly uses an elastomer such as ethylene propylene rubber (EPR). The liner material sets the temperature (limit ≈ 350°F/177°C) and operating fluid type (see Appendix B).

In both the ANSI and AWWA designs, the rubber liner is typically molded over the flange faces acting as a flange gasket, although some designs use an adhesive to bind an independent rubber liner inserted into the body flow port. It is important to realize that for these designs, normal gaskets are not to be used because leakage is highly likely at the flange joint. A liner is not normally used for high temperature service. Therefore, from a fluid compatibility perspective, the body material becomes an important design consideration (for example, using a 300 series stainless steel for both temperature and fluid.) Sealing also becomes metal-to-metal, usually along a tapered surface.

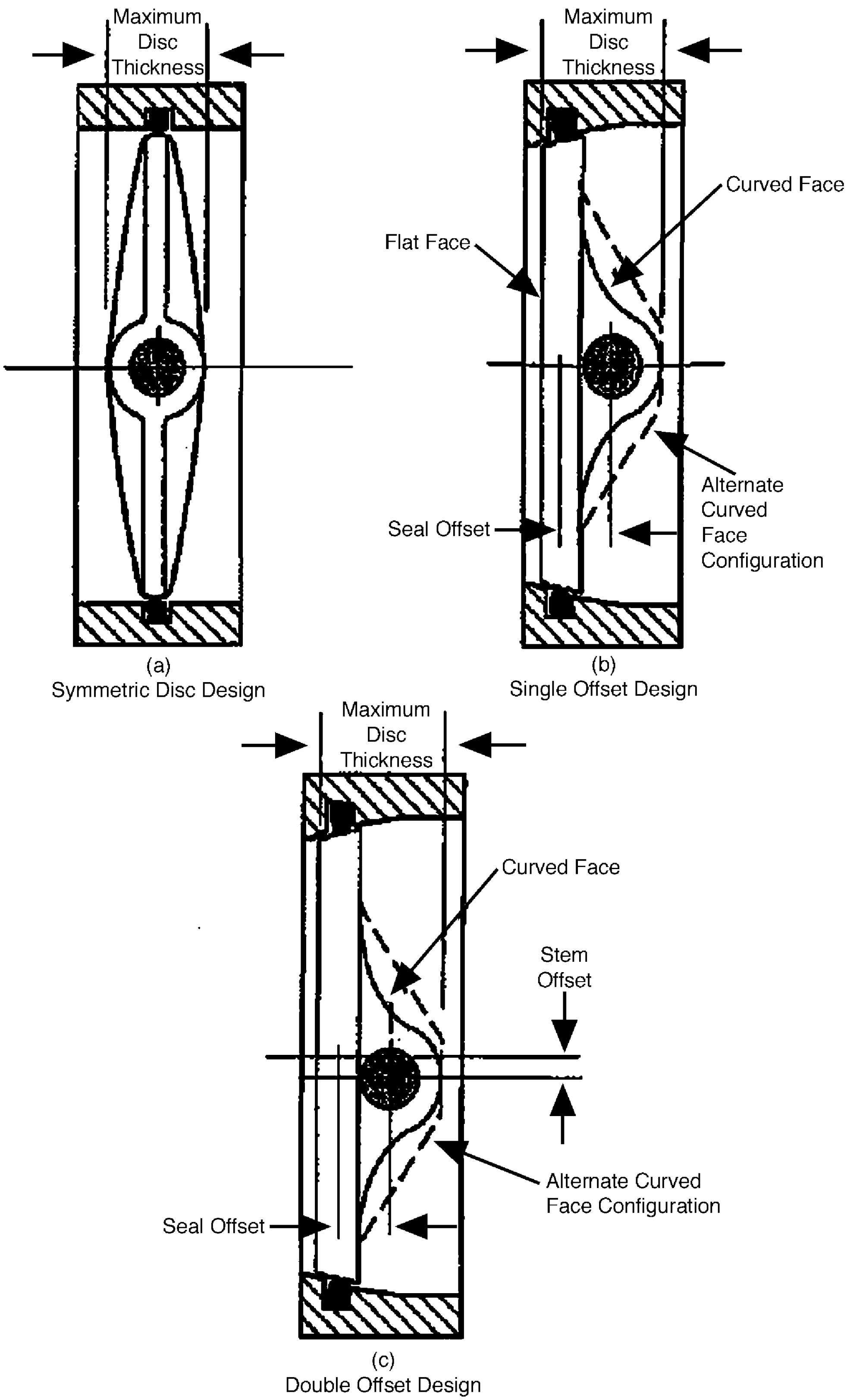


Figure 2-2
Typical Variations in Butterfly Designs

2.4 Disc

The most common butterfly valve disc shapes used in U.S. nuclear power plants can be divided into two basic disc designs: conventional symmetric (concentric) disc and non-symmetric (offset or eccentric) disc designs as shown in Figure 2-2.

2.4.1 Symmetric (Lens Type) Disc with Concentric Shaft

The symmetric disc type design shown in Figure 2-2(a) is generally referred to as the standard disc, conventional disc, or lenticular disc. Flow and total torque characteristics of a symmetric disc valve do not depend on the flow direction; thus, the valve can be installed either way without affecting performance. Symmetric disc designs are typically furnished with a rubber-lined body to provide a seal in the fully closed position, as shown in Figure 2-3. In this design, the shaft penetrates the rubber liner. An enlarged hub area around the shaft is provided with an interference fit with the rubber liner to prevent leakage around the shaft in the fully closed position. The disc hub area maintains a continuous contact against the body liner throughout the disc rotation. Normally, these valves are limited to low-pressure service (Class 300/PN 50). However, some manufacturers produce Class 600/PN 100 valves of this design.

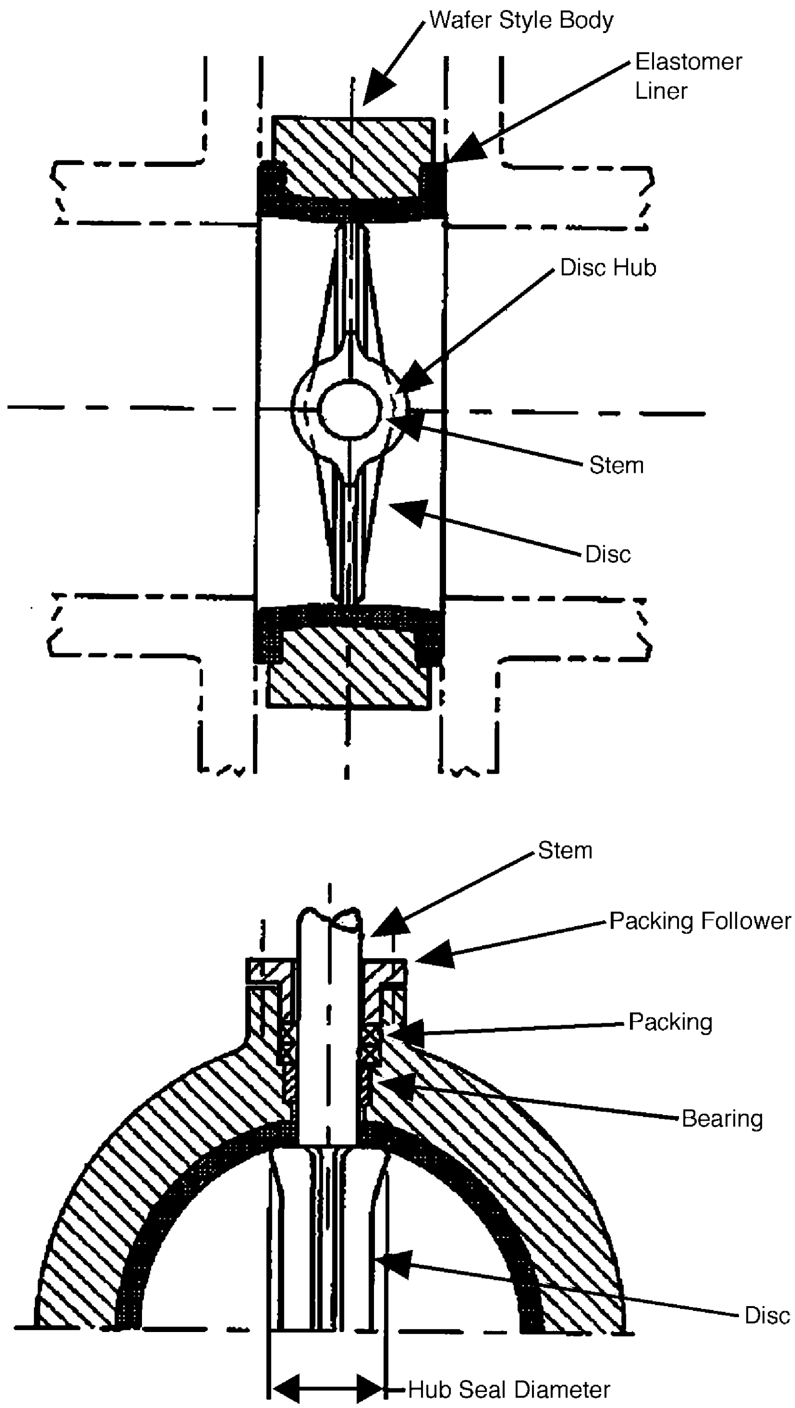


Figure 2-3
Typical Symmetric Disc Design with Elastomer Lined Body

Application considerations for symmetric disc include:

- Simple and compact construction compared to non-symmetric disc butterfly valves.
- Bi-directional service allowed due to the symmetric disc shape.
- Smaller dynamic torque in the closing direction than in the opening direction because the hydrodynamic torque is typically self-closing (see *Application Guide for Motor-Operated Valves in Nuclear Power Plants, Volume 2: Butterfly Valves* [1] for a discussion of hydrodynamic torque). This is particularly beneficial in applications where the isolation valve is required to close.
- Sliding action of interference fit between the seat and disc causes higher seat wear than in non-symmetric disc valves particularly in the presence of dirty water and marine growth.
- The disc hub area of the body liner will wear more and be an initial source of leakage because the disc hub is in continuous contact with the liner. It is also possible to have tearing of the liner in this location.
- The elastomer can take on a permanent set, thus creating a small bulge in the liner for those valves in normally closed position for extended periods. This can form an obstruction that will resist opening, possibly causing the actuator to stall (torque switch actuation) when opening.

2.4.2 Non-Symmetric Disc with Single Offset (High-Performance Butterfly Valve)

The single offset non-symmetric disc design, Figure 2-2(b), appeared sometime in the early 1960s. Another term used is eccentric design. Because of the improved sealing characteristics over the symmetric design, it was possible to build offset valves capable of higher pressure ratings. This resulted from the fact that uniform 360° sealing of the disc/seat, as well as improved leak tightness, could take place because of the elastic deflections due to differential pressure across the disc and shaft. The first Class 600 (PN 100) valves began to appear commercially because of this change. In the 1970s, some manufacturers began to refer to the design as a high-performance butterfly valve—a term that survives to this day. *High Pressure-Offset Seat Butterfly Valves* [7] provides additional information about construction of these valves. As shown in Figure 2-2(b), the shaft centerline (and the center of disc rotation) is offset axially from the plane of the valve seat along the pipe centerline. The valve seat is a continuous cone shape (versus the cylindrical shape used for symmetric designs). Another variation is shown in Figure 2-4. This design is available in resilient as well as metal-to-metal seat. The disc face away from the shaft is typically flat or has a small curvature and is commonly referred to as the flat face. The other disc face is generally convex and contoured to accommodate the shaft. This face is generally referred to as the curved face or shaft side of the disc.

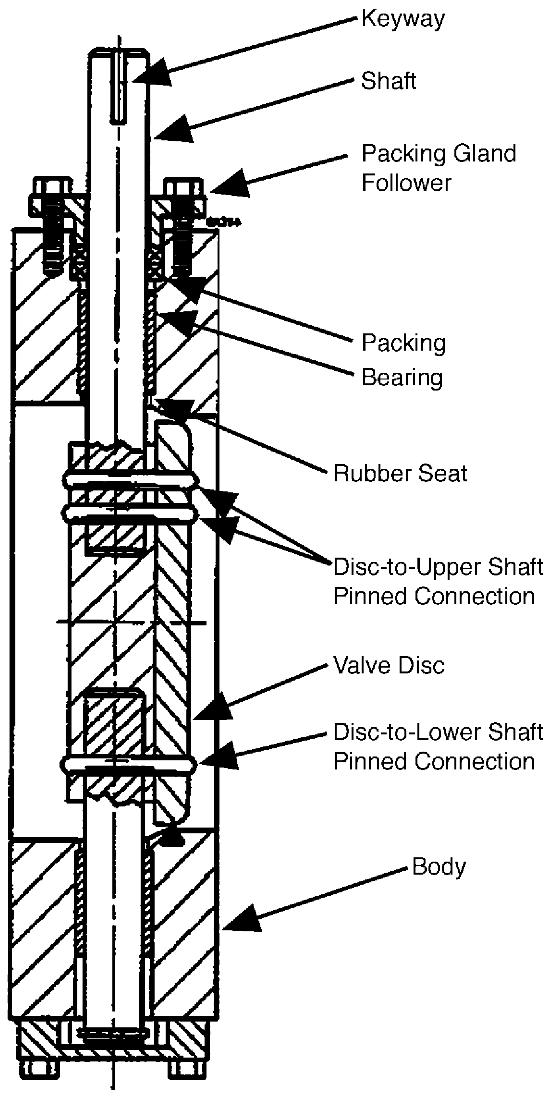


Figure 2-4
Cross Section of a Typical Non-Symmetric Butterfly Valve

Flow and torque characteristics of the valve depend on the flow direction with respect to the disc. When the shaft is on the downstream side (or the flat face of the disc is on the upstream side) of the flow direction, the installation is commonly referred to as shaft downstream or flat face forward (Figure 2-5). Similarly, when the shaft is on the upstream side (or the curved face of the disc faces the upstream side), the installation is referred to as shaft upstream or curved face forward.

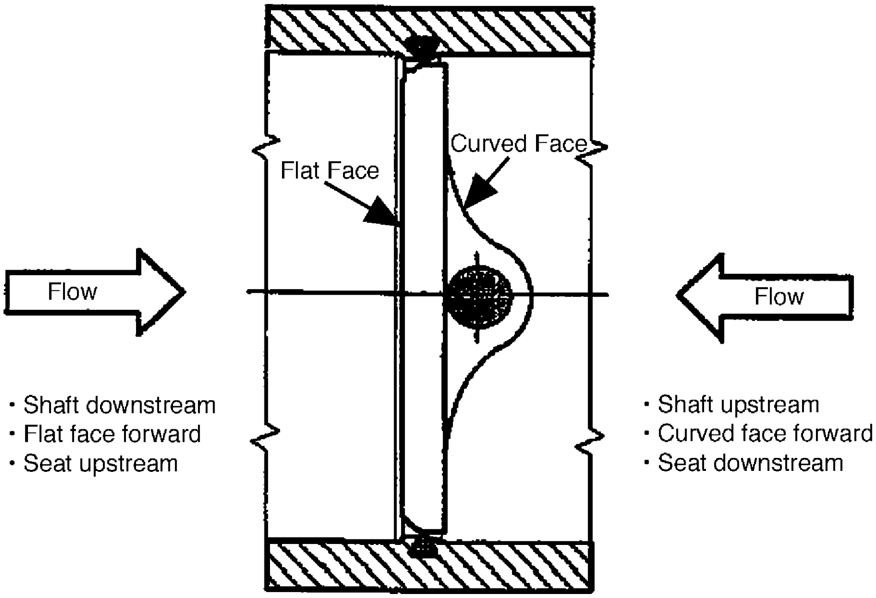


Figure 2-5
Valve Disc Flow Orientation Terminology

Application considerations for non-symmetric valves include:

- If the shaft is installed upstream, improved leak tightness can be obtained because elastic deflections due to differential pressure across the disc and shaft tend to close up the clearances between the seat and disc mating surfaces. However, torque requirements are higher than shaft downstream.
- If the shaft is installed downstream, the packing is isolated from the higher pressure and the hydrodynamic torque requirements are lower. With the packing isolated, on-line packing replacement is possible. However, leak tightness is smaller than shaft upstream.

2.4.3 Non-Symmetric Disc with Double Offset Shaft

In this non-symmetric disc with double offset shaft design, the shaft has a seat offset (similar to the single offset design) and a relatively small lateral shaft offset as shown in Figure 2-2(c). The seat is also a cone shape. The magnitude of the shaft offset varies from 1/32 in. to 1/8 in. (0.8 mm to 3.2 mm). The additional offset provides less interference during seating, thus reducing seat wear. This design is available with resilient seats as well as metal seats. In the double offset design, the resultant force due to differential pressure across the disc in the closed position does not pass through the shaft centerline.

Application considerations for non-symmetric discs with a double offset shaft include:

- The double offset provides a cam-like action that has less seat wear than the single offset design and enhanced sealing capability in certain applications.
- For the shaft-downstream installation, an external torque may be required to prevent disc opening from the closed position due to differential pressure in designs that have large disc offset [1].
- The considerations noted earlier for the single offset design apply to the double offset design as well.

2.4.4 Non-Symmetric Disc with Triple Offset Design

Another variation in disc design that has been common for many years in European applications, but relatively uncommon in U.S. nuclear power plants until the 1990s, is the triple offset seat design shown in Figure 2-6. The main feature of this design is that, in addition to the seat and shaft offsets described above, the shaft has an additional (third) offset with respect to the disc centerline. This offset requires that the seating cone be rotated and an offset applied to the cone axis. This results in totally eliminating seat closing friction. It provides a tight seal due to the fact that the disc rotates into the seat as opposed to across the seat. (Note the similarity to a tilting disc check valve.) In other words, it is torque seated (in contrast to the other three disc designs shown in Figure 2-2, which are all position seated). This obviously has enormous benefits from a seat leakage perspective. With position seating, it is necessary to rely strictly on the seat seal design. Turning the shaft past the closed point simply opens the valve and increases leakage. With the triple offset, it is not possible to go through the seat. Additional force increases the sealing action. Although Figure 2-6 is correct as an illustration of how a triple offset works, most manufacturers do not have the seating (disc) plane at an angle. Rather they keep the disc plane normal, allowing the flow and machine of the seat and disc at an angle. This is shown in Figure 2-7.

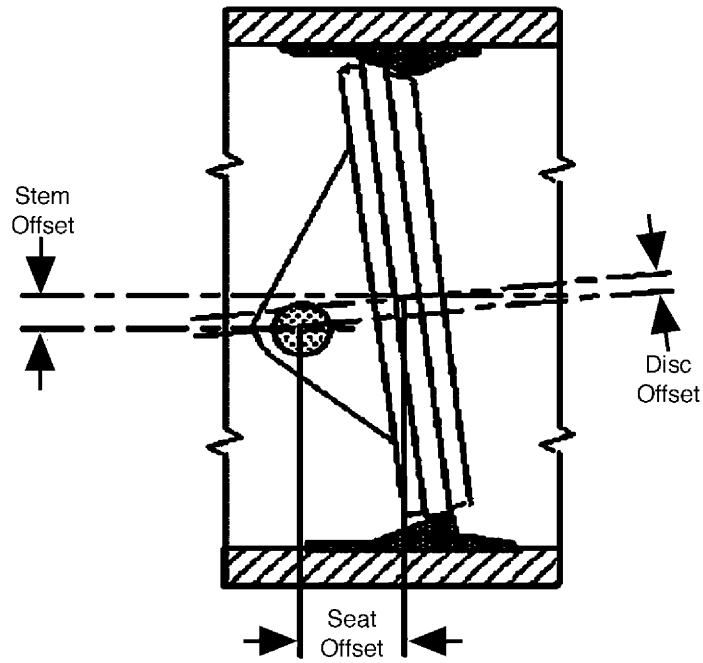


Figure 2-6
Triple Offset Butterfly Valve (Functional Depiction)

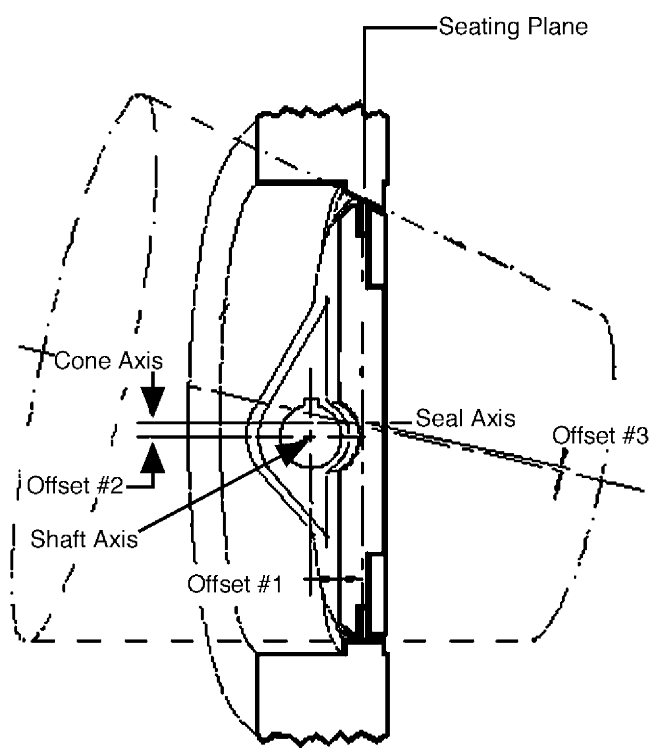


Figure 2-7
Triple Offset Butterfly Valve (Usual Manufacturing Configuration)

Application considerations for triple offset valves include:

- Because triple offset valves are provided with torque seating rather than position seating, they are more likely to achieve little or no leakage.
- With metal-to-metal seats, higher temperatures are possible with low leakage.
- The geometry of the disc/seat is cone-shaped, and the disc rotates into the seat without sliding across the seating surface. Seat wear is minimized and long life is possible without the deterioration of the seat. This characteristic, along with the low leakage capabilities, makes this design ideal for containment isolation valves.
- With appropriate cone angles, unseating forces will be lower than seating forces, resulting in a valve less likely to be stuck closed.
- Compared to other butterfly designs, triple offset valves have higher initial costs.

2.4.5 Special Discs

Butterfly valves with special disc designs are used primarily in throttling service or to decrease the torque required to actuate the valve. These special designs include fishtail discs (see Figure 2-8) for torque control and serrated edge or orifice discs (see Figure 2-9) for flow, noise, and cavitation control. In general, these valves are not required to give zero leakage in the fully closed position and are used primarily in a fully open or partially open position. In most cases, the shape of the disc makes the valve unidirectional.

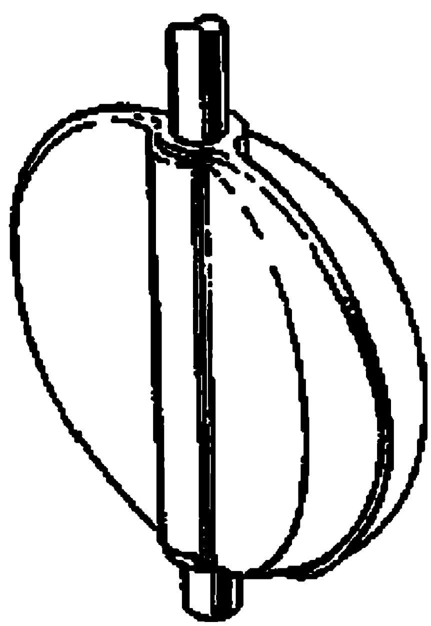


Figure 2-8
Fishtail Disc

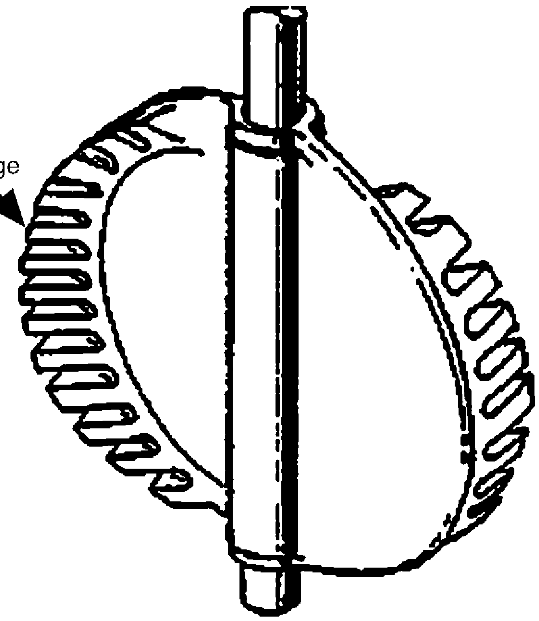


Figure 2-9
Special Disc Design for Noise and Cavitation Reduction

2.5 Valve Shaft, Shaft Connections, and Seal

2.5.1 Shaft and Shaft Connections

Butterfly valve shafts are designed to transmit actuator output torque to the valve disc and to support the disc against fluid-induced and seating forces. The valve shaft in large butterfly valves is normally the two-piece shaft design (stub-shaft type) as shown previously in Figure 2-4, although a one-piece design is sometimes found in smaller sizes. The engagement length of the shaft-to-disc connection is usually 1.5 shaft diameters in two-piece shaft designs. The other end of the upper shaft is connected to the actuator by a single key, double key, spline, square head connection, or other special design. The lower shaft usually is not connected to the disc. The shaft is supported on both sides of the disc by sleeve bearings and, with few exceptions, is connected to the disc by pins, keys, or both.

2.5.2 Disc Pin Shaft Connections

Disc pins are used to connect the disc to the shaft and transmit the actuator torque. They are considered to be part of the valve trim, which is the metallic valve material exposed to the process fluid. The choice of valve trim depends on various parameters including fluid temperature, chemistry, and velocity. For large butterfly applications, fluid chemistry is the primary consideration in order to prevent corrosion. However, fluid chemistry is known only to the user. Therefore, trim selection must be a joint effort between the user and manufacturer. The manufacturer has developed trim sets of materials that have well-known corrosion resistance properties. These properties consider both the individual resistance as well as the collective resistance (that is, galvanic corrosion). Typically, corrosion problems occur in open loop water systems where the chemistry was not properly identified. For example, a power plant near the ocean that uses river water for its service water and river-water chemistry for specifying valve trim can actually have brackish water conditions.

Pins connecting the disc to the shaft generally have the following characteristics:

- The pins have strength adequate for the loads applied.
- The pin material is adequate for the environment both individually and collectively with other trim materials (galvanic corrosion). This material is commonly the same as the shaft material.
- The fit between the shaft, pin, and disc is close enough to prevent fatigue failure from vibration.
- The pin is prevented from coming loose.
- The pin is removable with ordinary tools or grinder (required only for ease of maintenance).

Pin materials are usually made from stainless steel. These include 300 series, 400 series, and 17-4 PH. Other high-alloy materials are offered.

One or more pins are used to couple a disc to the shaft, depending on the service. There is normally no pin in the lower shaft of a two-piece design. Typically, a pin is a taper design or, to a lesser extent, a dowel. Nearly all taper pins are round. However, some have a rectangular cross section with only one side tapered and are inserted into similar holes in the shaft. Pins pass either through the centerline of the shaft or at a tangent to the outside diameter surface of the shaft.

Taper pins are used for several reasons:

- The tape angle used (less than $\approx 16^\circ$) results in a joint that is self-locking (that is, the frictional forces holding the pin in place are greater than the reactive forces left over from inserting the taper).
- The pin can be driven out easily (that is, the force needed is less than what was used to insert the pin).
- The pin can be secured from one end. If designed properly, securing can enhance the amount of force holding the pin in place.

All pin designs are created to be tight after insertion to prevent fatigue failure caused by vibration and valve operation. To keep the pin tight, the pin is secured. Securing the pins can be done in different ways. Taper pins can be inserted in tapered holes or grooves (tangential) and typically welded at the small (or both) ends. In some cases, the small end of the pin may be threaded and designed to protrude through the shaft where a washer/nut is attached. Normally, non-tapered (straight) pins are cylindrical and held in place in a similar way. Some designs use staking to secure the pin.

2.5.3 Shaft Sealing

Sealing of the shaft in butterfly valves (and quarter-turn valves in general) is easier than that for rising shaft valves such as gate and globes. The most commonly used butterfly valve shaft seals are:

- **Pull-down packing gland (stuffing box).** Both live-loaded packing (for example, with Belleville springs) and conventional bolt torque preloaded pull-down packing glands (as shown in Figure 2-4) are used with butterfly valves. The stuffing box is usually designed to accept a minimum of four packing rings. Flexible graphite packing rings with composite carbon/graphite end rings are commonly used in this type of design. Also available are PTFE and PTFE-impregnated graphite for both sealing capability and reduced torque.
- **V-type (self-adjusting chevron type) packing**. V-type packing is well suited for quarterturn valves in general. Line pressure acts on the inside surface of the V-rings to create a seal across the shaft. Therefore, correct orientation of the V-rings in the packing cavity is required (inverted V). Ethylene propylene terepolymer (EPT) rubber and composite PTFE are the typical packing materials for this type of packing design. The shaft seal design for butterfly valve applications should allow for easy packing replacement and in-service adjustment.
- **Mechanical O-ring cartridge.** This type of shaft seal is not common. This is due to its inability to stop leakage (no adjustment), difficulty in replacement (must remove motor operator) online, and short service life due to high abrasive content of river water.

2.6 Valve Bearings

As shown in Figures 2-3 and 2-4, sleeve-type bearings are used to support the valve shaft against forces due to differential pressure across the disc assembly. These bearings are installed in the valve body hubs or trunnions. Corrosion-resistant and self-lubricating bearing materials (such as solid bronze, graphite-impregnated bronze, and PTFE-impregnated fabric with stainless steel backing) are commonly used. Stainless steel bearings (often with some surface treatment) are also used in some applications. Metal type sleeve bearings are typically designed such that the shaft-to-bearing contact stresses do not exceed one-fifth of the compressive strength of the bearing or shaft material at operating temperature. Frequency of stroking can be the determining factor due to wear considerations when choosing between metal and non-metal bearings.

Operating experience shows that the bearing/shaft coefficient of friction does not exceed 0.25 throughout the design life of non-stainless steel bearings in clean water service. For stainless steel bearings, however, the coefficient of friction can be as high as 0.6. A lower coefficient of friction (0.15 or even less) can be obtained with PTFE and other self-lubricating reinforced plastic bearing materials in clean fluid applications. The effects of raw water (that is, river or lake water) is not clear as reported by *EPRI MOV Performance Prediction Program: Friction Coefficients for Non-Metallic Butterfly Valve Bearing Material* [9]. The valve manufacturer should be consulted for the bearing material coefficient of friction or the bearing friction factor applicable to their specific valve designs [1, 9].

In addition to the sleeve bearings, which carry the forces induced by differential pressure across the disc, valves larger than 20 inches (DN 500) are typically equipped with one or two thrust bearings to support the weight of the disc assembly and to keep the disc centered with respect to the seat. The torque contribution from these thrust bearings to the total operating torque requirements is normally negligible.

2.7 Valve Seats

Valve seat design can be metal-to-metal seal or soft-seal using elastomers or plastics against metal. Most valves in a power plant use the soft-seal design due to the need for low leakage characteristics.

2.7.1 Metal-to-Metal

Metal-to-metal seats are suitable for high temperatures and offer greater wear characteristics. They are less likely to be affected by marine growths that can collect at the seating surfaces. On the other hand, metal-seated valves require significantly higher torques for tight shutoff, adversely affecting actuator sizing and cost. This includes the torque seated triple-offset valves.

2.7.2 Rubber Seats

Rubber seats in butterfly valves are usually in the 40–80 durometer hardness range (based on shutoff pressure requirements) with 65–70 durometer being the most typical. Continuous exposure to high temperature or certain fluid environments (such as air) can cause hardening of the rubber material with age, thus causing an increase in seating/unseating torque and loss of sealing performance. Valve manufacturers provide recommendations for the seat replacement frequency to ensure satisfactory seal service. However, plants must validate these frequencies and adjust accordingly.

The valve leak tightness requirements may dictate a preferred flow direction for non-symmetric designs. The shaft upstream (seat ring downstream) is the preferred direction from a sealing standpoint because elastic deflections due to the differential pressure across the disc tend to close up the clearances between the disc and seat mating surfaces, thus providing a tighter seal. On the other hand, torque requirements are higher [1].

A large number of combinations of valve seat designs and materials are available to meet the variety of applications and operating conditions. Figures 2-3, 2-4, and 2-10 show the variations most commonly found in nuclear power plant applications of butterfly valves. Based on seat leakage and seat torque requirements, valve seats can be divided into nominal leakage seats, low leakage seats, and tight shutoff seats. Except for externally pressurized elastomer seats of inflatable designs (Figure 2-11), which are not commonly used in nuclear power plants, tight shutoff seats require higher seating/unseating torque than the nominal leakage seat designs.

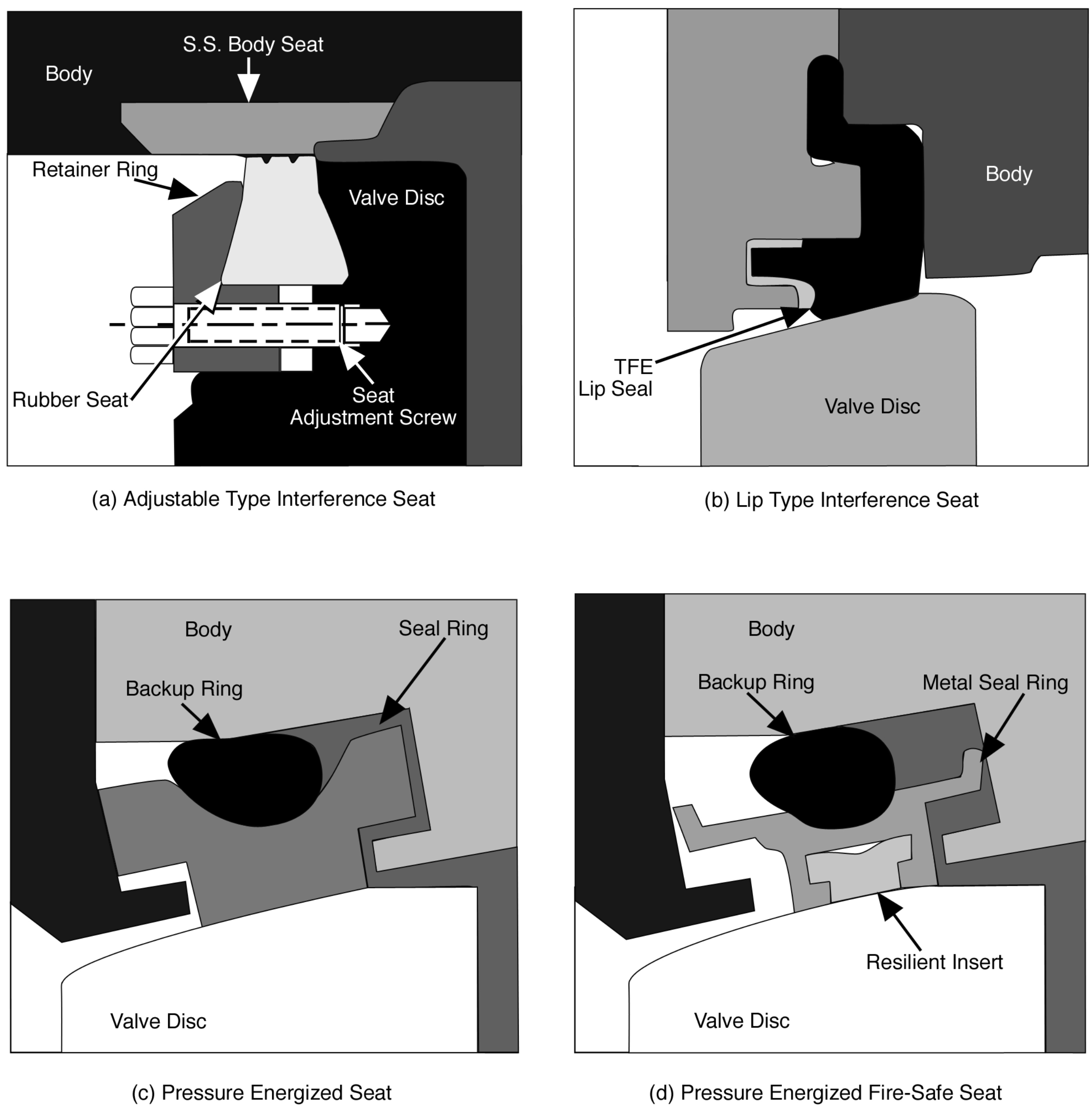


Figure 2-10
Typical Seat Designs

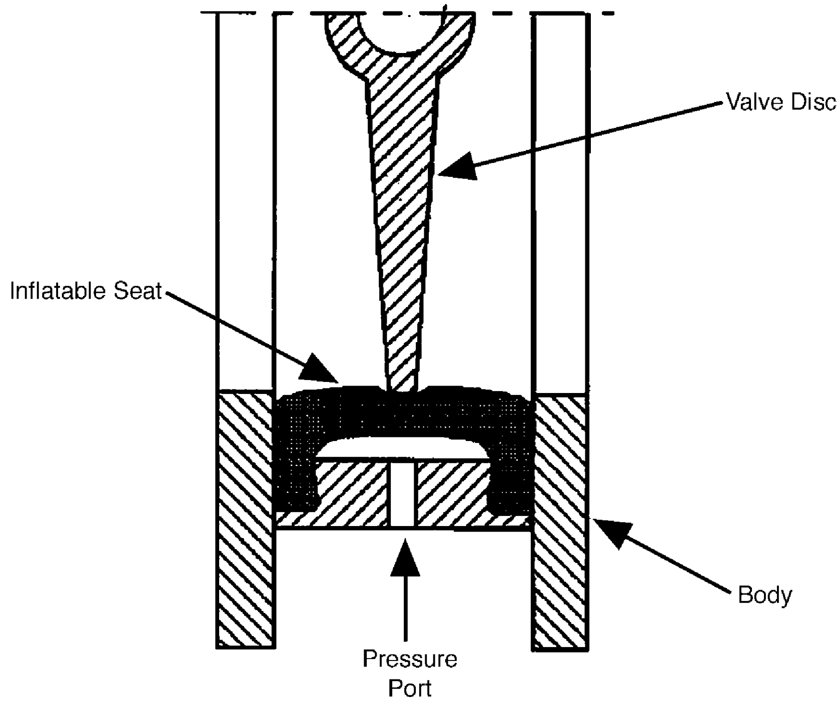


Figure 2-11
Inflatable Seat Butterfly Valve

The elastomeric seat can be disc mounted or body mounted. Disc-mounted replaceable seats depicted in Figure 2-10(a) are preferable from a maintenance perspective since they allow replacement without removing the valve from the piping. However, this may be irrelevant for larger sizes where the disc weight is considerable and the disc is difficult to remove from the valve in the field. Therefore, body-mounted replaceable seats would be satisfactory. (The AWWA standard [8] requires that rubber seats be replaceable at the installation site for valves 30 inches [DN 750] and larger.)

Stainless steel or nickel-copper alloy seating surfaces are recommended for frequently operated valves (more than once a month). Rubber seats should be resistant to microbiological attack and ozone attack. Provisions should be made for ease of maintenance (for example, adjustment or replacement of seats) by providing proper access to the valve.

Sealing in the fully closed position is achieved by proper contact load between the sealing surfaces on the disc and body. Descriptions of the most commonly used methods to cause this contact load follow.

2.7.2.1 Interference Type Seats

In interference type seats, sealing is achieved by elastically deforming the seat. The amount of interference is preset to ensure sealing under the design differential pressure. Figure 2-3 shows the elastomer-lined body design in which the liner also acts as the interference type seat. Figure 2-4 shows a non-symmetric valve with an interference seat. This type of seat must have an adjustment method to properly seal. Figure 2-10(a) shows the normal method of adjusting this type of seat. The adjusting screw exerts force on the elastomer. Because the elastomer is incompressible, it is forced into the seat area. (While the figure shows the elastomer on the disc, having the elastomer mounted to the body is just as common. The mechanism of adjustment is the same.)

The amount of seat adjustment is a compromise. If seat tightness is important, higher seal forces and hence higher adjustment torques are required. However, as the seal force is increased, the amount of actuator seating torque is increased. Other factors come into play as well. These include elastomer relaxation (creep) and the fluid chemistry compatibility. Seat adjustment is discussed further in Section 3.3.

2.7.2.2 Self-Energized Type or Pressure-Energized Seats

Figure 2-10(b) is a lip-type self-energized seat. It substitutes the adjustment by using a seat that is designed to be more flexible and thus conforms more easily to the metal surface. Figures 2-10(c) and 2-10(d) show two commonly used designs of pressure-energized seats. In pressure-energized seats, the line differential pressure at the fully closed position is used to generate proper contact load between the seat and its mating surface. The seal ring is typically made of reinforced PTFE or other composite elastomer/plastic material and has a shape that permits the upstream high-pressure fluid to increase the contact pressure at the sealing interface. Figure 2-10(d) shows a pressure-energized metal seat design that provides a tight shutoff and meets the fire-safe sealing requirements.

2.7.2.3 Inflatable Type Seal

In inflatable type seat designs, air is supplied under the resilient seat member after seating and is removed before unseating the disc. The air is supplied through a valve that is cammed open or closed as the shaft turns. Figure 2-11 shows an inflatable type elastomer seat design for a symmetric disc valve. Typical materials used for the seal ring in the elastomer type seal design are EPT or nitrile.

2.7.2.4 Material Limitations

Selected elastomer material limitations are shown in Appendix B. This information is provided for comparison only. An engineering review is necessary before using them for design purposes. For safety-related applications, the equipment qualification report governs.

2.8 Manual Actuators

A manual actuator can be as simple as a lever arm connected directly to the shaft. In larger valves, it can be a hand wheel connected to a worm gear (see Figure 2-12) or a traveling nut (see Figure 2-13) to produce the required torque.

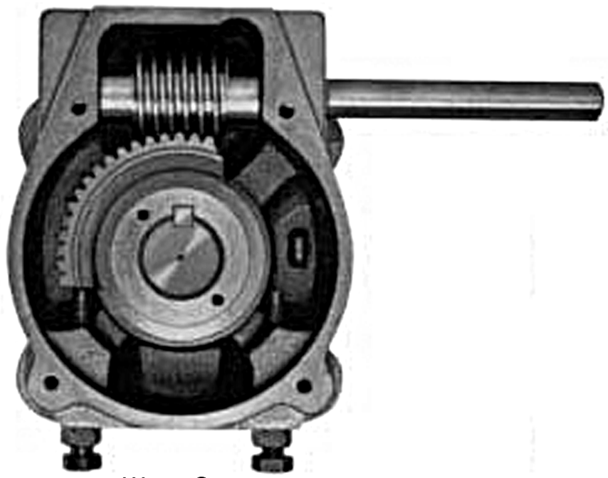


Figure 2-12
Worm Gear Manual Actuator

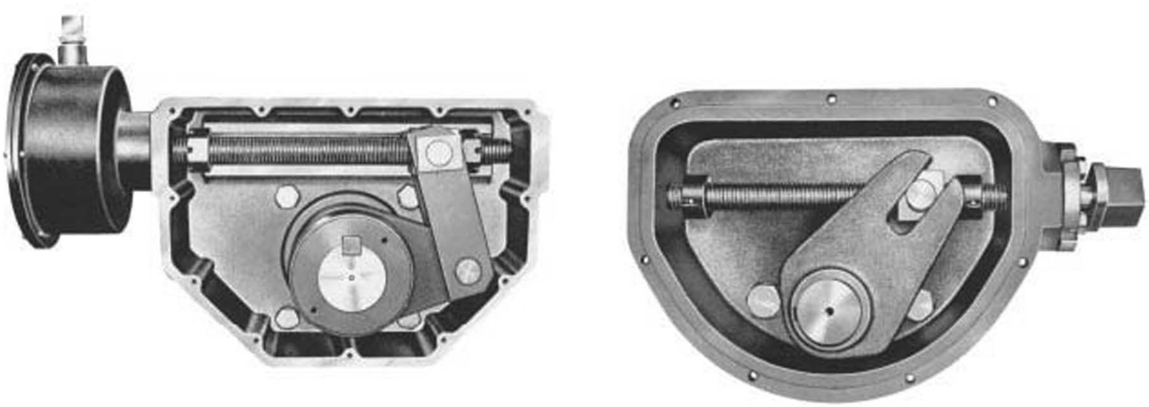


Figure 2-13
Traveling Nut Manual Actuator

2.8.1 Operating Force

Manual actuators should not require more than 60 pounds (270 N) of operating personnel force during the majority of the travel and 150 pounds (670 N) of peak force to complete the stroke. Table 2-1 provides the average tangential force and the corresponding torque values as functions of the hand wheel diameter. These values are based on tests performed by the U.S. Navy. They represent the hand wheel rim pull achievable by an average person. The effort that can be exerted by an average person also depends on the orientation of the hand wheel relative to the person.

Table 2-1 can be used to judge the results of increasing the size of a hand wheel for a valve that is difficult to open if it has been determined that additional torque is necessary and structurally allowable. The size chosen should not hinder normal access to the valve.

Table 2-1
Hand Wheel Torques

Hand Wheel Diameter in. (mm)	Average Tangential Force from Tests lb (N)	Resulting Torque ft-lb (N-m)
Less than 4 (100)	50 (220)	Less than 8 (11)
4 to 6 (100 to 150)	60 (270)	10 to 15 (14 to 20)
7 to 9 (180 to 230)	100 (440)	29 to 38 (39 to 51)
10 to 14 (250 to 360)	125 (550)	52 to 72 (70 to 98)
15 to 23 (380 to 580)	145 (640)	90 to 136 (122 to 184)
Over 24 (610)	150 (670)	150 (203) and higher

2.9 Operation Practices and Precautions

2.9.1 Hydraulic Effects

Depending on the design and direction of flow, butterfly valves can open or close by themselves under flow conditions. Therefore, care should be used when operating a lever-operated manual butterfly valve to prevent personal injury. Most worm gear operators have self-locking gear trains to prevent the valve disc from drifting.

2.9.2 Closing a Leaking Valve

When a butterfly valve is leaking, it should be verified that the valve is not in the closed position before attempting to close it further. If it is in the actual closed position, trying to close it further will open it (providing there are no positive stops), making it leak more. Plant personnel who are operating valves should receive training to ensure that they understand this property of the butterfly valve. Also, if the actuator does not have position indication, the shaft should be match-marked to the valve neck or trunnion with an approved marker when the valve is in the closed position.

3
BUTTERFLY VALVE TROUBLESHOOTING, PROBLEMS, AND REPAIR

3.1 Introduction

Section 3 is based on butterfly valve problems that have been identified through a review of operational experience events. The complete analysis is shown in Appendix C. Section 3.2 discusses the troubleshooting process. Section 3.3 provides information on troubleshooting specific failure modes as identified by the operation experience review. Methods for prevention and repair are also discussed.

3.2 Troubleshooting Process

Troubleshooting is the systematic approach to data collection, failure analysis, and a test/measurement plan that results in high confidence that the complete cause of system or equipment degradation has been corrected and that the system or equipment has been restored to normal operation.

The formal process for troubleshooting has been documented in EPRI's *System and Equipment Troubleshooting Guide* [10]. The process is divided into two main parts: a preliminary evaluation and the formal process of troubleshooting. The discussion of preliminary evaluations describes the following:

- Identifying the issue.
- Defining the problem.
- Determining and validating operating conditions.
- Comparing to previous conditions to determine if symptoms adversely affect system/component performance or reliability. If this last step is confirmed, then detailed troubleshooting begins.

Detailed troubleshooting consists of the following:

Step 1: Performing a system walkdown and collecting additional system and component data.

Step 2: Identifying failure modes and system effects.

Step 3: Developing and implementing a troubleshooting plan.

Step 4: Determining if results identify the cause(s) of the problem. If so, performing the corrective actions; otherwise, collecting additional data.

Step 5: Determining if the corrective actions restore performance. If so, confirming if the root cause is discovered and documenting; otherwise, collecting additional data.

The next section of this guide supports Steps 2 and 3 and parts of Steps 4 and 5 of this detailed troubleshooting process.

3.3 Butterfly Valve Failure Modes and Discussion

The analysis provided in Appendix C identifies the following failure modes.

- Seat leakage 48%
- Failed to stroke 33%
- High operating torque 5%
- Failed stroke-time requirement 3%
- Leaking valve flange 2%
- Packing leakage 2%
- Other 6%

3.3.1 Seat Leakage

The failure mechanisms for seat leakage were as follows:

- Incorrect adjustment
- Foreign material
- Seat wear/aging
- Damaged seat
- Incorrect installation
- Actuator drive key failure
- Freezing conditions
- Other failures

3.3.1.1 Incorrect Adjustment

The leading cause for seat leakage was incorrect valve adjustment. Adjustment is divided into two broad areas. First, the disc position must be optimal for proper contact load with the seat. Second, the seat must be adjusted in some valves (see Section 2.7).

3.3.1.1.1 Disc Position

The exact position of the disc for zero leakage is difficult to achieve based strictly on valve position. In most cases, the only way to know that the valve is at the optimal closed position is by testing or visual observation. Testing is always preferred. This is especially true for pressure-energized seats that need enough differential pressure to properly seal.

Testing can be performed in place after reinstallation, but testing before reinstalling is much more reliable, regardless of valve size. During plant discussions, it was discovered that even 96-in. (DN 2400) valves were removed. Testing can be done in several ways. A very effective method is to install a blind flange on one side of the valve and pressurize the valve seat through a tap in the flange. This blind flange test can save considerable time because the initial adjustment is often not perfect, even on the valve models that have a procedure for setting adjustment screws.

Key Technical Point

A very effective method for testing a valve for leakage is to remove the valve, install a blind flange on one side of the valve, and pressurize the valve seat through a tap in the flange. This is a relatively easy process and allows adjustments to the seat position prior to reinstalling the valve.

A method to find the correct position in an installed valve is to use an ultrasound probe to listen to the valve as it is turned by hand with system fluid running through it. However, there can be considerable background noise, which sometimes makes this method difficult.

Another approach to achieving optimal seat closure when access to the seat is still achievable (and the use of a test plate is impractical) is to use a feeler gage or high-intensity light to check the gap between the seat and disc.

Once a sweet spot has been determined, it is important to effectively lock the valve in the correct closed position. The use of a gag is very effective. If this is not possible, match marks should be used as a minimum. The valve mechanical stops should be reset if available. If the actuator is mounted, this is also the time to set the valve actuator position limit stops. The valve should then be installed in the system carefully to avoid disc movement.

Key Technical Point

Once a sweet spot has been determined, it is important to effectively lock the valve in the correct closed position. The use of a gag is very effective. If this is not possible, match marks should be used as a minimum. The valve mechanical stops should be reset if available.

While *in situ* testing should not be necessary following this process (unless the valve is for containment isolation), some plants fine-tune to get zero leakage. It is important to remember that leakage can be reduced by opening the valve as well. It should be ensured that the adjustments are small and that the stops are reset. In any case, the adjustments must be done carefully or the disc can go through the seat.

Determining the seat position also applies if the seat has just been replaced. The sweet spot is specific to the installed ring. It must be verified when seat replacement has taken place.

Key O&M Cost Point

Determining the seat position also applies if the seat has just been replaced. The sweet spot is specific to the installed ring. It must be verified when seat replacement has taken place.

If the valve uses a motor operator, it is important to ensure that the mechanical stops and the limit switches are synchronized. Each time an adjustment is made, this synchronization must be maintained.

Key Technical Point

If the valve uses a motor operator, it is important to ensure that the mechanical stops and the limit switches are synchronized. Each time an adjustment is made, this synchronization must be maintained.

3.3.1.1.2 Seat Adjustment

There are two ways to approach the seat adjustment: 1) use a controlled uniform torque on the seat adjustment screws; or 2) begin with a uniform clearance between the seat and its mating surface and adjust each screw sequentially to achieve uniform contact with the surface. Following the latter method, a final incremental turn of each screw (that is, a quarterturn) is necessary to cause seat compression. Both methods accomplish the same goal. For initial tightening, both methods should achieve satisfactory results. Normally, the manufacturer's recommendation should be followed.

One effect that must be addressed in seat adjustment after the seat has been loaded is elastomer relaxation or creep. Creep is unavoidable in all elastomers. The effect of creep is to lower the seating forces, which in turn lowers the leak tightness margin and can allow leakage later. Creep progresses with time. However, most of it takes place within the first 24 hours of tightening. Therefore, retorquing or readjusting the seat approximately 24 hours later can result in better seat performance. A measure of creep is compression set. Comparisons of various butterfly material compression sets are found in Appendix B, Table B-4.

Key O&M Cost Point

Elastomers relax with time or creep. This results in lowering the seating forces and can result in leakage later. Most creep takes place within 24 hours. Therefore, retorquing or readjusting a seat 24 hours later can result in better seat performance.

3.3.1.2 Foreign Material

No instances of foreign material intrusion were associated with maintenance performance. Foreign material was transported from another location by the fluid flow. In one case, the valve was used for blowdown of the drywell after an integrated leak rate test (ILRT). Enough airborne material was transported to cause the subsequent local leak rate test (LLRT) failure. In another case involving a service water system, silt from outside the plant flowed in. Solutions varied, however. Because using the heating, ventilation, and air conditioning (HVAC) for an ILRT blowdown was unavoidable, the most practical recourse was to inspect and clean the valve after blowdown was complete before conducting the closeout LLRT. On the other hand, for the service water system, strainers were installed to prevent recurrence.

3.3.1.3 Seat Wear/Aging

All elastomeric materials naturally age. Aging is a chemical process. Aging results in changes to physical properties, which are usually detrimental to valve function. One physical property that deteriorates is the ability to conform to the disc surface and to return (that is, increase in hardness and decrease in resiliency.) Because of this deterioration, leakage becomes more likely. While aging is an inevitable process, it can be accelerated by several factors:

- Temperature
- Oxidation
- Adverse chemical environment
- Radiation (gamma and UV)

Of all of these factors, temperature is the largest contributor to seat aging, followed by oxidation in HVAC systems.

Key Technical Point

Temperature is the largest contributor to seat accelerated aging, followed closely by oxidation in HVAC systems.

The thermal life of plastics and elastomers is inversely related to their operating temperature (that is, the higher the temperature, the shorter the material life). A rule of thumb, known as the *10°C rule,* suggests that a material's life is doubled for each 10°C (18°F) *decrease* in operating temperature. Conversely, the life is halved for each 10°C (18°F) *increase*. More accurate modeling of thermal life using other models indicates that the temperature effect can be even more pronounced than predicted by the 10°C rule. These models use 7–8°C (12–15°F) for halving/doubling material life.

The effect of oxidation cannot be discounted for some applications. HVAC system isolation valves are affected directly. The net result of the aging process is for the seat to become hard or brittle. This means that it will not conform as well to the metal side of the seat and so defeats the purpose of using the soft seat in the first place. Placing an elastomer in HVAC service can be compared with improper storage of the material (for example, on the shelf, removed from package). Some sources recommend a shelf life of three years after cure date when exposed to air [11]. This time frame illustrates the rapidity with which oxidation can degrade material. Of course, not all materials are equal. Table B-2 in Appendix B presents a list of materials ranked with respect to heat aging and oxidation combined.

Since plant valves are not internally exposed directly to sunlight, UV radiation is not a concern. Gamma and neutron do not typically present a problem in power plant during normal operation. However, gamma can be a great concern during accident conditions. Table B-3 in Appendix B presents a list of materials ranked with respect to the effect of radiation on tensile properties. Other factors can also come into play when considering radiation.

Special attention is normally given to chemical-containing systems (such as demineralized water system ion-exchange resin recharging) during design. Therefore, chemical compatibility is usually not a problem for most elastomers used in power plants.

3.3.1.4 Damaged Seat

Damaged seats were caused by high flow. The cavitation effects of high flow are severe and are discussed in Section 3.4.

3.3.1.5 Incorrect Installation

During testing, the valve was found to be leak-sensitive to the direction of flow. It could not be determined if this occurred due to a design or maintenance problem. As discussed in Section 2 (for non-symmetric valves), if the shaft is downstream (which lowers torque requirements) leakage is more likely. See Section 3.3.3 for a discussion of excessive torque.

3.3.1.6 Actuator Drive Key Failure

Drive key problems were primarily driven by the use of incorrect material, resulting in a direct failure or indirect failure due to corrosion. Incorrect material is typically a manufacturing problem, introduced by improper design (unlikely) or an incorrect substitution during manufacturing assembly. Plant maintenance can also be a factor with incorrect substitution. However, drive keys are designed to fail before other parts in an actuator. Therefore, if the drive key has failed, it can also be indicative of another problem such as excessive binding of the shaft.

3.3.1.7 Freezing Conditions

Although not identified in the failure analysis, one plant reported instances where water accumulated in the lower stub shaft and thrust-bushing region of the valve body. Even after draining the piping for cold weather operations, there was residual water in this location. When the water froze, the result was distortion of the brass bushing and shear failure of the pins that connect the bushing to the stub shaft. This resulted in the stub shaft dropping slightly, which pulled the disc off center by about 1/8 in. (3.2 mm). This resulted in damages (bends) to the taper pins that were holding the disc to the two stub shafts. The result was seat leakage.

3.3.1.8 Other Failures

- Liner separation from the body was seldom identified although it was discussed with plant engineers. It was usually the failure of the method of bonding of the liner to the body. In one case, it led to corrosion of the body and subsequent failure of the bearing surface due to corrosion product transport.
- One event consisted of the failure of the air supply to an inflatable seat. The air supply valve failed open, thus the seating surface was already inflated before the valve was closed. The disc could not fully close the seating area, and a leakage path existed.
- One report discussed the failure of the disc by splitting in half.

3.3.2 Failed-to-Stroke/Failed Stroke-Time Requirement

The failed-to-stroke and failed stroke-time requirements share similar failure mechanisms and are discussed jointly. Failed-to-stroke also includes partial stroking. These failure modes are usually the result of a piece part failure. The primary failure mechanisms include:

- Disc/shaft taper pin failure
- Actuator drive key failure
- Shaft binding
- Gear actuator failure
- Seat wear/aging
- Foreign material
- Damaged seat
- Incorrect operation
- Incorrect installation
- Incorrect adjustment
- Lack of stroking

3.3.2.1 Disc/Shaft Pin Failure

Pin failures were primarily the result of high flow/vibration-induced fatigue. High flow/vibration is discussed in Section 3.4 of this report. Incorrect material was also a problem. The following information discusses pin design in general and provides considerations for determining the correction of the failure.

As discussed in Section 2.5.2 of this report, shaft connections (for example, taper pins, straight pins, or key) are part of the valve trim. The pins connect the disc to the shaft and generally have the following characteristics:

- The pins have strength adequate for the loads applied.
- The pin material is adequate for the environment both individually and collectively with other trim materials (galvanic corrosion). This material is commonly the same as the shaft material.
- The fit between the shaft, pin, and disc is close enough to prevent fatigue failure from vibration.
- The pin is prevented from coming loose.
- The pin is removable with ordinary tools or grinder (required only for ease of maintenance).

Problems with pins could be traced to the fact that they did not meet one or more of these conditions. Failures analyzed indicated that the characteristics most likely to cause problems were those listed in the second and fourth bullets in this list. Either corrosion occurred, the looseness of the pin resulted in a loose fit and fatigue failure, or the pin simply fell out. Corrosion is discussed in the previous section.

3.3.2.1.1 Taper Pin

A taper pin permits a tight fit between shaft, pin, and disc and is easy to drive out. The taper pin must be secured from coming loose, however, because the residual reaction force remaining from driving it in will cause the pin to loosen with vibration. Securing is usually achieved by the use of a threaded end with a nut or with a tack weld at both ends. A threaded end has the advantage of allowing the taper pin to be pulled tight. It contributes to securing the pin. This is analogous to preloading a bolt. The tighter the pin, the less the joint is susceptible to fatigue failure (see Section 3.3.2.1.4 for a discussion of welding).

The taper pin connection can have a problem if the taper hole was not machined (reamed) in a single setup and/or if the shaft or disc clearance is excessive. This will result in the parts not being line-to-line and will permit movement between the parts, possibly resulting in fatigue failure.

3.3.2.1.2 Straight Pin

The straight pin is typically used with a tangential connection (see the next section) and is either designed with a threaded end and nut or an interference fit. A common method to achieve the interference fit is the roll-pin. It is also relatively easy to remove using a drift, but may have to be drilled or pressed out. Again, pin tightness is important to prevent fatigue failure.

3.3.2.1.3 Tangential Versus On-Center Connections

Most connections using the taper pin use an on-center connection in which the taper pin goes through the centerline of the shaft and disc. Correspondingly, the straight pin is found more commonly in a tangential application in which the pin passes through the small gap between the disc and shaft tangent to the shaft. The advantage of tangential connection is to spread the pin shear forces over a larger area and reduce the chance of fatigue failure. The use of the tangential connection should be considered if the on-center connection failures cannot be resolved.

Key Technical Point

Pin tightness is essential to prevent disc/shaft movement and resultant pin fatigue failure.

3.3.2.1.4 Welding

While welding can be an excellent method of securing certain joints, it can also be the reason for failures. When welding the ends of taper pins, it is important to consider the effects of shrinkage associated with any weld joint. This is particularly true on the large end where the shrinkage could actually pull the pin out slightly. The pin would not be loose, of course, but the connected parts could have been loosened and fatigue failure would be possible. In addition, if the weld joint is not properly prepared, locations of high stress can exist that cause the weld to crack. In addition, if the weld material, procedure, or performance of the weld has a problem, the weld may fail. This is particularly true for field repair.

Key Technical Point

While welding is an excellent method of securing pins, it can be the reason for failures because of the effect of weld shrinkage if not done properly.

3.3.2.2 Actuator Drive Key Failure

Drive key problems were caused by the use of incorrect material that had inadequate strength or inadequate resistance to corrosion. Incorrect key materials are typically a manufacturing problem, introduced by improper design (unlikely) or an incorrect substitution during manufacturing assembly. Plant maintenance can also be a factor with incorrect substitution. If a key failure has occurred, however, it can also be indicative of other problems such as excessive binding of the shaft or incorrect operation by overtorquing the valve.

3.3.2.3 Shaft Binding

This failure mechanism was due to corrosion product buildup in the packing area. Corrosion product buildup was nearly always the result of using an incorrect material. Most failures were associated with the packing gland (the valve part that compresses the packing). The glands were made of a non-stainless steel product (typically ductile iron). The material corroded because of a small amount of packing leakage. Any material that corrodes exhibits a volume increase. In this case, the buildup was sufficient to close the small clearance between the gland and shaft. As the material continued to corrode, it built up a compressive force against the stem. This resulted in the shaft and gland being frozen together. When the shaft was rotated, the operating torque was resisted by the follower studs that were connected to the gland follower.

Key O&M Cost Point

Most shaft binding failures were due to a packing gland made from a material that corroded readily in a water environment. The corrosion product (rust) filled the gap between the shaft and the gland, freezing the parts together. When the shaft was rotated, the operating torque was resisted by the follower studs that were connected to the gland follower.

3.3.2.4 Gear Actuator Failure

Gear actuator failure causes were not always clear. In one case, excessive shaft clearance was identified. The excessive clearance led to improper engagement of the worm and gear and subsequent failure. In another case, the drive sleeve material was improper and failed due to high stresses.

3.3.2.5 Seat Wear/Aging

An extensive discussion on causes of seat wear and aging are found under the seat leakage failure mode discussion in Section 3.3.1.

3.3.2.6 Foreign Material

This mechanism was associated with service water systems (for example, raw or untreated water). The material migrated from the system intake, but the cause for the debris was not reported. Service water systems that draw directly from a lake, river, or ocean should have strainers. However, small material sizes can sometimes cause failures. One plant has solved this problem by modifying the bearings of new and overhauled large butterfly valves to add dirt rings. Depending on the vendor, a pocket is milled in the valve body or a pocket is milled in the shafts to allow for installation of rubber O-rings inboard of the bearing. The O-rings provide a seal and prevent sand and sediment intrusion into the bearings.

3.3.2.7 Damaged Seat

A discussion on seat damage is found under the seat leakage failure mode discussion in Section 3.3.1.

3.3.2.8 Incorrect Operation

Although this mechanism had a very low frequency, discussions with some plant personnel indicated that they considered it a concern. Most valves in a plant are of the rising stem type—turning the hand wheel closed tends to close them more. When personnel do not know that the valve being closed is a butterfly valve (or when they do not understand the mechanics of the valve), it is possible for the disc to pass through the closed position and begin to open. (The exception to this is the triple offset design.) It is important that the type of valve is identified on the identification tag or the operating procedure, or that operating personnel understand butterfly design and be able to identify a butterfly valve in the field. In addition, match-marking the shaft/body hub when the closed position has been determined (see Section 3.3.1) is important.

Turning through the seat is not the only way to cause seat leakage. In one of the events, a mechanical leveraging device was used while trying to ensure that the valve was closed. The valve had mechanical stops, and the excessive force caused the valve disk to split down the shaft. When the valve was subsequently operated, it would not move. As explained before, it was assumed that the valve would seal better with additional torque as with rising stem valves.

3.3.2.9 Incorrect Installation

Incorrect installation can result in a torque requirement that exceeds the actuator output resulting in stroke failure. See Section 3.3.3 for a discussion of excessive torque.

3.3.2.10 Incorrect Adjustment

If the valve stops have not been adjusted properly, the valve can stroke past the closed position and become stuck. If excessive leakage is not noted, the failure will not be seen until the next attempt to open the valve. See Section 3.3.1 for a discussion on incorrect adjustment.

Although not reported in the event survey, one plant reported during plant discussions that the worm shaft on a Limitorque H6BC rolled off the worm gear because the mechanical stops were set incorrectly. The Limitorque HBC manual operator uses two types of mechanical stops. The H0BC-H3BC uses either a hex-nut or key-nut type stop mounted in an enclosure attached to actuator housing. This type of stop limits the number of revolutions of the worm (drive) shaft. The H4BC-H7BC uses a setscrew type position limit stop that directly prevents the worm gear from turning beyond a certain point. Therefore, it is very important to understand each design and have procedures that correctly set the open and closed position stops.

3.3.2.11 Lack of Use

For valves in normally closed position for extended periods, the elastomer can take on a permanent set, thus creating a small bulge in the liner. This can form an obstruction that will resist opening, possibly causing the actuator to stall (torque switch actuation) when opening. The preferred method to prevent this is to stroke the valve occasionally if possible. A change in seat materials can also be discussed with the vendor. See Table B-4 of Appendix B for comparisons of compression set resistance.

3.3.3 Excessive Torque

During the experience review, it was determined that excessive torque was caused by:

- Incorrect installation
- Seat wear/aging
- Shaft binding

In addition, insights from plant discussions suggest that the following should be considered:

- Higher than assumed bearing sleeve coefficient of friction
- High bearing loads
- Foreign material in bearing
- Liner deterioration
- Upstream component effects
- Shaft orientation (vertical or horizontal)
- Pressure locking

3.3.3.1 Incorrect Installation

As discussed in Section 2.4.4, flow and torque characteristics of the non-symmetric valve depend on the flow direction with respect to the disc. This difference and the reasons for it are discussed in Section 3.7.3.1 of EPRI's *Application Guide for Motor-Operated Valves in Nuclear Power Plants* [1]. This problem has also been identified to the industry through U. S. Nuclear Regulatory Commission (NRC) Information Notice 91-58 [12]. Since the valve can be installed either way, it is important to ensure the correct orientation before installing. When the shaft is on the downstream side of the installed flow direction, as shown in Figure 3-1, the peak dynamic torque can be significantly lower than when installed with the shaft facing upstream.

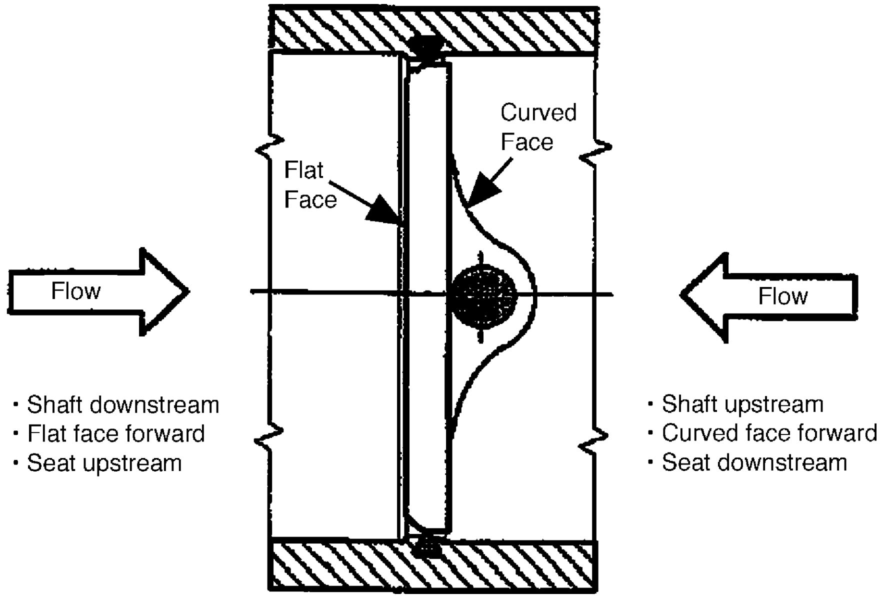


Figure 3-1
Valve Disc Flow Orientation Terminology

Another advantage of having the shaft downstream is that the stem packing is on the low-pressure side of a closed valve and a potential pathway for system leakage is eliminated. This feature can be important for such applications as containment isolation valves. If seat leakage is the only problem, and other options have been tried, facing the shaft upstream can provide a better seal because the differential pressure will move the disc toward the seat.

3.3.3.2 Seat Wear/Aging

As the seat material ages, it becomes hard. This can result in excessive seating torque requirement. See the discussion of seat wear/aging in Section 3.3.1.

3.3.3.3 Shaft Binding

In addition to the corrosion buildup phenomenon mentioned in Section 3.3.2.3 for shaft binding, another source of failure was corrosion products entering the bearing from another location in the valve. The event analysis showed that a liner had failed, allowing the carbon steel body to corrode. The corrosion products moved toward the bearing area. They were small enough to enter the annulus between the bearing and shaft, causing the bearing coefficient to increase. The following discussion on foreign material provides more detail.

As indicated previously, the following failures were not found during review. They are included because they have the potential to cause excessive torque requirements.

3.3.3.4 Higher-Than-Assumed Bearing Sleeve Coefficient Of Friction

The total torque required to seat or unseat the valve disc is the sum of four components:

- Bearing torque
- Seat torque
- Packing torque
- Hydrostatic torque

As discussed in EPRI's *Application Guide for Motor-Operated Valves in Nuclear Power Plants* [1], the bearing torque is directly proportional to the coefficient of friction. Coefficients of friction are always based on empirical information. For stainless steels (for example, 300 series, 17-4 PH), early assumptions put the value at about 0.25. EPRI's *Potential Inadequacies in the Prediction of Torque Requirements for and Torque Output of Motor-Operated Butterfly Valves* [13] reports the use of this assumption and indicates that dynamic testing determined the bearing coefficient of friction was 0.634 for a particular valve using stainless steel bushings. *Application Guide for Motor-Operated Valves in Nuclear Power Plants* [1] gives a value of 0.6 for non-bronze material (for example, stainless steel) and 0.25 for bronze bushings. Another document, *EPRI MOV Performance Prediction Program* [9] also provides coefficients.

3.3.3.5 High Bearing Loads

Related to the coefficient discussion above (but associated with new purchases) is an understanding of how small pressure drops can cause tremendous bearing loads. This condition is not always well understood by plant personnel. For instance, just an additional 10 psid (69 kPa) drop across a 72-in. (DN 1800) valve when it is closed results in almost 41,000 lbs (182 kN) of additional force to be supported by the bearings. This can result in an additional significant torque requirement for the actuator. (For coefficient of friction of 0.25 and a 6-in. (152.4-mm) stem diameter, this would add a bearing torque requirement of 5125 ft-lbs (6920 N-m). Therefore, when buying new valves, it is necessary to verify the torque requirements supplied with the various quotes to ensure that they are consistent with those discussed in EPRI's guidelines [1, 9]. (Sometimes an unrealistic bearing coefficient of friction is assumed in order to use a smaller actuator in a valve quote to be more competitive. This nonconservatism is easily overlooked because 10 psid (69 kPa) does not seem large.)

Key O&M Cost Point

A small additional pressure drop can cause tremendous additional bearing loads for large valves. A 10 psid (69 kPa) drop across a 72-in. (DN 1800) valve can result in a torque requirement of 5125 ft-lbs (6920 N-m).

Although not reported in the event survey, during plant discussions one plant reported that their circulating water pump discharge valves routinely experienced failures to fully stroke open. This was attributed to failure of the thrust collar. The valves were vertically oriented, with the shaft vertical. The original vendor noted that the original rolled pins used to secure the thrust collar to the shaft had degraded over time and allowed the shaft to move through the thrust collar under the weight of the disk. This resulted in high friction as the end of the shaft rotated on the endplate and some valve disc-to-body interference due to the slight misalignment. The valves stopped mid-stroke when stroked open.

Another factor that contributed to high torque loads was the stabilizing bushing used in the base of the valve operator stand. This was a bearing external to the valve body in addition to the two within the body. Over time, corrosion had fused the bushing to the shaft. When removed, it was discovered that the bushing was rotating in the floor stand bore, rather than the shaft rotating in the bearing.

3.3.3.6 Foreign Material in Bearing

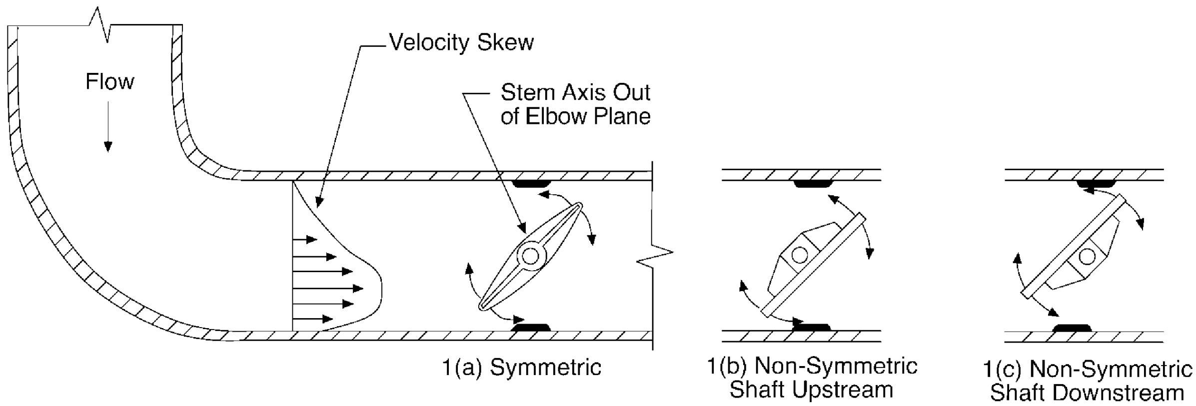
AWWA valves are designed to be used in fresh (potable) water service. Shaft bushings are typically PTFE-impregnated fiberglass bushing. If the valve is installed with the shaft in the vertical direction and used in open cycle systems, it can seize due to silt buildup between the shaft and the bushing. Frequent operation of the valve will avoid this. This type of failure was reported by one plant as causing plant shutdowns for up to two months.

3.3.3.7 Liner Deterioration

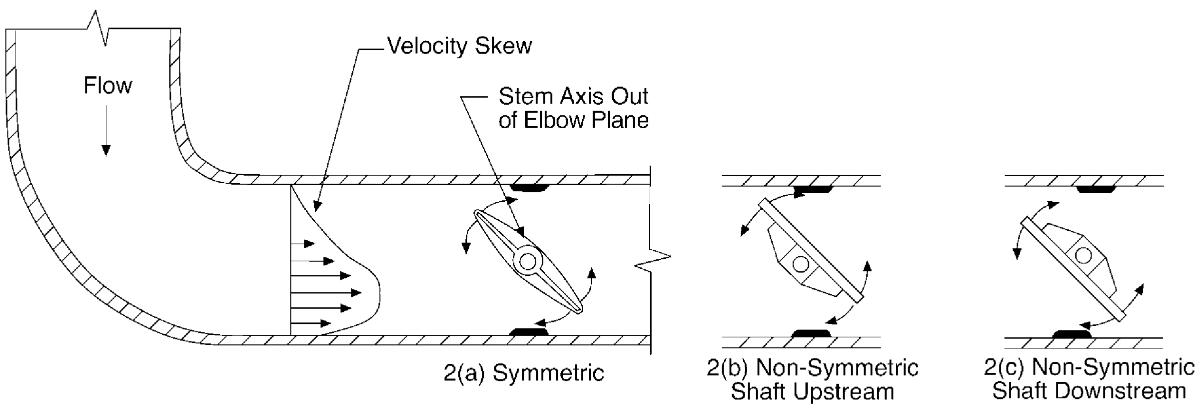
Liner deterioration due to aging on lined butterfly valves can occur if the liner material is not selected properly. This has been discussed in Section 3.3.1 (Seat Wear/Aging). Liner deterioration can also occur due to high fluid velocity or incorrect use as a throttling valve. Throttling can result in cavitation, which is a thermodynamic process that results in high impact bubble collapse on adjacent valve materials. No materials are immune to cavitation. Elastomers are highly susceptible. See Section 3.4 for an additional discussion on vibration and cavitation.

3.3.3.8 Upstream Component Effects

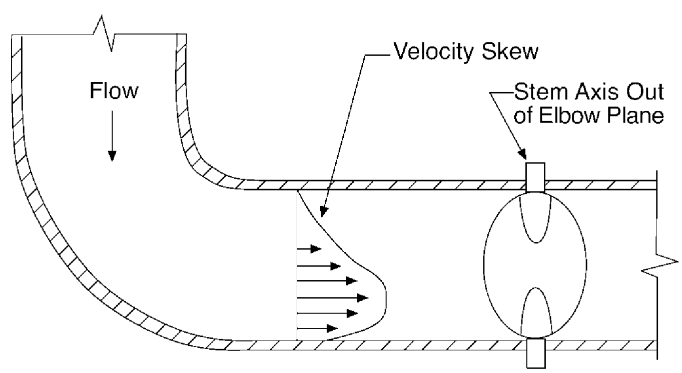
Ideally, butterfly valves should be installed in straight pipe runs with a minimum of eight pipe diameters of straight upstream pipes. However, in typical applications, butterfly valves are installed within short distances of other piping components that can have a significant effect on valve performance, especially in high flow rate applications. Upstream elbows, tees, pumps, and other valves result in velocity skews and turbulence, which can significantly increase the required dynamic torque. Figure 3-2 shows three different valve orientations with respect to an upstream elbow.



Configuration 1
(Velocity Skew Assists Closing Action)



Configuration 2
(Velocity Skew Opposes Closing Action)



Configuration	$C_{\mu p}$. max
1(a)	2.00
1(b)	2.00
1(c)	2.00
2(a)	1.60
2(b)	1.60
2(c)	2.00
3	1.25

Configuration 3
(Velocity Skew Symmetric About Stem Axis)

Figure 3-2
Effects of Upstream Disturbance, Shaft Orientation, and Disc Opening Direction on Hydrodynamic Torque

In configuration 1, the velocity skew tends to assist valve closing. In configuration 2, the velocity skew tends to oppose valve closing. In configuration 3, the velocity skew has little effect on the valve because the flow is nearly symmetric around the disc. Configuration 3 (where the valve shaft and the elbow are in the same plane) is typically recommended because it has the least effect on the valve performance in both the closing and opening direction.

An upstream elbow model has been developed to bound torque requirements with an upstream elbow in a given orientation and proximity from valve inlet [1]. The upstream elbow model can be used to estimate the effect of other upstream piping components on butterfly valve torque requirements. As indicated above, the effect of an upstream disturbance diminishes after 8 to 10 pipe diameters. However, two out-of-plane elbows produce a swirl that can persist for more than 20 pipe diameters.

3.3.3.9 Shaft Orientation (Vertical or Horizontal)

In offset valves, shaft upsteam or downstream is important for seat leakage and torque requirements. On the other hand, vertical or horizontal orientation applies to both the symmetric and non-symmetric designs. The vertical shaft with actuator on top is the preferred orientation. In applications where the valve shaft is horizontal, the hydrostatic torque component results from the variation in the static head of the process fluid from the top to the bottom of the valve disc due to gravity, as shown in Figure 3-3. Depending upon the direction of the hydrostatic torque and the direction of shaft rotation to open or close the valve, this torque component can assist or oppose the actuator in the seating direction. In general, the hydrostatic torque can be neglected except for very large valves—30 inches (DN 750) and larger.

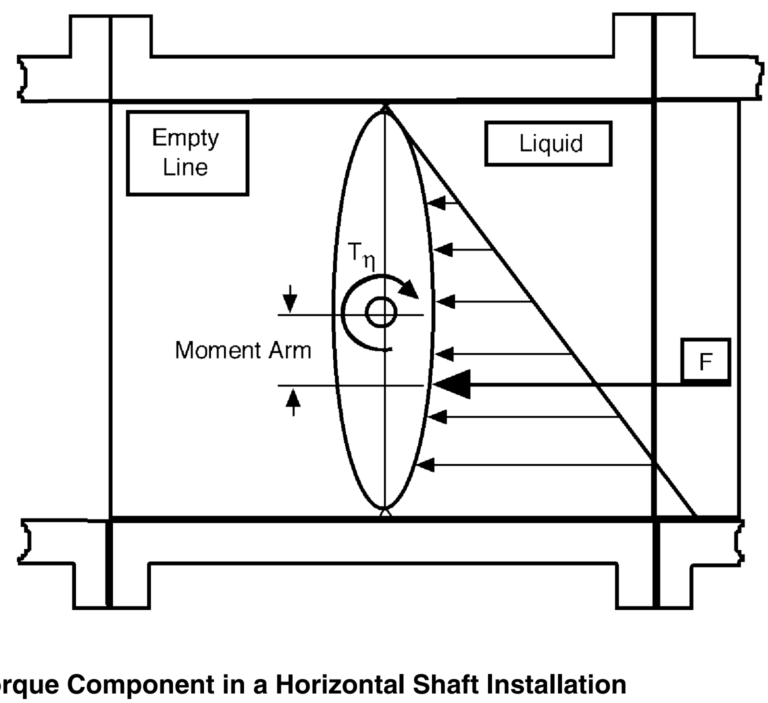


Figure 3-3
Hydrostatic Torque Component in a Horizontal Shaft Installation

The hydrostatic torque becomes zero (or negligibly small) under any one of the following conditions:

- The valve shaft is vertical. This orientation results in a zero moment arm for symmetric and single offset discs and a negligible moment arm for a majority of the double offset disc designs.
- The liquid levels in both the upstream and downstream pipes are the same (either full, empty, or partially full).
- The process fluid is air, gas, or steam.

In some large valves, the magnitude of the hydrostatic torque component can be high enough to overcome the total seating/unseating torque. In the absence of valve operator resistance, the valve can open by itself under hydrostatic torque.

For double and triple offset disc designs (where the shaft is offset from the pipe centerline), the pressure drop across the valve disc gives another hydrostatic torque component, which is referred to as ΔP-induced hydrostatic torque. This hydrostatic torque component can be very significant, especially for large valves under high-pressure drop [1].

3.3.3.10 Pressure Locking

Both symmetric and non-symmetric disc butterfly valves can experience very high unseating torque requirements if an incompressible fluid is trapped between two tight seal valves. An increase in the pressure of the trapped liquid or water (such as by heating) can lead to a pressure-locking scenario. In addition to the increase in the bearing torque, double and triple offset disc valves will have a ΔP-induced hydrostatic torque component.

3.3.4 Packing Leaks

Packing leaks are generally the result of improper stem finish or improper packing. Generic repacking instructions covering these problems and other considerations are contained in Appendix D. For a detailed discussion, see EPRI's *Valve Packing Performance Improvement* [14].

Packing leaks can also be attributed to flow-induced vibration. Mitigating the effects of vibration is discussed in Section 3.4. To deal with this type of leakage, one plant reported using a type of rubber-cored packing for these valves. The rubber-cored packing compensates for some of the oscillations encountered. The plant reported that this type of packing has been at least as effective as conventional graphite/PTFE packing.

3.3.5 Flange Leaks

Flange leaks are generally the result of flange face finish, insufficient bolt torque, or incorrect bolt torquing sequencing. For a detailed discussion of flanged-joint connections, see *Assembling Bolted Connections Using Sheet Gaskets* [3] or *Assembling Bolted Connections Using Spiral-Wound Gaskets* [15].

3.4 Vibration and/or Cavitation from High Velocity Flow or Throttling

Vibration and/or cavitation are the result of either high velocity service, throttling, or both. The relationship is complex, but each condition can make the other worse. Cavitation is a direct result of excessive throttling and can be the indirect result of high-velocity flow.

3.4.1 Vibration from High-Velocity Service

When a flow rate gets high enough, a structure in the flow path will vibrate noticeably. The vibration is the result of several factors, including the effect of flow turbulence and the interaction of the flow forces with structural items. For the butterfly valve, the structure affected most is the disc. This disc vibration is transmitted through the shaft. As the flow increases, so does the vibration. The vibration finally reaches a level that results in high bearing wear and packing deterioration. The amount of flow required to reach this level is dependent primarily on the disc shape and the orientation of the disc to the flow. Installations with velocities over 16 ft/sec (5 m/sec) have reported damage.

The most common method of reducing vibration is to turn the disc slightly into the flow stream (throttle), thus placing a preload on one side of the disc, which in turn resists the flow forces causing the vibration. However, if this problem involves a system that is operating close to its design limits, this strategy may not work. For example, if the circulating water system flow margin to maintain condenser vacuum at required levels is near zero (as may happen in summer), the small flow loss incurred by this throttling may not be acceptable. One plant reported that they had eliminated high velocity-induced vibration in their circulating water system by changing the valve to one that has a domed disc. An example is provided in Figure 3-4. A small amount of preloading was still required.

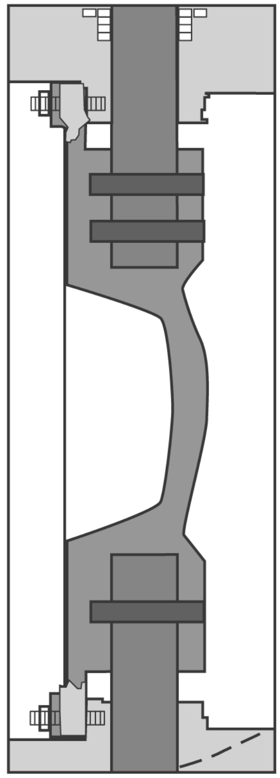


Figure 3-4
Dome Type Disc

3.4.2 Vibration and Cavitation from Throttling

Throttling produces high velocities by reducing the flow area. These high velocities act in the same manner as described above. In addition, the results of excessive throttling of any valve can result in damage to the valve internal surfaces or to the downstream piping. The reason for this damage is cavitation.

Cavitation is best understood by considering what happens to a fluid such as water when it goes through a reduced area of flow such as a valve. As the water goes through the valve, the velocity must increase in order to maintain the mass flow rate (that is, pounds of water per second). When the velocity goes up, however, the pressure at that point must go down in order to conserve the energy of flow. If the valve flow area is small enough, the pressure will drop low enough for some of the water to start to vaporize (form very small steam bubbles) because of the thermodynamic properties of water (vapor pressure). After passing through the small flow area, the velocity slows down, and the process is reversed (that is, the pressure starts to rise and the vapor is returned to its liquid state). The return to the liquid state is usually a violent process. The bubbles are collapsing, and the amount of force involved is very high (as high as 1,000,000 psi/68,948 MPa). If the bubble collapses next to a hard object, such as the valve or the downstream piping, the collapse can cause severe and often rapid damage to the surface. Of course, because the bubble is very small, the individual damage is very small. However, collectively over time, continued bubble collapsing can cause overall severe surface damage regardless of the material.

From an operating standpoint, cavitation can be detected only as a crackling sound similar to frying bacon. The crackling is the vibration caused by the collapsing bubbles. Because the number of bubble collapses that can be heard varies with flow and flow geometry, the amount of sound cannot be used as a measure of cavitation severity. In addition, it can be masked by the flow noise and not be heard at all when it occurs.

The damage from butterfly valve cavitation can occur either in the valve (typically the backside of the disc) or in the downstream piping, although it is more likely to be in the piping. This is because the valve is very short and pressure recovery is more likely to take place in the piping.

Severe throttling can also have an erosive effect, particularly where the valve is just off the seat. The high velocity flow of the water impinging on the body will wear away the surface. This is particularly true with raw water that contains sediment.

Therefore, throttling flow with a standard butterfly valve must be done with care. Normally, when it is necessary to overcome some operational concern, throttling can be tolerated without limit down to 30% open [1]. Continuous throttling at flows lower than that should include valve walkdowns to listen for the characteristic sound both in the valve and in the downstream piping. If the sound is heard, the valve and piping should be scheduled for visual inspection at the next outage. Any time a butterfly valve is used for throttling, the valve and piping should be scheduled for visual inspection at the next valve rebuild. One plant reported cavitation damage to downstream carbon steel piping because of the system startup transient that throttled the butterfly valve for about one minute to fill the system and prevent water hammer. The plant expressed surprise as to the damage that could occur over the years, one minute at a time. One way to design away cavitation concerns is to install a valve that is one or two sizes smaller than the nominal pipe diameter by using reducing fittings. This will help keep the valve >30% open.

4 CONDITION MONITORING AND PREVENTIVE MAINTENANCE

4.1 Introduction

Butterfly valve condition monitoring (CdM) and preventive maintenance (PM) practices are not extensive. Failures reviewed in Section 3 that can be addressed by CdM/PM can be attributed to several causes:

- Corrosion
- Aging
- Throttling and cavitation erosion

Corrosion is normally attributed to incorrect material selection. Material selection is part of the specification process. CdM/PM for corrosion prevention is therefore an inspection program from which results are used to correct material deficiencies. A good discussion on corrosion in open cycle systems is found in Section 4 of *Corrosion Evaluation of Service Water Materials* [16].

Aging, on the other hand, is limited to elastomers and packing. CdM/PM is then associated with replacements at end of life. Inspections are used to detect elastomer and packing end-of-life and, using history and manufacturer's recommendations, to establish an appropriate PM.

Throttling and cavitation erosion are a result of incorrect valve operation. While it is not always clear that the valve is overly throttled and/or if cavitation is present with the valve in service, CdM/PM inspections will detect these problems.

4.2 Condition Monitoring

4.2.1 Inspection

4.2.1.1 Types of Inspections

The inspections are divided into three groups:

- **Walkdown/External**: To be used during system operation in order to detect problems in a timely manner to permit scheduling of repairs as appropriate.
- **Internal**: To be used to assess the condition of the valve during an outage without disassembly. Routine PM activities such as packing replacement and off-center design seat replacement are also carried out. Large valves can be left in place or removed.
- **Teardown**: To be used when a complete overhaul of the valve is carried out due to problems or PM. Large butterfly valve refurbishments require extensive on-site machining capabilities. Consequently, such jobs may have to be done at an off-site repair facility.

Attributes of the inspections are provided in the following sections.

When inspections require removal and reinstallation of the valve, problems can occur if the valves are improperly handled. Provisions should be made for permanent lift points or, absent that, a means of access to the valve internals.

A walkdown inspection of all butterfly valves is highly recommended immediately after startup to detect changes from the previous cycle. These changes should be monitored until their effects are understood. Inspections during operation are also necessary in order to monitor condition.

4.2.1.2 Use of Checklist

The areas of the inspection along with attributes are provided in section 4.2.1.3. An inspection checklist that repeats the attributes but not the type of failure (for example, corrosion or looseness) is provided in Appendix E. Not all attributes will necessarily be applicable to a valve design. Therefore, the left column of the checklist allows the user to designate the attribute as applicable or not.

4.2.1.3 Attributes of Checklist with Type of Failures

Note: An indication of (R) following the type of failure in the checklist (in Section 4.2.3 and Appendix E) indicates a part subject to replacement under a PM program as discussed in Section 4.3.

4.2.2 Walkdown/External Inspection

The areas to be inspected in walkdown/external inspection and the possible types of failures include:

- Valve stroke: Inspections should include cycling of the valve, usually before the inspection takes place. This will allow observation of various parts under motion and permit detection of anomalies that can later lead to malfunctions.
- Actuator: Corrosion, freedom of movement, hand wheel looseness
- Body general: Corrosion, through-wall leakage
- Bonnet/top cover plate: Leakage, corrosion
- Bottom cover: Leakage, corrosion
- External bolting/threads: Corrosion, looseness
- Flange/weld downstream: Corrosion, through-wall leakage
- Flange/weld upstream: Corrosion, through-wall leakage
- Live load springs (if installed): Proper height, corrosion
- Packing bolts: Corrosion, looseness
- Packing gland/follower: Corrosion, leakage, gland bottomed out (cannot compress packing further), flange not level, looseness
- Packing: Leakage
- Trunnion lower: Corrosion
- Trunnion upper: Corrosion

4.2.3 Internal Inspection

The areas to be inspected in internal inspection and the possible types of failures include:

- Valve stroke: The same as described in Section 4.2.2
- Body general: Internal body corrosion, underliner corrosion
- Body liner (symmetric): Cracking, hardening, missing pieces, ripping, separation from body, nicks, cuts, other signs of deterioration
- Seating surface (non-symmetric): Cracking, hardening, missing pieces, ripping, nicks, cuts, other signs of deterioration
- Body seat mating surface (rubber seat on disc): Corrosion

- Disc general condition: Corrosion
- Disc position stop: Bent, fracture, missing
- Disc seating edge surface: Corrosion, replaceable seat deterioration (if equipped)
- Disc/lower shaft connection (pins): Looseness, corrosion (see Section 4.6) (R)
- Disc/upper shaft connection (pins): Looseness, corrosion (see Section 4.6) (R)
- Gasket face: Corrosion, gouges
- Seat retainer bolt/screws: Corrosion, loose, missing
- Seat retainer: Corrosion
- Shaft lower general condition: Corrosion
- Shaft upper general condition: Corrosion
- Shaft/actuator connection: Loose fit

4.2.4 Teardown

The areas to be inspected in teardown and the possible types of failures include:

- Valve stroke (before removal): The same as described in Section 4.2.2
- Bearing lower: Corrosion, deterioration
- Bearing upper: Corrosion, deterioration
- Body general: Internal body corrosion, underliner corrosion
- Body liner (symmetric): Cracking, hardening, missing pieces, ripping, separation from body, nicks, cuts, other signs of deterioration
- Seating surface (non-symmetric): Cracking, hardening, missing pieces, ripping, nicks, cuts, other signs of deterioration
- Body seat mating surface (rubber seat on disc): Corrosion
- Disc general condition: Corrosion
- Disc position stop: Bent, fracture, missing
- Disc seating edge surface: Corrosion, replaceable seat deterioration (if equipped)
- Disc/lower shaft connection (pins): Looseness when removing, corrosion, cracks (see Section 4.6)
- Disc/upper shaft connection (pins): Looseness before removing, corrosion, cracks (see Section 4.6)

• Gasket face:	Corrosion, gouges
• Packing box:	Corrosion, free of old packing (See Appendix D)
• Seat retainer:	Corrosion
• Shaft actuator end:	Corrosion, cracking
• Shaft lower bearing area:	Corrosion, galling
• Shaft lower total indicated runout (TIR):	See Section 4.7
• Shaft packing area:	Corrosion (pitting, general)
• Shaft upper bearing area:	Corrosion, galling
• Shaft upper TIR:	See Section 4.7
• Shaft/actuator connection:	Connection, loose fit

4.3 Preventive Maintenance

4.3.1 Purpose

The purpose of preventive maintenance (PM) is to arrest a degradation process and restore full functional capability. Preventive maintenance can take one of three forms:

- Periodic restoration
- Periodic replacement
- On-condition restoration (predictive maintenance)

Periodic restoration includes, for example, cleaning, changing lube, or overhaul (tear-down, rebuild). Periodic replacement includes replacement of piece parts to complete replacement of the component. On-condition restoration entails monitoring one or more parameters indicative of the progress of some failure mode considered the cause of performance degradation.

4.3.2 Recommended Tasks

Based on the analysis of failures described in Section 3, the following preventive maintenance tasks are recommended. Not all of these tasks are applicable to all valves (for example, lubrication or retorquing stem/disc connections).

- Condition monitoring (as described previously)
- Periodic seat replacement (non-symmetric design)
- Periodic repacking
- Replace seal O-rings

- Lubrication
- Retorque stem/disc connections
- Complete teardown and rebuild
- Vendor recommendations

4.3.3 Determining Applicable Valves, Tasks, and Intervals

Setting up a PM program requires the establishment of an expert panel including experienced individuals from maintenance, engineering, and operations. They should be personnel who have worked on or operated the valves. The expert should perform the following activities:

- Determine which valves are critical to plant operation (that is, failure would result in reduced power or limiting condition for operation).
- Review the critical valves and determine the part failures that have or could cause component failures and what type of PM (periodic restoration, periodic replacement, or on-condition restoration) is appropriate.
- Determine the appropriate interval for the PM.
- Review the non-critical valve list to determine if PMs are needed for them for economic reasons and to determine the appropriate interval.

The details for setting up a PM program are described in EPRI's *Preventive Maintenance Basis* [17]. A chart providing insight into the steps of the process is reproduced in Appendix F.

An efficient method of establishing a reliability-centered PM program without the large upfront costs of conducting a failure modes and effects analysis (FMEA) is contained in *Comprehensive Low-Cost Reliability Center Maintenance* [18].

The following section contains guidance on carrying out some of the PM tasks.

4.4 Lubrication

Butterfly valves are designed for extended service with minimal wear and servicing. Generally, no regular lubrication is required. Shaft bearing surfaces are normally factory lubricated. If the valves are equipped with O-ring seats or shaft seals, they usually have permanently locked-in lubricant to prevent flow medium from penetrating major bearing surfaces. Where lubrication is used, care should be taken not to allow it to come in contact with elastomers in the valve unless specifically directed by the manufacturer's instructions.

4.5 Seat Replacement

The seats will harden and begin to leak over time. Leakage will begin and increase. The seats will tend to tear and chunk, possibly leaving a large gap in the seat. Therefore, leaking seats should be replaced at the next plant shutdown if possible. If the valve is not critical, temporary patching can be used, but only as a last resort and then only until the next outage. Depending on the valve service and experience, elastomeric seats should be replaced every 4 to 10 years. However, any time that a valve requires corrective or preventive maintenance permitting free access to the seat, replacement of the seat should receive strong consideration.

Seat design dictates the ease of replacement. The most difficult are those seats that are held in the body with an adhesive or are otherwise molded to the body. While some can be replaced in the field or in the shop, it is usually very difficult. Most plants find it more effective to send them to the manufacturer or authorized repair facility. The inflatable seals are easier to replace and should not require shipping offsite. Finally, the high performance valves with the offset designs usually involve removing a seat retainer, secured by screws, that allows easy access to the seat.

Some plants have replacements ready to go and can just swap out the valves. The valve is then rebuilt during the operating cycle instead of during outage critical path time. This approach requires a cost/benefit analysis based on budget considerations and time constraints. Larger butterfly valves can cost multiple hundreds of thousands of dollars. However, if critical path is involved, such investments can have large paybacks.

It is necessary to verify that the retaining ring fasteners are present when removing a valve. Some plants have found that although the vendor manual shows fasteners for the seat retainer ring, they are missing. They are held in place between the flange faces of the valve and pipe. This can be a safety concern. Because the ring can stick initially, this situation will not be immediately obvious, and the ring can come loose later, dropping or falling on personnel in the vicinity.

4.5.1 Other Elastomer Parts

Any time that equipment is refurbished, soft parts must be replaced in accordance with good maintenance practices. It is recommended that when seats are replaced, other elastomer parts be replaced even if disassembly of the other subassemblies is not required.

4.6 Disk-to-Shaft Connections

During CdM/PM inspections, pins should be examined for the following conditions. The basis for these inspections is found in Section 3.

- Cracking or other failures that indicate strength is inadequate.
- Corrosion (either general or pitting) that indicates the material is inadequate for the environment.
- Close fit between shaft, pin, and disc to prevent fatigue failure from vibration.
- The pin has not come loose.

Some plants have found that ultrasonic examination (UT) can be used effectively to detect cracking.

Pins that are not degraded should normally be removed easily. When encountering difficulty in removing degraded pins, some plants have found that a high-pressure water jet cutting process (using an abrasive material such as boron carbide) is very effective in pin removal, reducing overall maintenance time from three days to one-half day.

Key Technical Point

Some plants have found that ultrasonic examination (UT) can be used effectively to detect cracking.

Key O&M Cost Point

When encountering difficulty in removing degraded pins, some plants have found that a high-pressure water jet cutting process (using an abrasive material such as boron carbide) is very effective in pin removal, reducing overall maintenance time from three days to one-half day.

4.7 Total Indicated Runout (Bent Shaft)

Total indicated runout (TIR) is a recommended inspection only when opening/closing torque is excessive and examination of the bearing (and the shaft in contact with the bearing) does not appear to be the problem. TIR is a measure of bending. It is the lateral difference between the actual axis of a shaft and the desired or straight axis of the part. Usually, it is considered to be the maximum difference.

The best way to carry out TIR measurement is to hold and rotate the shaft between the end centers in a lathe (or between a chuck and a center) using a dial indicator to measure the lateral displacement. While this will also include the effect of out-of-roundness, it will usually be very small compared to shaft axis displacement.

4.8 Repacking

Properly installed packing on a butterfly valve used in a power plant should perform well for long periods. Low pressures, temperatures, and a stem that rotates rather than moving axially should not challenge available materials and configurations. Guidance on repacking valves is contained in Appendix D.

4.9 Other Problems and Solutions

The following have been suggested by one plant as solutions to commonly encountered problems:

- For deterioration and tearing of liners in lined-body butterfly valves, replace lined-body butterfly valves with single or double offset butterfly valves. Use metal-seated valves where possible.
- Consider the use of the triple offset valve where seat leakage in containment isolation valves continues to be a problem.
- If metal-seated butterfly valves with pressure-energized seats leak at low differential pressures, replace metal seats with elastomeric seats, or use metal-seated valves that do not depend on differential pressure for tight shutoff.
- To deal with large butterfly valves that have rotated too far in the closing direction (the disc passes through the seat and cannot be rotated back to the closed position), remove the gearbox, and move the valve disc to the open position by continued rotation in the closing direction. Position the gearbox to the open position and reinstall.
- If water hammer occurs due to rapid closing of butterfly valves, install a gearbox.
- Repairing lined-body butterfly valves requires stem removal, which is often difficult or impossible. Replace lined-body butterfly valves rather than attempting to repair them. If a field repairable valve is preferred, replace with an offset design butterfly valve.
- If graphite buildup on the stem causes failure of stem seals, use braided graphite packing rather than die-formed rings.

5
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6
LISTING OF KEY INFORMATION

The following list provides the location of "Key Point" information in this report.

Key O&M Cost Point

Emphasizes information that will result in reduced purchase, operating, or maintenance costs.

Referenced Section	Page Number	Key O&M Cost Point
3.3.1.1.1	3-4	Determining the seat position also applies if the seat has just been replaced. The sweet spot is specific to the installed ring. It must be verified when seat replacement has taken place.
3.3.1.1.2	3-5	Elastomers relax with time or creep. This results in lowering the seating forces and can result in leakage later. Most creep takes place within 24 hours. Therefore, retorquing or readjusting a seat 24 hours later can result in better seat performance.
3.3.2.3	3-10	Most shaft binding failures were due to a packing gland made from a material that corroded readily in a water environment. The corrosion product (rust) filled the gap between the shaft and the gland, freezing the parts together. When the shaft was rotated, the operating torque was resisted by the follower studs that were connected to the gland follower.
3.3.3.5	3-14	A small additional pressure drop can cause tremendous additional bearing loads for large valves. A 10 psid (69 kPa) drop across a 72-in. (DN 1800) valve can result in a torque requirement of 5125 ft-lbs (6920 N-m).
4.6	4-8	When encountering difficulty in removing degraded pins, some plants have found that a high-pressure water jet cutting process (using an abrasive material such as boron carbide) is very effective in pin removal, reducing overall maintenance time from three days to one-half day.

Key Technical Point

Targets information that will lead to improved equipment reliability.

Referenced Section	Page Number	Key Technical Point
3.3.1.1.1	3-3	A very effective method for testing a valve for leakage is to remove the valve, install a blind flange on one side of the valve, and pressurize the valve seat through a tap in the flange. This is a relatively easy process and allows adjustments to the seat position prior to reinstalling the valve.
3.3.1.1.1	3-3	Once a sweet spot has been determined, it is important to effectively lock the valve in the correct closed position. The use of a gag is very effective. If this is not possible, match marks should be used as a minimum. The valve mechanical stops should be reset if available.
3.3.1.1.1	3-4	If the valve uses a motor operator, it is important to ensure that the mechanical stops and the limit switches are synchronized. Each time an adjustment is made, this synchronization must be maintained.
3.3.1.3	3-5	Temperature is the largest contributor to seat accelerated aging, followed closely by oxidation in HVAC systems.
3.3.2.1.3	3-9	Pin tightness is essential to prevent disc/shaft movement and resultant pin fatigue failure.
3.3.2.1.4	3-9	While welding is an excellent method of securing pins, it can be the reason for failures because of the effect of weld shrinkage if not done properly.
4.6	4-8	Some plants have found that ultrasonic examination (UT) can be used effectively to detect cracking.

A
GLOSSARY

Bearing	The cylindrical shaped surfaces for the shaft to transfer hydraulic loads to the trunnion (body). It can be an elastomer or metal.
Body	That part of the valve that contains the seat and is connected to the piping.
Disc	That part of the valve that is connected to the shaft and rotates to control flow.
Disc position stop	A protrusion (or protrusions) in the body that prevents the disc from traveling more than 90 degrees from the open position (disc parallel to flow).
Disc seating edge	That part of the disc that engages the seat.
Elastomer	A general name for non-metallic materials used for liners and soft seating surfaces.
Gland	The cylindrical-shaped device that applies load directly to the packing. The gland force is provided by a gland follower (sometimes known as a gland flange). The gland and follower are frequently manufactured as one piece.
Liner	An elastomer that covers the inside circumference of the body and which may or may not extend completely from end to end.
Non-symmetric	A butterfly valve design in which the shaft connects to the side of the disc (see Figure 2-4).
Packing box	The annular chamber provided around the shaft in a sealing system into which deformable packing is compressed using a gland.
Rubber	A type of elastomer (for example, butyl rubber, EPDM, or neoprene)
Seat retainer	A flat metal ring that is used on a non-symmetric valve to hold a replaceable elastomeric seat in place.

Seat retainer bolt/ screws See seat retainer.

Seating surface The liner in a symmetric valve or the elastomeric seat in a non-symmetric valve.

Shaft A round bar that passes through the upper and lower trunnions and is connected to the disc. The shaft can be one piece or two pieces. A two-piece shaft has an upper (and lower) shaft passing through the upper (and lower) trunnion and connected to the upper (and lower) part of the disc.

Symmetric A butterfly design in which the shaft passes through the center of the disc (see Figure 2-3).

Total indicated runout (TIR) A measure of bending that is the lateral difference between the actual axis of a shaft and the desired or straight axis of the part. It is usually considered the maximum difference (see Section 4-7).

Trunnion The upper and lower parts of the valve that house the bearings. It is usually considered part of the body.

B
ELASTOMER CHARACTERISTICS

The tables on the following pages provide characteristics of interest when reviewing the elastomers found in butterfly valves. They provide qualitative measures in order to assess material failures. Table B-1 is a general table of characteristics, Table B-2 is specific to heat resistance and oxidation resistance, Table B-3 is for radiation resistance, and Table B-4 covers temperature and compression set. Tables B-2 and B-3 are meant to be complementary (that is, they cover the same materials). These tables are for comparison purposes only. An engineering review is necessary before using them for design purposes. For safety-related applications, the equipment qualification report governs.

Table B-1
Elastomer Resistances for Various Environments

Common Name Base Polymer	Chemical Name	Swelling in Lubricating Oil	Oil and Gasoline	Water Absorption	Low Temp	Chlorinated Hydrocarbons
Butyl rubber	Isobutylene isoprene	P	P	VG	G	P
EPDM or EP rubber	Ethylene propylene	P	P	E	E	P
Neoprene	Chloroprene	G	G	G	G	P
Nitrile or NBR or Buna N	Acrylonitrile butadiene	VG	E	G	F-G	F

O = outstanding, E = excellent, VG = very good, G = good, F = fair, P = poor

Table B-2
Elastomer Resistances to Oxidation and Heat Aging

Common Name Base Polymer	Chemical Name	Oxidation Resistance	Oxidation Ranking	Heat Aging Resistance	Heat Aging Ranking	Combined Oxidation Heat Aging Ranking
Butyl Rubber	Isobutylene isoprene	E	2	VG	3	2.5
EPDM or EP Rubber (EPR)	Ethylene propylene	E	2	E	2	2
Neoprene	Chloroprene	VG	3	G	4	3.5
Nitrile or NBR or Buna N	Acrylonitrile butadiene	G	4	G	4	4

O = outstanding, E = excellent, VG = very good, G = good, F = fair, P = poor

Table B-3
Elastomer Resistances to Radiation [19]

Material	Exposure Resulting in 25% Loss in Tensile Strength (Rad)
Butyl rubber	5×10^5
EPR (EPDM, EPT)	6×10^8
Neoprene rubber	2×10^7
Nitrile rubber	6×10^8

Table B-4
Elastomer Temperature Limits and Set Resistance [3, 5]

Material	Temperature Limit °F (°C)	Compression Set Resistance
Butyl Rubber	200 (93)	Fair
EPR (EPDM, EPT)	250 (121)	Fair
Neoprene Rubber	200 (93)	Excellent
Nitrile Rubber	200 (121)	Very Good

C OPERATING EXPERIENCE

C.1 Introduction

As discussed and analyzed in Section 4, operational experience was reviewed to determine what issues caused reportable events involving butterfly valves. The sources of this information included Operations and Maintenance Reports, Operational Experience Reports, and EPIX failure data as well as U.S. NRC Notices and Licensee Event Reports.

C.2 Analysis

Nine hundred events were identified that involved butterfly valve operations and maintenance. These events were reviewed in detail to determine events that involved the butterfly valve proper or the manual gear operator. Problems with the actuator, motor or air, and design basis were excluded as discussed previously in this report[1, 2]. Of the final 896 events analyzed, 129 met the above criteria.

C.3 Analysis of Events

Table C-1 is the analysis of the 129 events. The events are listed by date of occurrence.

C.4 Summary of Solenoid-Operated Valve Problems and Causes

Figures C-1 through Figure C-4 summarize butterfly valve experience. For the purpose of this guide, the following definitions were used when classifying the data:

- **Failure Mode**: The valve function that was prevented by the failure
- **Failure Mechanism:** The physical, chemical, mechanical, or electrical process that prevented the valve from accomplishing its function
- **Failure Cause**: The reason or root cause for the failure, related to human errors, defects, service stresses, and/or wear-out

C.5 Discussion

A discussion of the analyzed data is contained in Section 3 of this report.

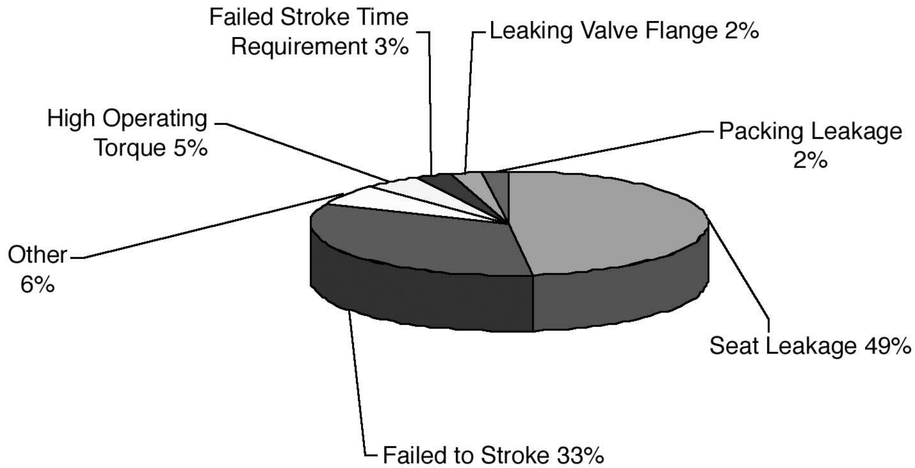

Figure C-1
Failure Modes

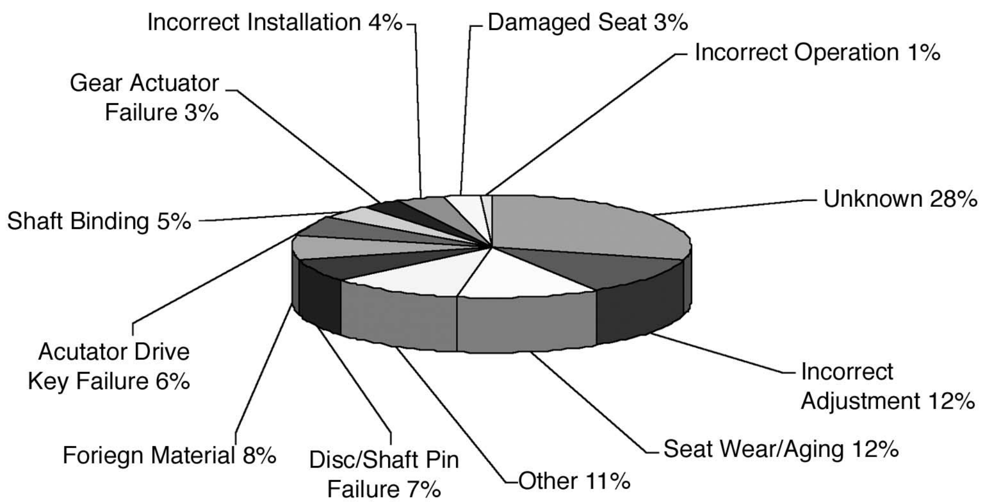

Figure C-2
Failure Mechanisms

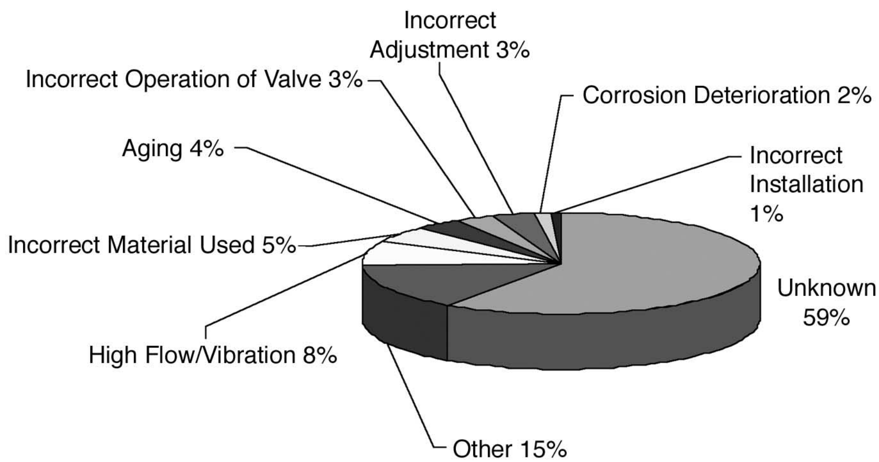

Figure C-3
Failure Causes

Table C-1
Analysis of Operating Events

Discovery Date	NSSS* Vendor	System	Containment Isolation	Failure Mode	Failure Mechanism	Failure Cause	Comments
2/16/1984	All	All		Other	Other	High flow/vibration	O&M R178
12/13/1989	General Electric (GE)	Service water (untreated)		High operating torque	Seat wear/aging	Aging	IN 90-21
6/13/1990	Combustion Engineering (CE)	Spray pond		Failed to stroke	Actuator drive key failure	Corrosion deterioration	Incorrect material used, OE3996
2/27/1991	Westinghouse (WEST)	Recirculation spray		High operating torque	Incorrect installation	Unknown	Shaft upstream, OE4483
12/23/1991	WEST	Service water (untreated)		Other	Other	Unknown	Valve body corroded by liner separation
4/30/1992	GE	Service water (untreated)		High operating torque	Incorrect installation	Unknown	Shaft upstream,. OE5272
5/29/1992	CE	Service water (untreated)		Seat leakage	Incorrect adjustment	Unknown	Improper valve/actuator alignment, OE5322
1/1/1993	WEST	Residual heat removal		Other	Other	High flow/Vibration	Weld, snubber failures
3/10/1993	WEST	All		Failed to stroke	Actuator drive key failure	Incorrect material used	OE5854
6/10/1994	GE	Closed cooling water		High operating torque	Other	Other	OE6709, used stainless steel bushings

Table C-1 (cont.)
Analysis of Operating Events

Discovery Date	NSSS* Vendor	System	Containment Isolation	Failure Mode	Failure Mechanism	Failure Cause	Comments
3/13/1995	ALL	All		Seat leakage	Other	Unknown	Liner failure
5/26/1995	W	Chilled water		Seat leakage	Incorrect adjustment	Unknown	Valve actuator travel-stop incorrectly adjusted
1/25/1996	WEST	Closed cooling water		High operating torque	Shaft binding	Corrosion product buildup	Liner failure, OE8016
9/24/1996	GE	Nitrogen purge		Seat leakage	Incorrect adjustment	Unknown	Valve actuator travel-stop incorrectly adjusted
3/1/1997	WEST	Feedwater pump condenser outlet		Failed to stroke	Disc/shaft pin failure	High flow/vibration	
3/3/1997	GE	Circulation water condenser outlet		Failed to stroke	Disc/shaft pin failure	High flow/vibration	
3/7/1997	WEST	Circulation water		Failed to stroke	Incorrect installation	Other	108-in. Pratt installed in vertical position
3/17/1997	WEST	Circulation water condenser Inlet		Failed to stroke	Gear actuator failure	Other	Excessive shaft clearances
3/27/1997	WEST	Unknown	Y	Seat leakage	Foreign material	Unknown	
4/12/1997	W	Service water (untreated)		Failed to stroke	Incorrect installation	Other	Shaft upsteam
5/17/1997	Babcock and Wilcox (B&W)	Containment ventilation	Y	Seat leakage	Seat wear/aging	Aging	

**Table C-1 (cont.)
Analysis of Operating Events**

Discovery Date	NSSS* Vendor	System	Containment Isolation	Failure Mode	Failure Mechanism	Failure Cause	Comments
5/20/1997	WEST	Service water (untreated salt water)		Failed to stroke	Disc/shaft pin failure	Unknown	
5/20/1997	WEST	Containment pressure relief		Packing leakage	Unknown	Unknown	
6/25/1997	GE	Service water (untreated)		Failed to stroke	Unknown	Unknown	
7/15/1997	WEST	Service water (untreated)		Failed to stroke	Other	Unknown	Bearing failure
9/12/1997	GE	Containment ventilation	Y	Seat leakage	Unknown	Unknown	
9/21/1997	GE	Closed cooling water		Seat leakage	Unknown	Unknown	
10/3/1997	WEST	Containment ventilation	Y	Seat leakage	Seat wear/aging	Aging	Uses T-Ring
10/10/1997	B&W	Containment ventilation	Y	Seat leakage	Incorrect adjustment	Unknown	Made unspecified seat adjustments and stopped leak
10/24/1997	WEST	Containment ventilation	Y	Seat leakage	Seat wear/aging	High flow/vibration	
10/31/1997	WEST	Containment ventilation	Y	Seat leakage	Seat wear/aging	High flow/vibration	Incorrect operation of valve, used excessive torque

Table C-1 (cont.)
Analysis of Operating Events

Discovery Date	NSSS* Vendor	System	Containment Isolation	Failure Mode	Failure Mechanism	Failure Cause	Comments
11/6/1997	WEST	Service water (untreated)		Seat leakage	Unknown	Unknown	
12/12/1997	GE	Service water (untreated)		Failed to stroke	Other	Unknown	Changed packing material to one with higher coefficient of friction
12/12/1997	WEST	Service water (untreated)		Seat leakage	Unknown	Unknown	
12/13/1997	WEST	Service water (Untreated)		Seat leakage	Unknown	Unknown	
12/27/1997	WEST	Service water (untreated)		Seat leakage	Unknown	Unknown	
1/1/1998	CE	Closed cooling water		Leaking valve flange	Unknown	Unknown	
1/6/1998	WEST	Containment ventilation	Y	Seat leakage	Unknown	Unknown	
1/12/1998	GE	Service water (untreated)		Failed to stroke	Disc/shaft pin failure	Other	Used excessive torque
1/22/1998	WEST	Service water (untreated)		Failed to stroke	Actuator drive key failure	Other	Used excessive torque
2/11/1998	WEST	Control room vent		Seat leakage	Unknown	Unknown	

Table C-1 (cont.)
Analysis of Operating Events

Discovery Date	NSSS* Vendor	System	Containment Isolation	Failure Mode	Failure Mechanism	Failure Cause	Comments
4/5/1998	CE	Containment ventilation	Y	Seat leakage	Unknown	Unknown	
4/22/1998	W	Unknown	Y	Seat leakage	Unknown	Unknown	
4/28/1998	CE	Service water (untreated)	Y	Failed to stroke	Foreign material	Unknown	
5/18/1998	CE	Condensate		Failed to stroke	Other	Incorrect operation of valve	Broken stop nut
5/23/1998	WEST	Reactor auxiliary building vent		Failed to stroke	Incorrect operation of valve	Unknown	
6/1/1998	GE	Fuel pool cooling demineralization		Seat leakage	Unknown	Unknown	
6/13/1998	GE	Service water (untreated)		Failed to stroke	Actuator drive key failure	Unknown	
6/24/1998	WEST	Containment pressure relief	Y	Seat leakage	Unknown	Unknown	
7/1/1998	GE	Service water (untreated)		Leaking valve flange	Unknown	Unknown	
7/22/1998	WEST	Freshwater cooling		Failed to stroke	Unknown	Unknown	

Table C-1 (cont.)
Analysis of Operating Events

Discovery Date	NSSS* Vendor	System	Containment Isolation	Failure Mode	Failure Mechanism	Failure Cause	Comments
9/3/1998	WEST	Fire protection		Failed to stroke	Actuator drive key failure	Unknown	
9/12/1998	GE	Unknown		Seat leakage	Seat wear/aging	Unknown	Uses T-Ring
10/21/1998	GE	Service water (untreated)	Y	Failed to stroke	Seat wear/aging	Unknown	
10/27/1998	WEST	Containment vacuum relief		Seat leakage	Unknown	Unknown	
11/24/1998	WEST	Containment ventilation	Y	Seat leakage	Unknown	Unknown	
12/26/1998	GE	Cooling tower de-ice		Failed to stroke	Unknown	Unknown	
1/11/1999	CE	Closed cooling water	Y	Seat leakage	Incorrect adjustment	Unknown	Improperly seated
2/2/1999	WEST	Containment ventilation	Y	Seat leakage	Foreign material	Unknown	
2/3/1999	WEST	Containment ventilation	Y	Seat leakage	Seat wear/aging	Aging	
2/4/1999	B&W	Service water (untreated)		Failed to stroke	Disc/shaft pin failure	Unknown	
2/9/1999	WEST	Containment ventilation	Y	Seat leakage	Seat wear/aging	High flow/vibration	

Table C-1 (cont.)
Analysis of Operating Events

Discovery Date	NSSS* Vendor	System	Containment Isolation	Failure Mode	Failure Mechanism	Failure Cause	Comments
2/11/1999	W	Closed cooling water	Y	Seat leakage	Incorrect adjustment	Unknown	Found two indentations on the seat surface (at 12 and 6 o'clock)
2/25/1999	WEST	Containment ventilation	Y	Seat leakage	Foreign material	Other	Containment purge valve, dirt
3/5/1999	GE	Unknown	Y	Seat leakage	Unknown	Unknown	
3/17/1999	GE	Containment ventilation	Y	Seat leakage	Unknown	Unknown	
3/17/1999	WEST	Hydrogen purge	Y	Seat leakage	Unknown	Unknown	
4/16/1999	GE	Containment ventilation	Y	Seat leakage	Incorrect adjustment	Unknown	Valve actuator travel-stop incorrectly adjusted
4/19/1999	WEST	Service water (untreated)	Y	Seat leakage	Damaged seat	High flow/vibration	
4/21/1999	GE	Containment air cooling	Y	Packing leakage	Unknown	Unknown	
6/10/1999	GE	Circulation water condenser inlet		Other	Other	Incorrect installation	Excessive packing and actuator wear, high flow/vibration
6/13/1999	GE	Service water (untreated)		Failed to stroke	Unknown	Unknown	

Table C-1 (cont.)
Analysis of Operating Events

Discovery Date	NSSS* Vendor	System	Containment Isolation	Failure Mode	Failure Mechanism	Failure Cause	Comments
6/23/1999	GE	TB closed cooling water		Leaking valve flange	Unknown	Unknown	
8/16/1999	WEST	Service water (untreated)		Failed to stroke	Unknown	Unknown	
9/22/1999	GE	Containment ventilation	Y	Seat leakage	Unknown	Unknown	
10/5/1999	GE	Service water (untreated)		Seat leakage	Unknown	Unknown	
10/11/1999	WEST	Service water (untreated)	Y	Seat leakage	Damaged seat	High flow/vibration	
10/16/1999	B&W	Containment ventilation	Y	Seat leakage	Seat wear/aging	Incorrect adjustment	
11/2/1999	CE	Containment ventilation	Y	Seat leakage	Incorrect adjustment	Unknown	OE11477
11/4/1999	WEST	Service water (untreated)		High operating torque	Shaft binding	Corrosion product buildup	OE13412, incorrect material used
11/6/1999	B&W	Service water (untreated)		Failed stroke time requirement	Shaft binding	Corrosion product buildup	Incorrect material used
12/9/1999	WEST	Containment ventilation	Y	Seat leakage	Unknown	Unknown	

Table C-1 (cont.)
Analysis of Operating Events

Discovery Date	NSSS* Vendor	System	Containment Isolation	Failure Mode	Failure Mechanism	Failure Cause	Comments
12/16/1999	GE	Service water (untreated)		Failed to stroke	Shaft binding	Corrosion product buildup	Incorrect material used
1/11/2000	GE	Torus purge	Y	Seat leakage	Other	Unknown	Inflatable T-ring, inflated too soon
2/23/2000	WEST	Service water (untreated)		Failed to stroke	Disc/shaft pin failure	Unknown	
4/19/2000	WEST	Unknown	Y	Seat leakage	Seat wear/aging	Aging	
4/26/2000	CE	Containment ventilation	Y	Seat leakage	Incorrect adjustment	Other	Valve seat retaining ring screws loose, not checked periodically
5/3/2000	GE	Containment ventilation	Y	Seat leakage	Foreign material	Other	Containment purge valve
6/26/2000	B&W	Service water (untreated)		Failed to stroke	Actuator drive key failure	Unknown	
7/4/2000	GE	Condensate demineralization		Failed to stroke	Disc/shaft pin failure	Incorrect operation of valve	
9/16/2000	GE	Containment ventilation	Y	Seat leakage	Foreign material	Other	
9/21/2000	GE	Service water (untreated)		Seat leakage	Other	Unknown	Disc broken in half,. OE11791

Table C-1 (cont.)
Analysis of Operating Events

Discovery Date	NSSS* Vendor	System	Containment Isolation	Failure Mode	Failure Mechanism	Failure Cause	Comments
9/29/2000	GE	Condensate		Seat leakage	Seat wear/aging	Unknown	
10/28/2000	GE	Containment ventilation	Y	Seat leakage	Damaged seat	Incorrect operation of valve	
11/10/2000	WEST	Closed cooling water		Seat leakage	Unknown	Unknown	
11/16/2000	CE	Containment pressure relief		Other	Unknown	Unknown	Would not control flow correctly.
4/10/2001	WEST	Service water (untreated)		Failed to stroke	Other	Other	Pressure locking
5/29/2001	WEST	Closed cooling water		Failed to stroke	Seat wear/aging	Unknown	
7/2/2001	WEST	Circulation water		Failed to stroke	Other	Other	Torque tube failure
7/13/2001	WEST	Unknown		Failed to stroke	Unknown	Unknown	
7/19/2001	CE	Circulation water condenser outlet		Failed to stroke	Damaged seat	Incorrect adjustment	
10/12/2001	CE	Service water (untreated)		Failed stroke time requirement	Unknown	Unknown	

Table C-1 (cont.)
Analysis of Operating Events

Discovery Date	NSSS* Vendor	System	Containment Isolation	Failure Mode	Failure Mechanism	Failure Cause	Comments
10/14/2001	C	Circulation water condenser outlet		Failed to stroke	Seat wear/aging	Incorrect adjustment	
11/18/2001	B&W	Service water (untreated)		Failed to stroke	Gear actuator failure	Unknown	
12/20/2001	WEST	Containment ventilation	Y	Seat leakage	Seat wear/aging	Other	Near liner shaft penetration
1/6/2002	CE	Service water (untreated)		Failed to stroke	Foreign material	Other	Silt collected due to valve service and location in system
2/5/2002	WEST	Containment ventilation	Y	Seat leakage	Incorrect adjustment	Other	Insufficient seat load, inadequate maintenance procedure
2/7/2002	B&W	Closed cooling water		Failed stroke time requirement	Foreign material	Unknown	
3/23/2002	WEST	Hydrogen purge	Y	Seat leakage	Incorrect adjustment	Unknown	Uses T-Ring, valve position indicator installed incorrectly
4/11/2002	B&W	Closed cooling water		Failed to stroke	Gear actuator failure	Incorrect material	
4/11/2002	WEST	Service water (untreated)		Packing leakage	Unknown	Unknown	

Table C-1 (cont.)
Analysis of Operating Events

Discovery Date	NSSS* Vendor	System	Containment Isolation	Failure Mode	Failure Mechanism	Failure Cause	Comments
4/28/2002	GE	Fuel pool cooling		Failed to stroke	Gear actuator failure	Unknown	
5/31/2002	WEST	Stator cooling water		Failed to stroke	Other	Incorrect operation of valve	Disc split down shaft, overtorquing OE14213
6/20/2002	WEST	Circulation water condenser outlet		Failed to stroke	Unknown	Unknown	
7/9/2002	WEST	Closed cooling water		Failed to stroke	Incorrect adjustment	Unknown	Disc stroked thru seat, actuator bracket shaft bushing left off
7/27/2002	WEST	Circulation water		Failed to stroke	Disc/shaft pin failure	Corrosion deterioration	
8/9/2002	WEST	Closed cooling water		Failed stroke time requirement	Shaft binding	Corrosion product buildup	Incorrect material used
10/2/2002	WEST	Turbine building closed cooling water		Other	Actuator drive key failure	Incorrect adjustment	Valve drifted open
11/11/2002	WEST	Service water (untreated)	Y	Seat leakage	Foreign material	Other	Excessive debris in service water, added screen
12/10/2002	WEST	Service water (untreated)		Other	Shaft binding	Corrosion product buildup	Follower stud failure when valve operated, ductile iron gland, OE15167

Table C-1 (cont.)
Analysis of Operating Events

Discovery Date	NSSS* Vendor	System	Containment Isolation	Failure Mode	Failure Mechanism	Failure Cause	Comments
1/2/2003	WEST	Containment ventilation	Y	Seat leakage	Incorrect adjustment	Unknown	Valve travel-stop incorrectly adjusted, uses T-Ring
1/7/2003	WEST	Circulation water		Failed to stroke	Disc/shaft pin failure	High flow/vibration	Fatigue failure OE15619
1/28/2003	GE	Containment ventilation	Y	Seat leakage	Incorrect installation	Unknown	
2/18/2003	WEST	Containment ventilation	Y	Seat leakage	Incorrect adjustment	Other	Did not retorque valve seat (EPDM) after installation
2/27/2003	WEST	Containment ventilation	Y	Seat leakage	Actuator drive key failure	Other	Incorrect design
3/6/2003	GE	Containment ventilation	Y	Seat leakage	Foreign material	Unknown	
4/8/2003	B&W	Containment ventilation	Y	Seat leakage	Unknown	Unknown	
4/17/2003	CE	Containment air Cooling	Y	Other	Unknown	Unknown	
5/13/2003	GE	Condensate storage tank outlet		Seat leakage	Incorrect adjustment	Unknown	Valve travel-stop incorrectly adjusted

*Nuclear steam supply system

D
GENERIC REPACKING INSTRUCTIONS

D.1 Introduction

Valve packing leakage is a large contributor to overall plant leakage. However, attention to a few key factors can assist in preventing or mitigating it.

D.2 Packing Preload

Before repacking, the packing preload must be determined. Packing preload is the amount of compressive stress in the packing that occurs because of the gland force. The preload is measured in pounds per square inch (psi) or megapascals (MPa) and must be high enough to prevent in-service leakage. However, since all packing undergoes consolidation in-service that results in a loss of height (and therefore preload), the amount of preload must also include margin for this consolidation. EPRI testing and research has shown that 1.75 times system design pressure is adequate. The following formula computes the required gland bolt torque:

$$T_{gb} = (M)\,(K)\,(P_g)\,[(D_{sb})^2 - (D_s)^2](D_b)$$

In this formula:

- D_{sb} = diameter of stuffing box, inches (mm)
- D_s = diameter of stem, inches (mm)
- D_b = diameter of gland bolt, inches (mm)
- M = 0.0327 for ft-lbs (0.000393 for N-m)
- K = nut factor (if unknown, 0.16 is suggested for properly lubricated bolts and nuts)
- T_{gb} = required gland bolt torque, ft-lbs (N-m)
- P_g = preload packing stress, psi (MPa) = 1.75 × system pressure, but not less than 2000 psi (13.8 MPa)

D.3 Packing Removal

The system must be shut down, depressurized, drained, and cooled before work begins. Packing a back-seated, pressurized valve is always a dangerous practice. All other safety requirements and plant procedures must be observed. If possible, plant personnel should backseat the valve to limit entry of foreign material and to aid in valve stem inspection.

The process outlined here should be followed:

Remove the gland nuts, clean, and set aside in a suitable container. Raise the gland assembly and, if necessary, tie it off clear of the stuffing box. Clean and inspect the packing gland studs and repair them as required, unless they will be replaced with elongated studs for live-loading. Inspect the exposed valve stem and packing gland. Clean any chemical buildup or deposits with very fine emery cloth. Remove the old packing from the valve, pulling evenly with two packing hooks 180 degrees apart. Use of only one packing hook tends to break the old packing into small pieces. Exercise caution not to score the stem or stuffing box with packing hooks during removal.

D.4 Stem Inspection

The valve stem should be smooth and free of any series of pits or scratches in a line that could possibly act as a leakage path. Valves previously fitted with asbestos packing require a more thorough inspection. The reinforcing wire used in asbestos packing can often score a valve stem.

Excessive stem roughness can ruin graphite packing. A stem finish of 32 µin Ra (0.81µm) or better and free of scratches, pits, or voids deeper than 0.002 inches (51 µin) [21] is recommended. Graphite cannot tolerate rougher stem finishes that may have been acceptable for asbestos packing. If the stem finish does not meet this criteria, the foreman should be notified. Reconditioning or replacement may be necessary. Other conditions warranting remedial action include excessive play in the stem bushing and bent stems. Total indicated runout (TIR) should be checked against the manufacturer's specifications where a bent stem is suspected.

D.5 Packing Gland and Stuffing Box Inspection

The process outlined here should be followed:

Ensure that the compression face of the packing gland is flat with no chamfers or radii that tend to encourage packing extrusion. Inspect the stuffing box with a flashlight and small inspection mirror. The stuffing box must be clean and free of any old packing material, foreign matter, or scale. Incomplete removal of old packing can cause uneven loading on the stuffing box spacer or replacement packing. This will result in poor radial stress distribution and premature packing consolidation that will reduce sealing effectiveness. A way to ensure that all packing has been removed is to check the depth of the stuffing box at several locations with a machinist's scale.

Swab out the stuffing box with a clean cotton cloth to remove any remaining foreign matter. Re-inspect to ensure cleanliness. The stuffing box wall should be smooth and free of deep pits and scratches that can act as a leakage path. A finish of 125 µin (3.2 µin) Ra free of scratches, pit, or voids deeper than 0.006 inches (152 µin) [21] is recommended. Document the as-found condition and any necessary remedial action on the data sheet contained within the procedure.

D.6 Spacer and Packing Installation

Measure the stuffing box depth, stem O.D. and stuffing box I.D. to verify measurements previously obtained from the manufacturer's prints. Ensure that proper size and quantities of spacers and packing rings are on hand before installation begins. EPRI recommends the use of only five rings (three square graphite die-formed and two braided) and a carbon spacer at the bottom of the stuffing box if the valve previously used more than five rings [14, 22] (An engineering review should be completed before applying this method to valves with lantern rings and other methods of packing controlled leakage.) Spacers and packing rings must conform to nuclear grade material purity specifications where required. Packing rings should be die-formed to a density of 90 lb/ft^3 (1442 kg/m^3). The use of a passive corrosion inhibitor is recommended for nuclear systems.

If the bottom of the stuffing box is beveled, check to see that the seating surface of the replacement spacer has a matching profile to ensure full seating and proper load distribution. If the spacer is carbon and has a flat bottom, the bottom of the box must have a flat profile or a cushion ring is required.

Lightly coat the valve stem with a nickel-based anti-seize compound to aid in spacer and packing installation. Install the spacer (if required) and seat in the stuffing box with a valve packing tamping tool (can be obtained from the packing vendor). Measure the new stuffing box depth in several locations to ensure that the spacer is fully and evenly seated.

Install one braided graphite ring on top of the spacer or at the bottom of the stuffing box if no spacer is required. Use a tamping tool to seat the braided ring, taking care to ensure that the butt cut ends are properly mated.

Install the die-formed sealing rings one at a time. Carefully open the rings with a twisting action to install them over the stem. Never open them with a hinge-like action. Stagger the cuts in the rings by 120°, pre-seating each ring with the tamping tool.

Install a top ring of braided graphite packing, taking care to ensure that the butt cut ends are properly mated. Seat packing with tamping tool. The top braided ring should be flush or just below the top of the stuffing box.

(**Note:** The following steps assume that live-loading is not being installed.)

Coat the threads of the gland bolts and faces of the gland nuts and flat washer with a nickel-based anti-seize compound. Place the gland in contact with the top packing ring and properly position the gland assembly over the studs while guiding the gland lip.

Install a flat washer and gland nut (finger tight) onto each stud. Ensure that the proper end of the nut is down. (ASTM A194-2H nuts have a flat on one end, and the 2H on the other end. The 2H must be visible after installation.)

D.7 Packing Compression and Consolidation

Alternating between gland nuts, tighten evenly to the torque value calculated previously and stroke the valve (travel should be a minimum of the height of the packing). Check to ensure that the gland flange is not cocked during tightening. Consolidate the packing by repeating the tightening and stroking process again at least five times and until no further gland nut rotation is observed. Consolidation is essential to effective packing performance.

If the valve develops a packing leak in service, the specified torque value can be exceeded by as much as 15% in an effort to stop the leak. Proceed in increments of 5%. If this adjustment fails to stop the leak, the valve packing should be removed and the components inspected again to determine the root cause of the leakage. These can include excessive clearances between the stem and stem bushing (or packing gland), a bent stem, rough stem finish, improperly sized die-formed rings, or incorrect installation.

If this consolidation method does not give satisfactory results, consolidate each ring as it is inserted (that is, compress to full gland bolt load and stroke the valve once). After all rings are inserted, consolidate the entire box as above. This method will require additional tooling (for example, split bushings).

E
INSPECTION CHECKLISTS

This appendix contains the checklists discussed in Section 4.

Table E-1
Walkdown/External Inspection

Valve Tag ______________________ Unit ______________________

Size ______________________ Work Order No. ______________________

Manufacturer ______________________ Date of Inspection ______________________

Applicability	Inspection Area (If Applicable)	As-Found Condition(s)											Action				Remarks
		Satisfactory	Worn	Galled/gouged	Bent	Corroded	Eroded	Pitted	Leaking	Loose	Debris/bio attack	Broken	Accept as is	Repaired	Replaced	Temporary seal	
	Valve Stroke																
	Actuator																
	Body general																
	Bonnet/top cover plate																
	Bottom cover																
	Downstream flange/weld																
	External bolting/threads																
	Live load springs (if installed)																

Table E-1 (cont.)
Walkdown/External Inspection

Applicability	Inspection Area (If Applicable)	As-Found Condition(s)											Action				Remarks
		Satisfactory	Worn	Galled/gouged	Bent	Corroded	Eroded	Pitted	Leaking	Loose	Debris/bio attack	Broken	Accept as is	Repaired	Replaced	Temporary seal	
	Lower trunnion																
	Packing bolts																
	Packing gland/follower																
	Packing																
	Upper trunnion																
	Upstream flange/weld																

Table E-2
Internal Inspection

Valve Tag ______________________ Unit ______________________

Size ______________________ Work Order No. ______________________

Manufacturer ______________________ Date of Inspection ______________________

Applicability	Inspection Area (If Applicable)	As-Found Condition(s)											Action				Remarks
		Satisfactory	Worn	Galled/gouged	Bent	Corroded	Eroded	Pitted	Leaking	Loose	Debris/bio attack	Broken	Accept as is	Repaired	Replaced	Temporary seal	
	Valve stroke (before opening)																
	Body general																
	Body liner (symmetric)																
	Seating surface (non-symmetric)																
	Body seat mating surface (rubber seat on disc)																
	Disc general condition																
	Disc position stop																

Table E-2 (cont.)
Internal Inspection

Applicability	Inspection Area (If Applicable)	As-Found Condition(s)											Action				Remarks
		Satisfactory	Worn	Galled/gouged	Bent	Corroded	Eroded	Pitted	Leaking	Loose	Debris/bio attack	Broken	Accept as is	Repaired	Replaced	Temporary seal	
	Disc seating edge surface																
	Disc/lower shaft connection																
	Disc/upper shaft connection																
	Gasket face																
	Seat retainer bolt/screws																
	Seat retainer																
	Shaft lower general condition																
	Shaft upper general condition																
	Shaft/actuator connection																

Table E-3
Teardown Inspection

Valve Tag ____________________ Unit ____________________

Size ____________________ Work Order No. ____________________

Manufacturer ____________________ Date of Inspection ____________________

Applicability	Inspection Area (If Applicable)	As-Found Condition(s)											Action				Remarks
		Satisfactory	Worn	Galled/Gouged	Bent	Corroded	Eroded	Pitted	Leaking	Loose	Debris/Bio Attack	Broken	Accept As Is	Repaired	Replaced	Temporary Seal	
	Valve stroke (before removal)																
	Bearing lower (R)																
	Bearing upper (R)																
	Body general																
	Body liner (symmetric)																
	Seating surface (non-symmetric) (R)																
	Body seat mating surface																
	Disc general condition																

Table E-3 (cont.)
Teardown Inspection

Applicability	Inspection Area (If Applicable)	As-Found Condition(s)											Action				Remarks
		Satisfactory	Worn	Galled/Gouged	Bent	Corroded	Eroded	Pitted	Leaking	Loose	Debris/Bio Attack	Broken	Accept As Is	Repaired	Replaced	Temporary Seal	
	Disc position stop																
	Disc seating edge surface																
	Disc/lower shaft connection (R)																
	Disc/upper shaft connection (R)																
	Gasket face																
	Packing Box																
	Seat retainer																
	Shaft actuator end																
	Shaft lower bearing area																
	Shaft lower TIR* (R)																

**Table E-3 (cont.)
Teardown Inspection**

Applicability	Inspection Area (If Applicable)	As-Found Condition(s)											Action				Remarks
		Satisfactory	Worn	Galled/Gouged	Bent	Corroded	Eroded	Pitted	Leaking	Loose	Debris/Bio Attack	Broken	Accept As Is	Repaired	Replaced	Temporary Seal	
	Shaft packing area																
	Shaft upper bearing area																
	Shaft upper TIR* (R)																
	Shaft/actuator connection																

(R) Indicates part subject to replacement under a PM program
* See Section 4.7 of EPRI report 1007908

F
FLOWCHART: PM DEVELOPMENT PROCESS

The flowchart presented in this appendix presents a preventive maintenance development process.

Flowchart: PM Development Process

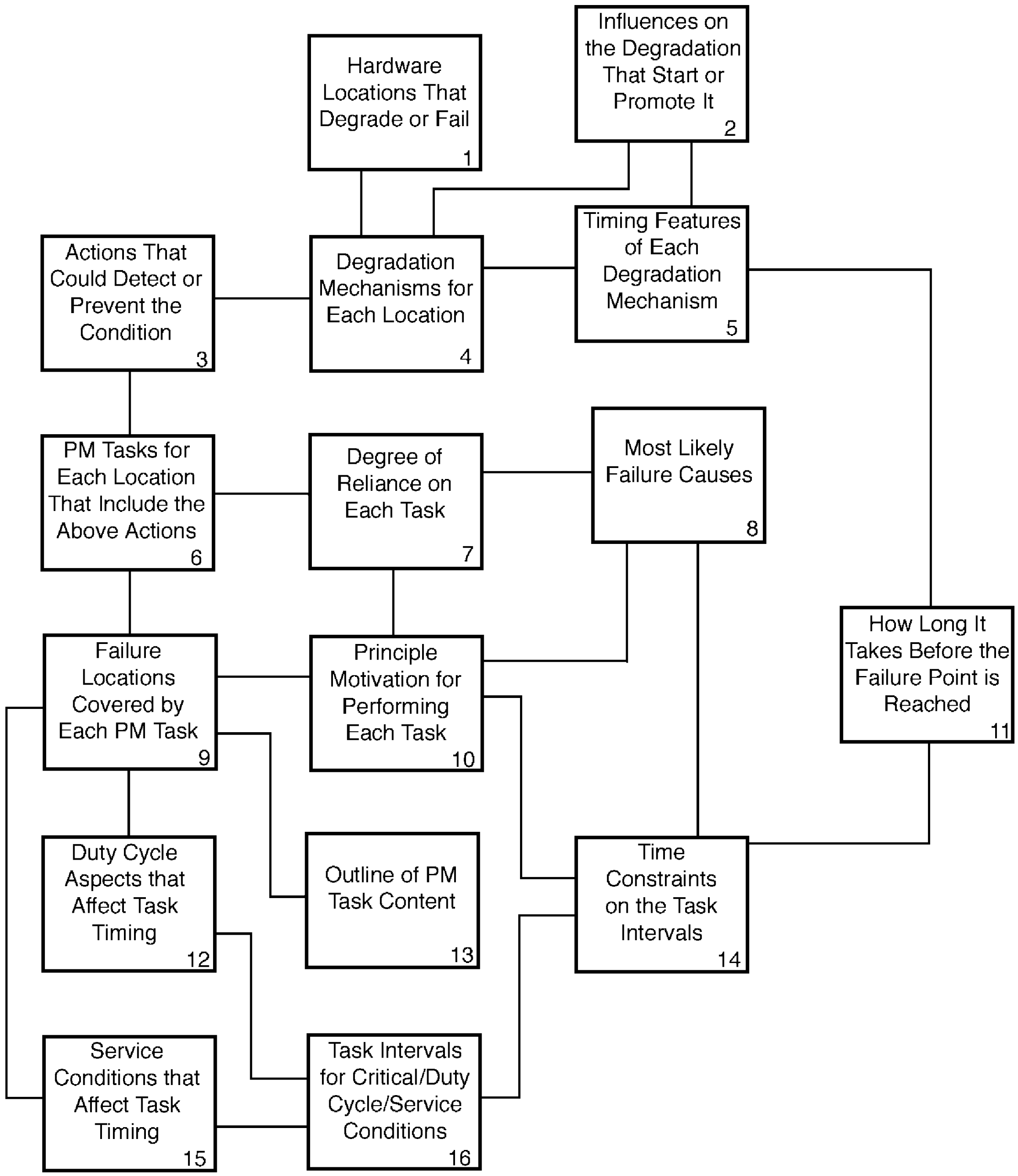

Figure F-1
Preventive Maintenance Development Process [17]

Note: Numbers shown in the steps in the flowchart refer to chapters in the source document.

Program:

Nuclear Power

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