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ENGINEERING PRACTICES FOR LARGE (4 INCHES AND OVER) GATE, GLOBE, AND CHECK VALVES IN WATER-COOLED NUCLEAR REACTOR SYSTEMS

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ENGINEERING PRACTICES FOR LARGE (4 INCHES AND OVER) GATE, GLOBE, AND CHECK VALVES IN WATER-COOLED NUCLEAR REACTOR SYSTEMS

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OAK RIDGE NATIONAL LABORATORY
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FOREWORD

This report was originally prepared by NUS Corporation under a subcontract with Union Carbide Corporation, Nuclear Division, as an activity of the RDT Standards Program at Oak Ridge National Laboratory. The information presented in this document was compiled by E. A. Saltarelli, J. L. Reneham, and S. B. Burwell of NUS Corporation under the direction of W. A. Bush of Oak Ridge National Laboratory as part of the preparatory work involved in the development of valve standards. This document is being issued as a reference to assist those specifying valve design and fabrication features for the United States Atomic Energy Commission Division of Reactor Development and Technology.

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ENGINEERING PRACTICES FOR LARGE (4 INCHES AND OVER) GATE, GLOBE, AND CHECK VALVES IN WATER-COOLED NUCLEAR REACTOR SYSTEMS

Abstract

Considerations relevant to the design, fabrication, and use of gate, globe, and check valves of 4-in. size and larger for water-cooled nuclear reactor systems are presented in this document. Long-term reliability of these valves is a major design objective reflected in the discussions of their applications with respect to functional requirements, flow characteristics, design criteria and analyses, material requirements, fabrication, maintenance, and special design features. Techniques which can be used to determine how well these guidelines for achieving high reliability have been followed are also outlined.

1. INTRODUCTION

Both general and detailed information on gate, globe, and check valves of 4-in. size and larger and considerations relevant to utilization of these valves in water-cooled reactor systems are presented in this document. These types of valves are described and illustrated, applications of these valves with respect to their functional characteristics are discussed, valve design criteria and analyses are defined, features of alternate valve designs are described, and the maintenance and in-service inspection considerations affecting valve service are outlined. Only the large sizes of these valves are considered in an attempt to limit the coverage of what would otherwise be an extremely voluminous subject. The intent is to provide the user of such valves, whether he is handling valve procurement or is a plant operator, with information on valve design and fabrication with accent on considerations introduced by nuclear application.

The design of a valve for nuclear use has as its objectives the achievement of a valve which will satisfy all of the safety, functional, and interface requirements imposed by the fluid system in which it is a component and the achievement of a valve which will provide maximum reliability over its service lifetime. The first objective can generally be assured by the specification of suitable requirements with their related tests and inspections. However, the increased long-term reliability necessary for nuclear valves imposes a new layer of requirements whose purpose is to insure improved quality in construction and performance over that of standard valves.

The emphasis on reliability derives in part from the necessity of insuring that nuclear plants do not constitute a threat to the safety of the general public, as reflected in AEC licensing requirements. Fluid systems necessary for the operation of the plant must function properly. In addition, the AEC operating license requires that standby engineered safeguard systems and their component valves be capable of operation at all times when the plant is operating. Thus, valve reliability is a major factor in the availability of a nuclear plant for power generation.

An additional incentive for relaibility is provided by the increased difficulty and consequent additional cost of performing maintenance on valves in radioactive fluid service. The radiological control procedures necessary in maintaining nuclear equipment are a significant cost item. The costs include such items as dosimeters, film badges, protective clothes, and the processing of these items. In addition, there are swipes, surveys, possible shielding, and special handling of tools and parts. In some cases, decontamination may be required before maintenance can be undertaken. Corrective maintenance is generally unanticipated and may require an unscheduled plant outage.

1.1 Reliability Design Considerations

While the major considerations involved in achieving reliability in fluid components are good design, comprehensive specifications, extensive inspection of materials and components, and adequate testing; the early design stage is the best and most efficient time to incorporate reliability into a valve. The reliability of the final product can be enhanced by a systematic review of the design with respect to all of the process system requirements. Because of the high reliability goals of nuclear application, other design factors such as weight, cost, and ease of fabrication must be subordinated to a degree. Twelve important design considerations that influence valve reliability are as follows.

1. Simplicity. The number of parts should be minimized, and complex mechanisms with moving parts and close clearance fits should be avoided.

2. Assembly and installation. Procedures and tools used for assembly and installation should be direct and simple, with minimum opportunity offered for error.

3. Maintenance and in-service inspection. Components requiring maintenance and in-service inspection should be accessible and designed for efficient periodic and corrective maintenance procedures.

4. Minimum contamination. Materials should be selected to minimize the release of elements contributing to radioactive activation, and traps where radioactive particles can accumulate should be avoided in the design.

5. Minimum radiation degradation. Materials selected for use in a valve should not be susceptible to radiation degradation in their intended application.

6. Minimum missile generation. The design should guard against component failure which could create missiles of stems, bonnets, etc., that might damage neighboring equipment critical to operation of the plant.

7. Minimum failure damage. Failure of the valve or its operative and control equipment should cause minimum damage to the process system.

8. Environmental extremes. The design should provide for the continued and reliable performance of the valve under all extremes of pressure, temperature, humidity, etc., of the environment, including post-accident conditions if required.

9. Liberal load and stress margins. Liberal design loads and safety margins should be used in the design of the valve. Allowances for manufacturing variations should be adequate and realistic.

10. Liberal performance margins. Design performance characteristics should be selected with a liberal margin from process system requirements.

11. Redundancy. Redundant or back-up components should be used where this technique improves reliability. However, it should be recognized that design complexity can offset the gain achieved by use of redundant design.

12. Proven components. Maximum use should be made of proved components and valve arrangements in the design.

While one should attempt to meet all the criteria, the valve requirements may necessitate the subordination of one or more of the preceding design considerations to satisfy a requirement of much greater importance for a particular application.

1.2 Costs of Nuclear Valves

The maximum reliability philosophy should apply from the basic material selection through fabrication, inspection, packaging, installation, and testing to a carefully planned and rigorously executed inspection and maintenance plan. As a result, the costs inherent in achieving the valve quality required for nuclear applications are significantly greater than those for ordinary production valves. The added cost of nuclear valves stems from many sources, among which may be

1. added features such as redundant stem seals, valve caps, and seal welding;
2. more comprehensive stress analyses;
3. corrosion-resistant materials;
4. more rigid leak-tightness requirements;
5. more rigid and extensive quality control such as radiography, dye penetrant inspection, testing, and cleaning;
6. added administrative requirements such as documentation, certifications, and reports; and
7. warranty provisions; that is, the cost of the valve will include the manufacturer's anticipated costs plus contingency in servicing the warranty.

Thus, the normal economic consideration of "functional effectiveness at lowest cost" is modified by the general high quality requirements for valves in nuclear applications. The ability of a manufacturer to achieve a certain consistent base level of quality must be insured. This places emphasis on finding suppliers fitted for the design and production of valves for the specialized functions of nuclear service; that is, suppliers whose manufacturing techniques and quality control methods consistently result in superior products. The practice of various companies, wherein they have lists of "eligible bidders" for supplying certain purchased items, is a reflection of this concept.

2. DESCRIPTION AND APPLICATION OF GATE, GLOBE, AND CHECK VALVES

As previously noted, only gate, globe, and check valves of 4-in. size and above are considered in this document. These three types of valves are described and their applications with respect to their functional characteristics are discussed in this chapter.

2.1 Gate Valves

The gate valve, illustrated in Fig. 2.1, is the most common type of valve used in industrial piping and is so designated because it controls flow by inserting or removing a gate-like disc across the flow path. The body of the valve forms the replaced section of piping and serves as the chief structural member. The bonnet is mounted on top of the body and supports the moving parts of the valve. The stem is used to operate the disc, and it penetrates the bonnet through a dynamic seal.

The gate valve is inherently suited for wide-open service, and as such, it is recommended for use only as a stop valve; that is, to fully shut off or fully turn on flow. Fluid may pass through the valve in either direction, and gate valves are often specified for process lines in which the direction of flow is frequently reversed. When open, fluid moves through the valve in a straight line with little flow restriction. Consequently, gate valves represent a minor flow resistance and cause only a small pressure loss in the flowing stream.

Seating is perpendicular or at right angles to the line of flow so that pressure forces acting over an area equal to that of the pipe bear directly against the gate valve. For example, a 6-in. gate valve holding fluid at a pressure of 300 psi puts a load of over 4 tons on one side of the disc. This is the full differential pressure if there is only atmospheric pressure on the downstream side. While the valve is seated, there is no wear or undue strain on the disc or seats. However, each time the valve is "cracked" open, there is a likelihood of wire drawing and erosion of seating surfaces by high-velocity flow. When the

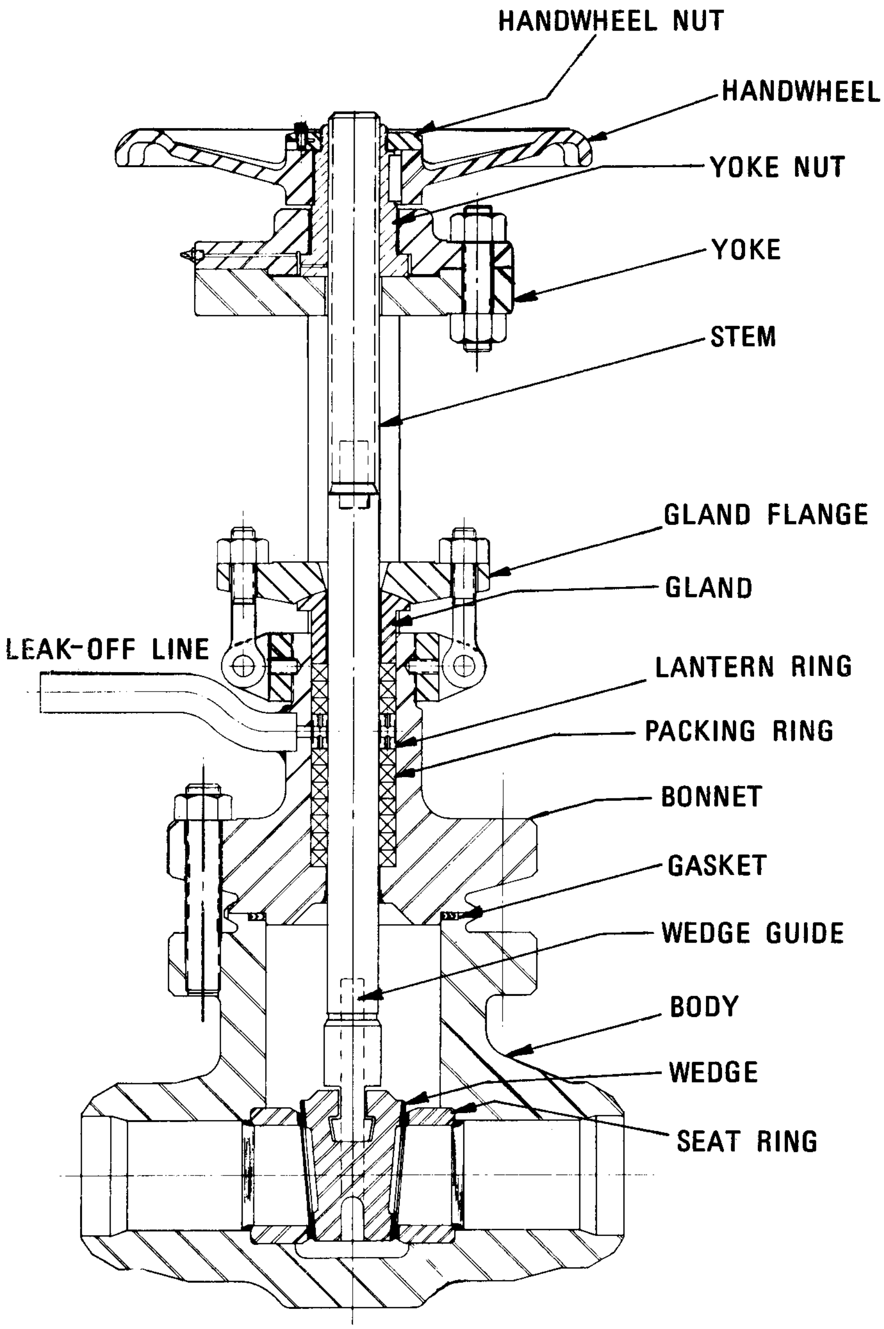


Fig. 2.1. Typical Gate Valve.
(Courtesy the Velan Engineering Companies)

disc is near the point of closure, the fluid pressure loads on the disc are high. Repeated movement of the disc near the closure point quickly results in galling and scoring of the downstream seating surfaces because of the greater load on these sliding surfaces. Gate valves generally use a wide seating surface of "even contact" seal to minimize the wear and scoring of these surfaces. The number of open-shut operating cycles is nevertheless limited, and gate valves are not recommended where frequent operation is necessary. A slightly opened disc may also cause turbulent flow with vibration and chattering of the disc.

For these reasons, gate valves are generally considered unsuitable for throttling service, and attempts to use them for this purpose shorten their life significantly. Because of the vertical disc orientation and the fact that a higher lift is required to raise the disc out of the flow stream into the body-bonnet area when the valve is opened, the gate valve requires more turns and more work to open it fully. This also means that the plans for installation of gate valves must provide for a space envelope adequate to accommodate stem travel if a rising-stem valve is used.

2.1.1 Disc Types

The gate valve has several variations in seating design that are primarily variations in the design of the disc with attendant seat modifications. The most common of these is the solid wedge-shaped disc with matching tapered body seating surfaces, as illustrated in Fig. 2.2. Downward force exerted on the disc by the stem jams the disc between the polished seats for closure. This is a strong simple design with a single part. It can be installed in any position with minimal probability of jamming caused by misalignment of parts. It is practical for turbulent flow because there is nothing inside to vibrate and chatter. Two of its disadvantages are that (1) refacing of the tapered seating surface is not easy and (2) there might be some sticking if subject to temperature changes where the body contracts more than the disc. For this latter condition, a flexible disc would provide better service.

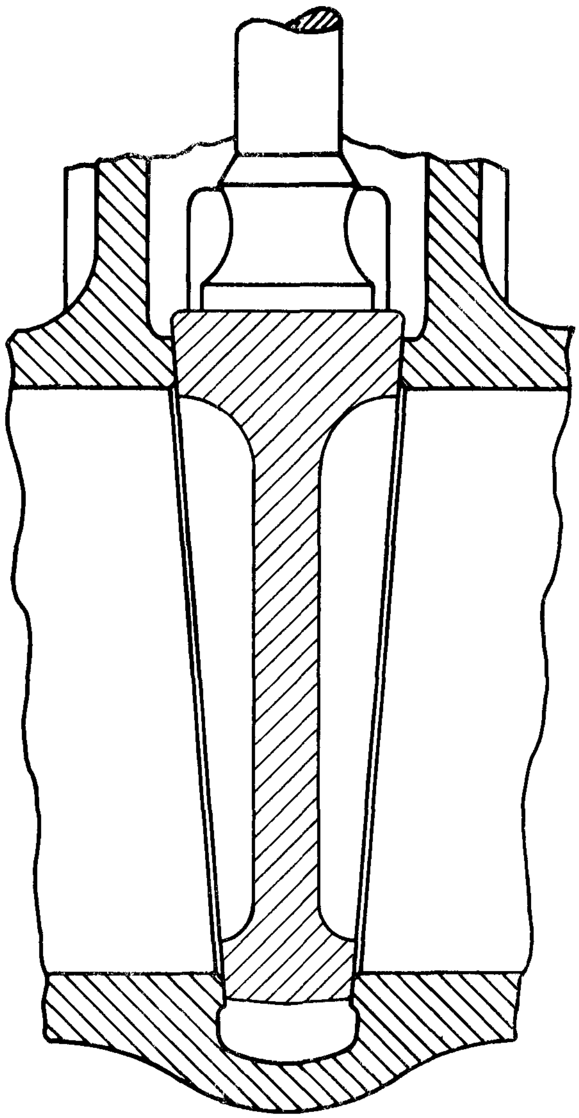


Fig. 2.2. Solid Wedge Gate With Matching Tapered Body Seating Surfaces.

The flexible wedge disc is illustrated in Fig. 2.3. Its limited flexibility is achieved by designing essentially matching wedges (disc faces) that are connected with a membrane. The advantage of a single-piece disc is retained while additional provision is made for one disc face to move slightly with respect to the other. This flexibility helps to make the valve tight on both faces and it also acts to prevent sticking. This flexibility helps to retain a seal in the event of pressure-temperature fluctuations or possible valve body distortions.

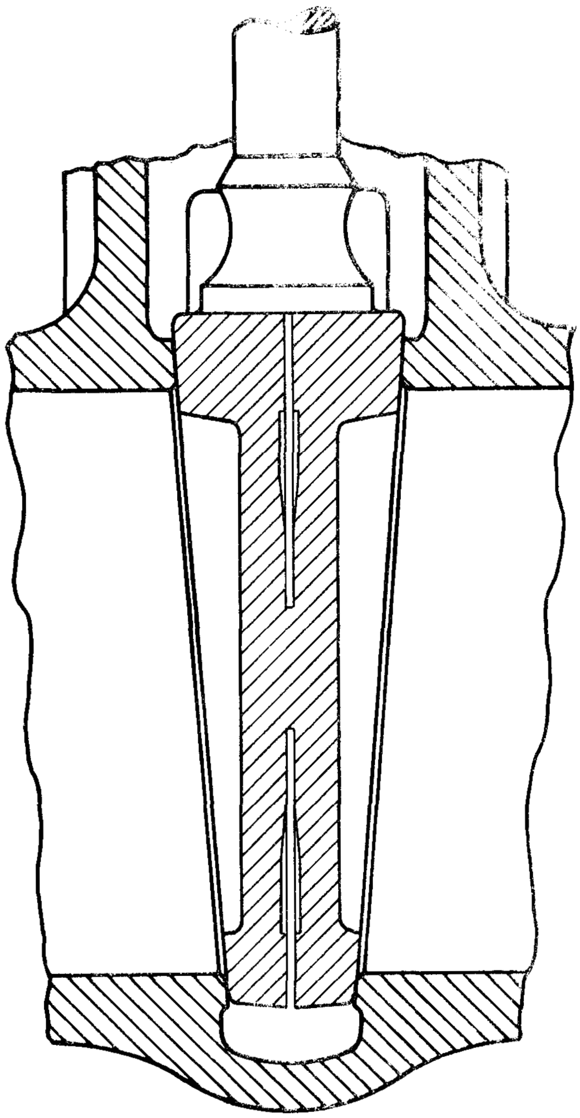


Fig. 2.3. Flexible Wedge Disc.

The split wedge disc, illustrated in Fig. 2.4, has increased flexibility with consequent improved ability to orient to the seat. The split wedge disc is a two-piece wedge disc that seats between matching tapered seats in the body with each disc seating independently. The two disc pieces fit loosely on the stem and are positioned flat against each seat as they pivot back-to-back when wedged between the valve seats.

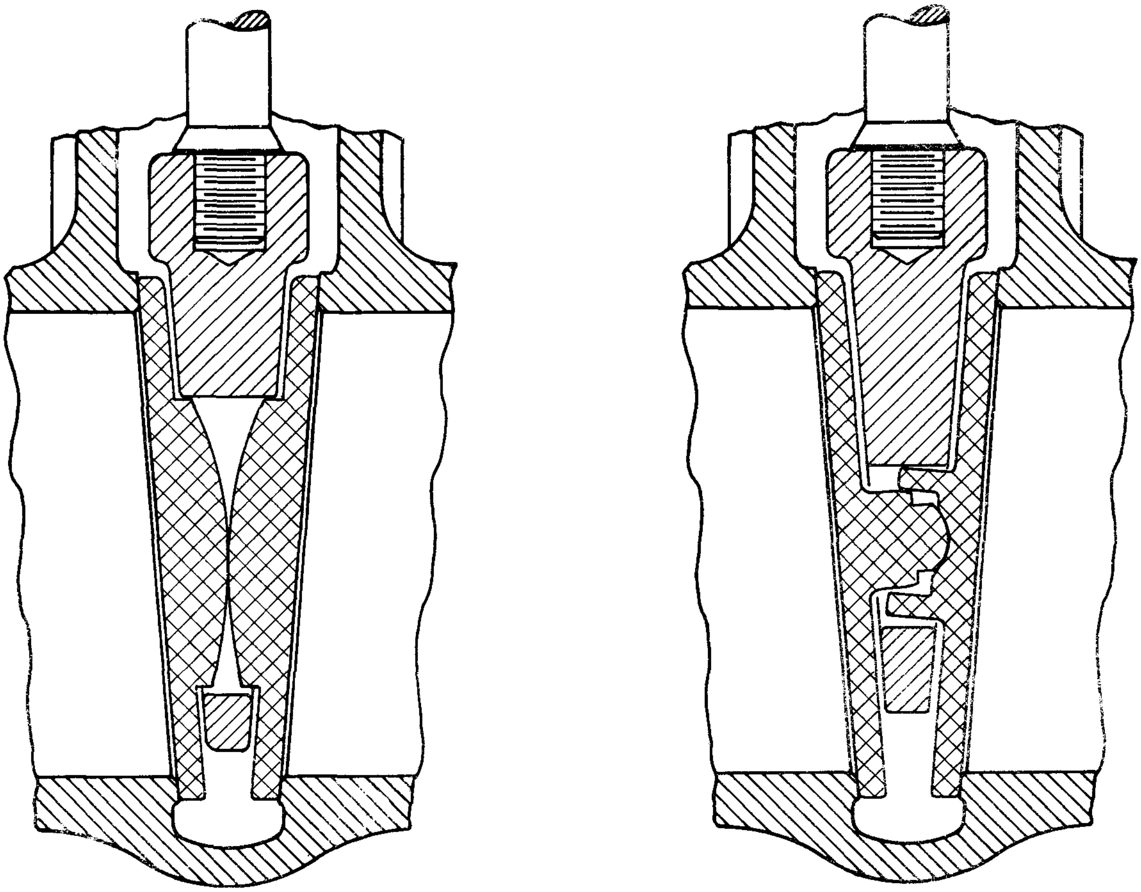


Fig. 2.4. Typical Split Wedge Discs.

Gate valves are also designed with parallel-faced discs and with the seats in the vertical plane. The closure is made as the parallel-faced disc is moved between matching parallel seats in the valve body. With the discs and body seats parallel, repairing or refacing to compensate for wear is easier than it is on the tapered wedge disc. The parallel orientation also causes the disc to "wipe" the seating face as the valve is closed, thereby tending to clean the contact area of any foreign matter. Large low-pressure gate valves often use a solid parallel-faced disc, whereby the disc seats on only one side since all seating force is derived from the fluid pressure differential. In most applications, the parallel-faced disc gate valve employs a split disc with a spreader arrangement to wedge the separate disc faces outward against the valve seats as the stem approaches its full down position, as illustrated in Fig. 2.5.

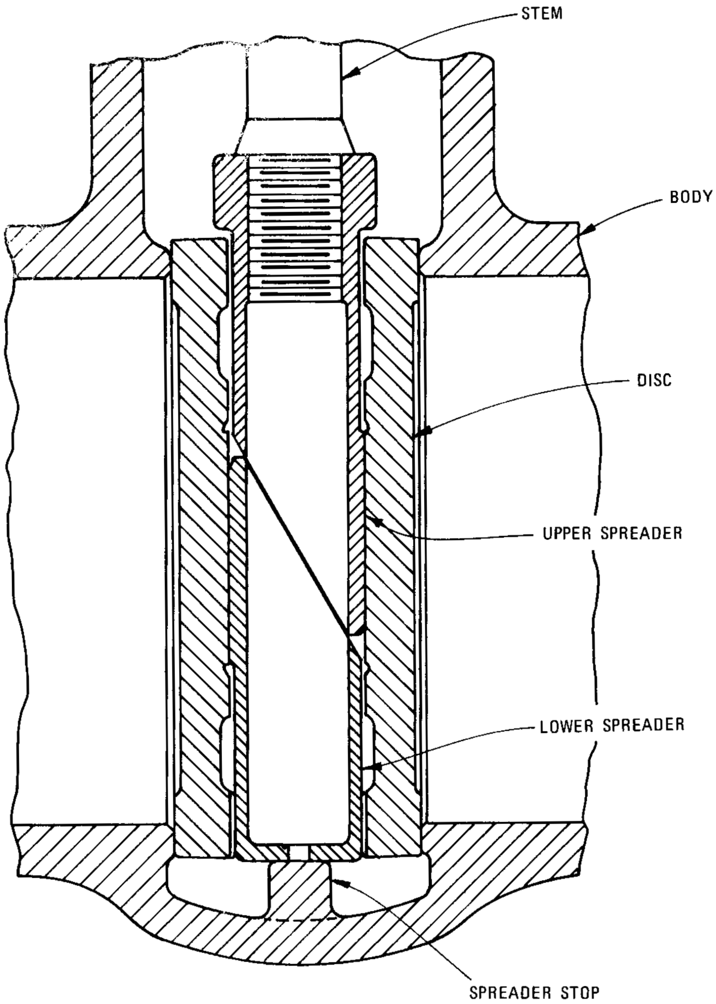
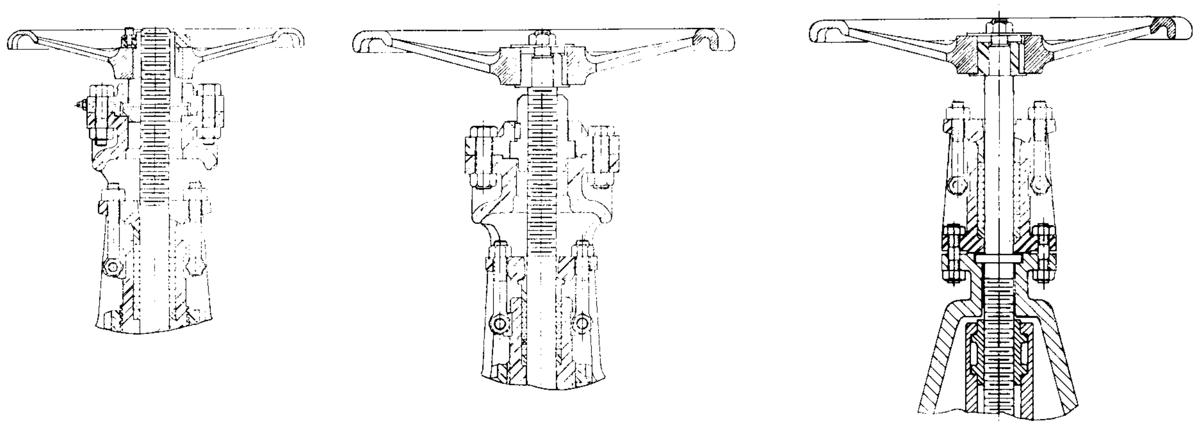


Fig. 2.5. Parallel Disc Gate With Spreader Arrangement.

2.1.2 Stem and Screw Arrangements

The stem in a gate valve has the single function of raising and lowering the disc. To keep the stem from being subjected to the stresses and strains of the disc service connections, a relatively loose disc-to-stem connection is generally used. Many gate valves are equipped with a guide slot on the outer edge of the disc that mates with a guide strip on the inside of the valve body to hold the wedge disc loosely centered between the valve seats as it is raised into its recess in the bonnet. The combination of a disc guide and the loose disc-to-stem connection precludes transmittal of the side forces on the disc resulting from fluid pressure and flow within the valve to the stem. The stem generally has a tapered shoulder for back-seating that mates with a corresponding tapered seat in the bonnet below the stuffing box. This provides a seal when the valve is fully open and reduces leakage to the stuffing box. It also provides a seal which permits the valve stem to be repacked without draining the system piping.

In most gate and globe valves, the stem incorporates a screw mechanism for opening and closing the valve. The stem is commonly operated with a handwheel, and the diameter of the handwheel is selected to limit the torque applied to the stem. Stem motion may be varied to suit service needs. The three stem screw arrangements most often encountered are illustrated in Fig. 2.6.



(a) RISING-STEM NON-ROTATING SHAFT

(b) RISING-STEM ROTATING SHAFT

(c) NON-RISING STEM

Fig. 2.6. Most Frequently Encountered Stem Screw Arrangements.

The majority of large high-pressure valves have the non-rotating rising-stem design illustrated in Fig. 2.6(a). The screw threads on the stem mate with the screw threads in the yoke nut. As the handwheel is turned, it rotates the yoke nut and moves the non-rotating stem axially through the stem packing. As the stem moves, it carries the disc with it. The straight in-and-out movement of the stem provides for a simple and reliable stem seal design. The screw threads are located outside of the fluid boundary and are easily accessible for lubrication and maintenance.

Smaller low-pressure valves sometimes have the rotating rising-stem design illustrated in Fig. 2.6(b). In this design, the stem nut is fixed to the bonnet either in the yoke, as shown, or just below the stem packing and inside the fluid boundary. The handwheel is keyed to the stem, and turning the handwheel causes the stem to rotate as it advances axially with the screw pitch. Because the handwheel also moves axially, this design is sometimes called a rotating-stem rising-handwheel arrangement. This design has the disadvantages of requiring excessive headroom clearance for the handwheel in the raised position, a rotating joint between the stem and disc, and of requiring frequent stem packing maintenance.

In certain process system installations, the lack of headroom over a valve may necessitate the use of a non-rising stem, as illustrated in Fig. 2.6(c). In this design, the stem nut is mounted in the valve disc, and rotation of the stem raises or lowers the disc as the stem nut advances on the stem threads. A collar prevents any axial motion of the stem, and the handwheel remains at one elevation. The major disadvantage of the non-rising stem is the location of the stem threads inside the valve fluid boundary where they are often exposed to a corrosive fluid or to sediment which damages the thread surfaces on operation. Lubrication is often impossible because of the solubility of the lubricants or because of problems associated with contamination of the process stream. Nor can the non-rising stem be fitted with a back-seating feature.

2.2 Globe Valves

The globe valve derives its name from the roughly globular shape of the valve body, as illustrated in Fig. 2.7. The body forms the flow passages for the movement of fluid through the open valve. The flow passage is blocked by closing the disc against the valve seat. Movement of the disc is accomplished by means of the stem which penetrates the bonnet through a dynamic seal.

Unlike the perpendicular seating of a gate valve, the globe valve seating is parallel to the line of flow. Contact between the disc and seat is broken completely with the initial opening movement of the valve

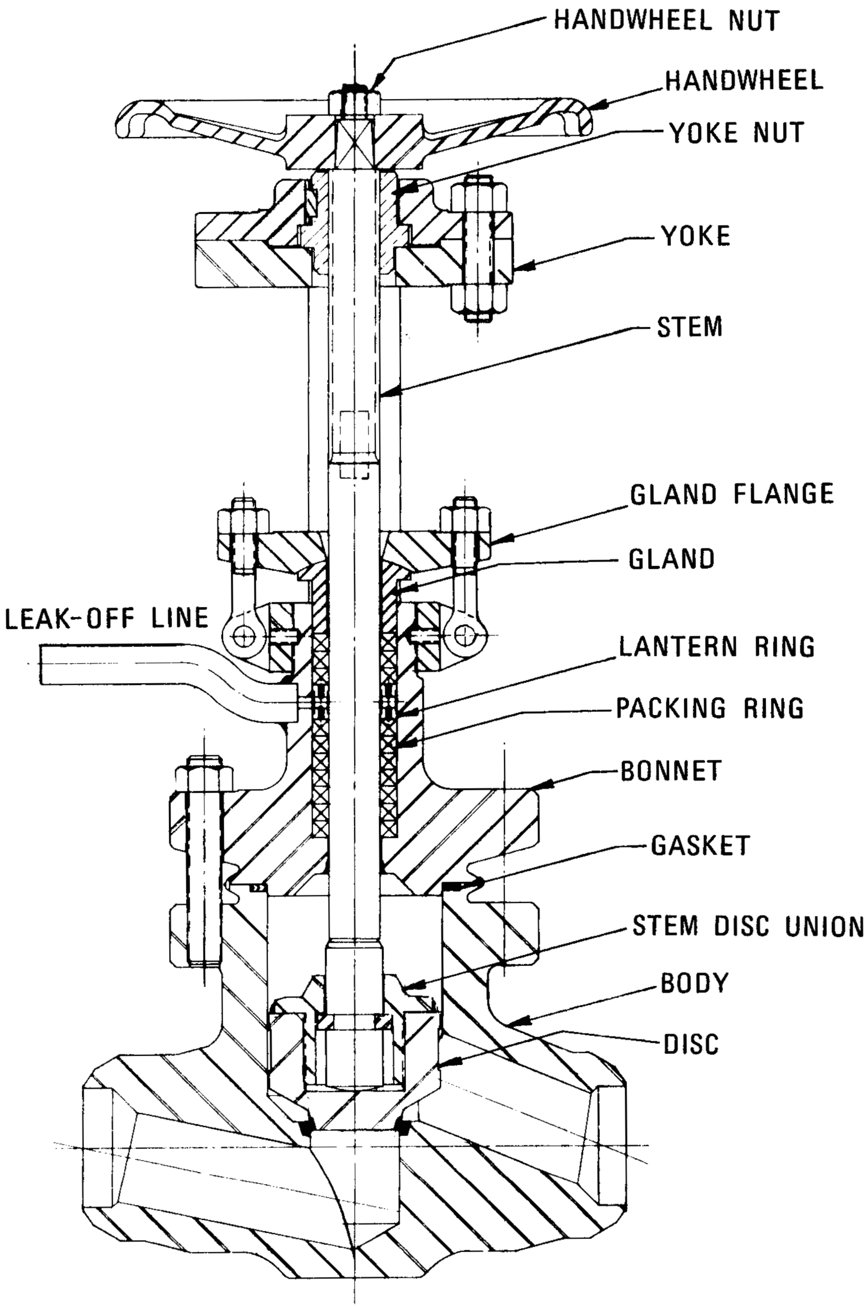


Fig. 2.7. Typical In-Line Globe Valve.
(Courtesy Velan Engineering Companies)

stem, and the sealing surfaces move apart almost perpendicularly rather than sliding upon each other as in the gate valve. Most globe valves are designed with a "line-contact" seal; that is, the sealing surfaces are in contact only along a narrow line. This line contact, being less subject to the individual irregularities of wider mating surfaces, generally makes a tight metal-to-metal seal more easily. The line-contact seal is described in detail in Section 4.2.

Fluid flow begins as soon as the disc and seat separate, allowing a more efficient throttling of flow with minimum seat erosion. The flow rate through the valve may be regulated closely by an appropriate adjustment of the handwheel. Globe valves are generally specified when throttling service is required. Some globe valves have special features to improve their throttling characteristics. Globe valves may be operated in the partially open or throttling mode without sacrificing seat life, and they will still provide a positive shutoff when fully seated. They continue to seat tightly after many operating cycles and are therefore used when valve operation is expected to be frequent.

Globe valves may be designed for shorter disc travel than gate valves, simplifying the stem screw design and reducing the overhead clearance requirements. Globe valves force the fluid stream to change direction as it follows the fluid passages through the valve body. As a consequence, globe valves represent a much greater flow resistance than gate valves, for example, and they may cause a significant pressure loss in the flowing stream.

2.2.1 Globe Valve Body Styles

The globe valve shown in Fig. 2.7 is a conventional in-line valve; that is, the inlet and outlet passages are designed on a common axis. The flow direction is changed a number of times as the fluid passes through the open valve, and these in-line globe valves represent a high resistance to the flow stream. Globe valves are offered with two alternate body styles that offer less resistance to the fluid flow. These are the angle body and the Y-body.

The angle valve utilizes the seating principle of the globe valve while providing a 90° turn in the piping, as illustrated in Fig. 2.8. Angle valves perform similarly to globe valves but offer slightly less resistance to the flow stream. They are ideal when the direction of

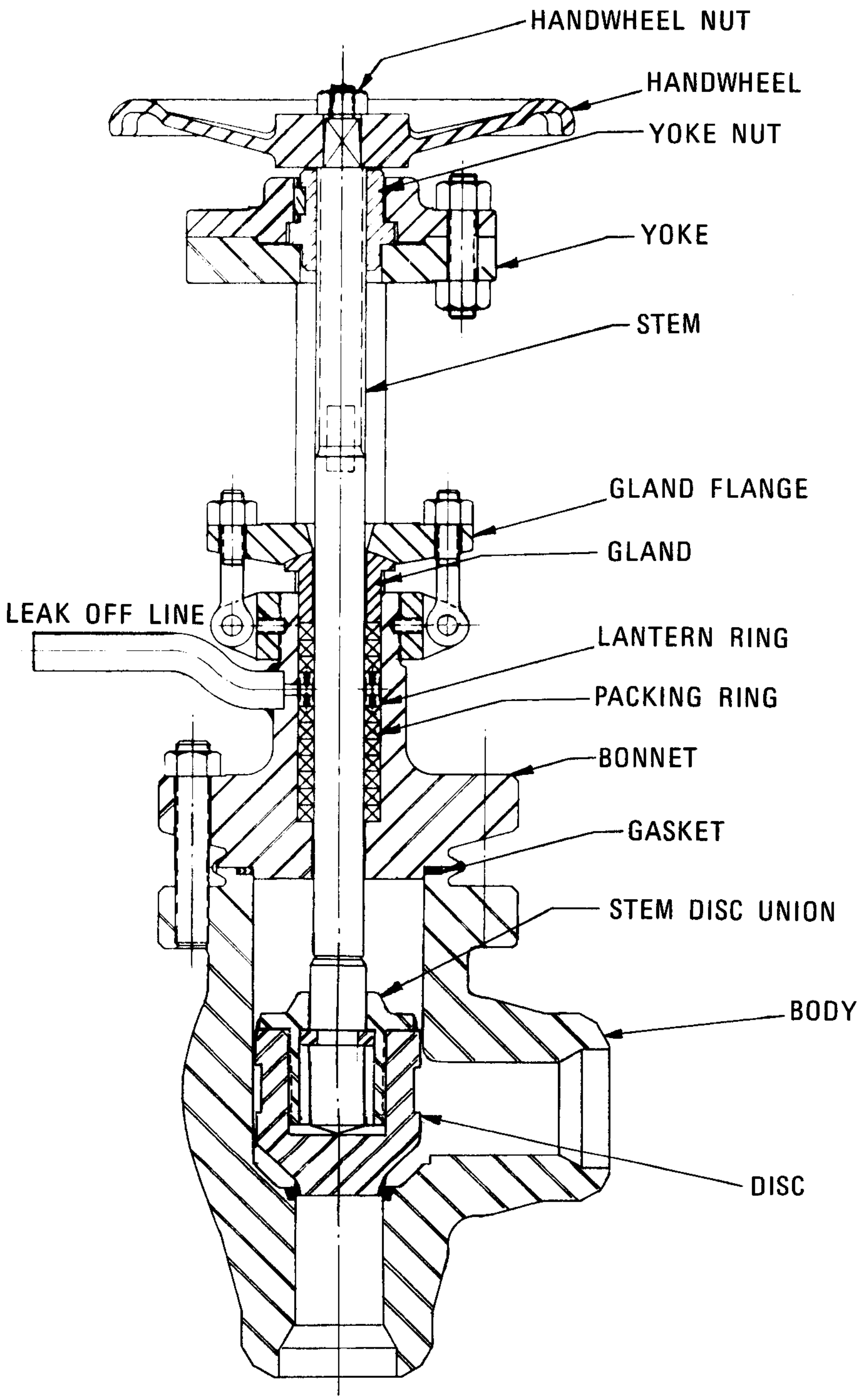


Fig. 2.8. Typical Angle Globe Valve.

flow must be changed inasmuch as they combine the physical results of an in-line globe valve and an elbow in one component, thereby eliminating a fitting. The flow resistance of the angle valve is also considerably less than the combined resistance of an in-line globe valve and an elbow in series. The angle valve can have the same seating variations as the regular in-line globe valve, and angle valves are also suitable for throttling control of flow and will provide positive tight closure after frequent use.

The Y-body valve, illustrated in Fig. 2.9, is still another variation of the globe valve. This valve minimizes the flow resistance while retaining the good throttling and cyclic operating life characteristics of its predecessor. The seat is positioned in the valve body at an angle roughly 45° to the inlet-outlet axis of the valve, and the stem-bonnet assembly makes an angle with the body roughly in the shape of a "Y". The directional change of the fluid necessary in this type of valve is reduced significantly, thereby lowering the fluid resistance losses. This reduction in fluid resistance loss is the principal advantage of the Y-body construction over the more conventional in-line globe valve.

2.2.2 Stem and Bonnet Arrangements

In addition to raising and lowering the disc, the stem in a globe valve guides the disc squarely to its seat and restrains the disc from fluttering when the valve is operated in the throttling mode. Thus, the disc-stem connection in the globe valve must be close fitting or the disc must be provided with a guide mechanism fixed to the seat or valve body. If the valve utilizes the rotating stem design, the disc must stop turning while seating force is applied by the stem. Once the disc contacts the seat, the seat turning must stop to avoid metal-to-metal friction between the disc and seat that would be destructive to the respective surfaces. In the rotating stem arrangement, the disc-stem connection must also provide a swivel action to achieve tight seating without seal damage. The back-seating feature is incorporated in the stems of globe valves.

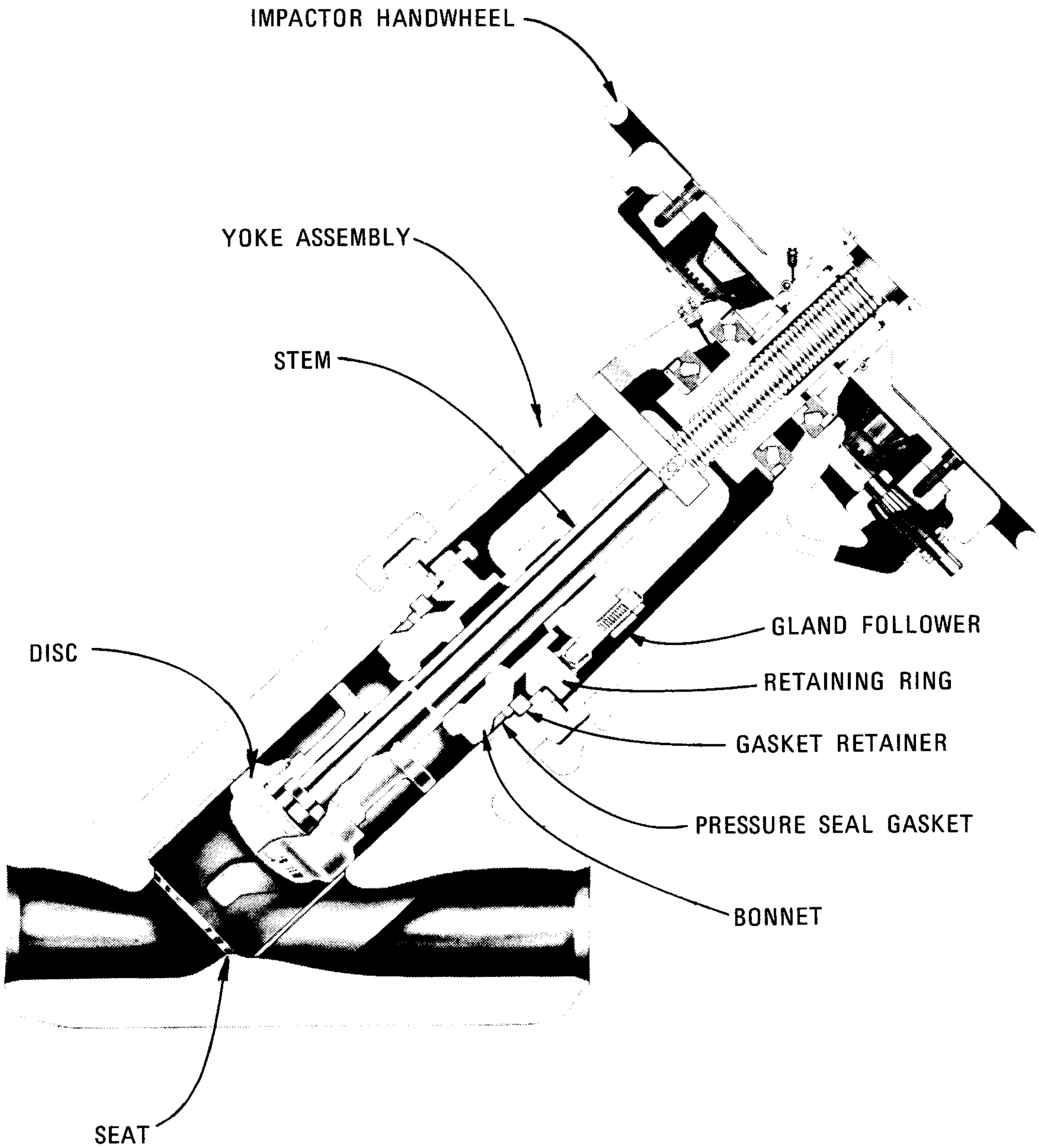


Fig. 2.9. Typical Y-Body Globe Valve.
(Courtesy Rockwell Manufacturing Company)

The bonnet-yoke-stem arrangements, such as inside or outside stem threads, used on gate valves are also used in globe valve construction. The globe valve generally has less disc travel from open to shut positions, resulting in fewer turns of the handwheel for operation. This means a savings in time, work, and wear of valve parts. The design also makes repair of the disc and seat relatively easy without removing the valve from the line. A seat lapping tool, fashioned much the same as the stem-disc assembly, with a jig used in place of the bonnet can be used to grind the seat and resurface the seat contact area.

2.3 Check Valves

Check valves are designed to permit flow in one direction only, and they automatically close when the fluid stream starts reverse flow. Functionally, check valves are used to prevent backflow in the process line. They close almost instantaneously in the event of a flow reversal without requiring any action or knowledge on the part of the plant operator.

There are three basic types of check valves: (1) swing check valves, (2) lift check valves, and (3) stop check valves. The swing check valve consists of a disc, generally attached to an arm, that swings open and out of the fluid stream when the fluid flows through the valve in the desired direction. The lift check valve consists of a guided disc which is lifted off of the valve seat by the upward flow of fluid passing through the valve. The lift check valve is physically similar to a globe valve without any stem and screw mechansim, and the position of the disc is controlled solely by the fluid flow conditions. The stop check valve is a lift check valve equipped with a stem and screw mechanism affording a positive shut-off capability for flow through the valve in either direction. When the stem of the stop check valve is in the full open position, the valve functions exactly like a lift check valve. When the stem is in the valve closed position, the valve functions like a closed globe valve.

In check valves, the weight of the disc (gravity) is used to initiate disc closure, and the fluid backflow pressure provides the disc seating force. Some swing check valves also have springs or external counterweights to improve disc closure characteristics. The fluid flow pressure holds the disc in the open position. It is important that check valves be operated with the disc in the full open position; that is, the disc should be tight against its stops. Most check valves are designed to achieve the disc full-open condition with relatively low flow rates for the size valve selected. Nevertheless, check valves installed in process systems that operate at partial flow a significant percentage of the time should be undersized rather than oversized. Most of the difficulties encountered with both swing and lift check valves have been caused by the selection of oversized valves. This results in disc fluttering in a partially open position, causing noisy operation and premature wear of the moving parts.

2.3.1 Swing Check Valves

The swing check valve is so designated because closure is accomplished by a swinging disc contacting the seat. The variations on the swing check valve include the conventional swing check valve illustrated in Fig. 2.10 and the tilting disc type illustrated in Fig. 2.11. The fluid friction

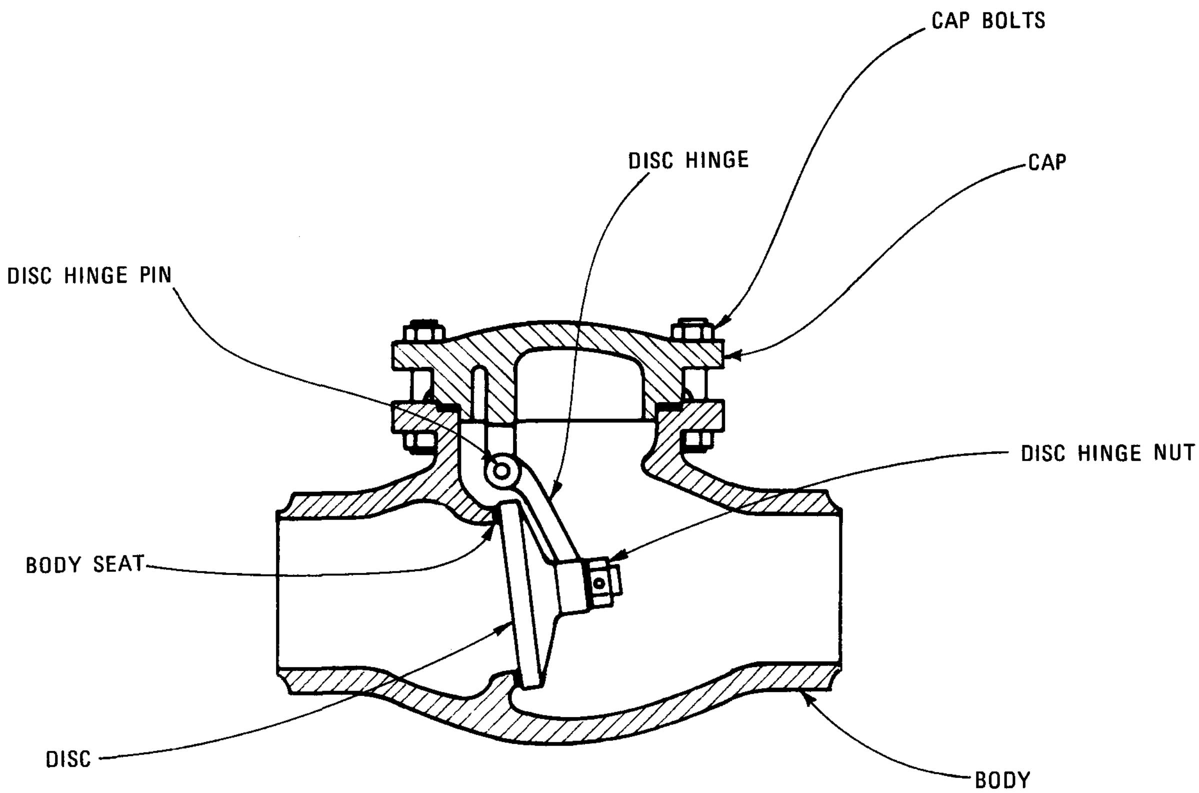


Fig. 2.10. Conventional Swing Check Valve.

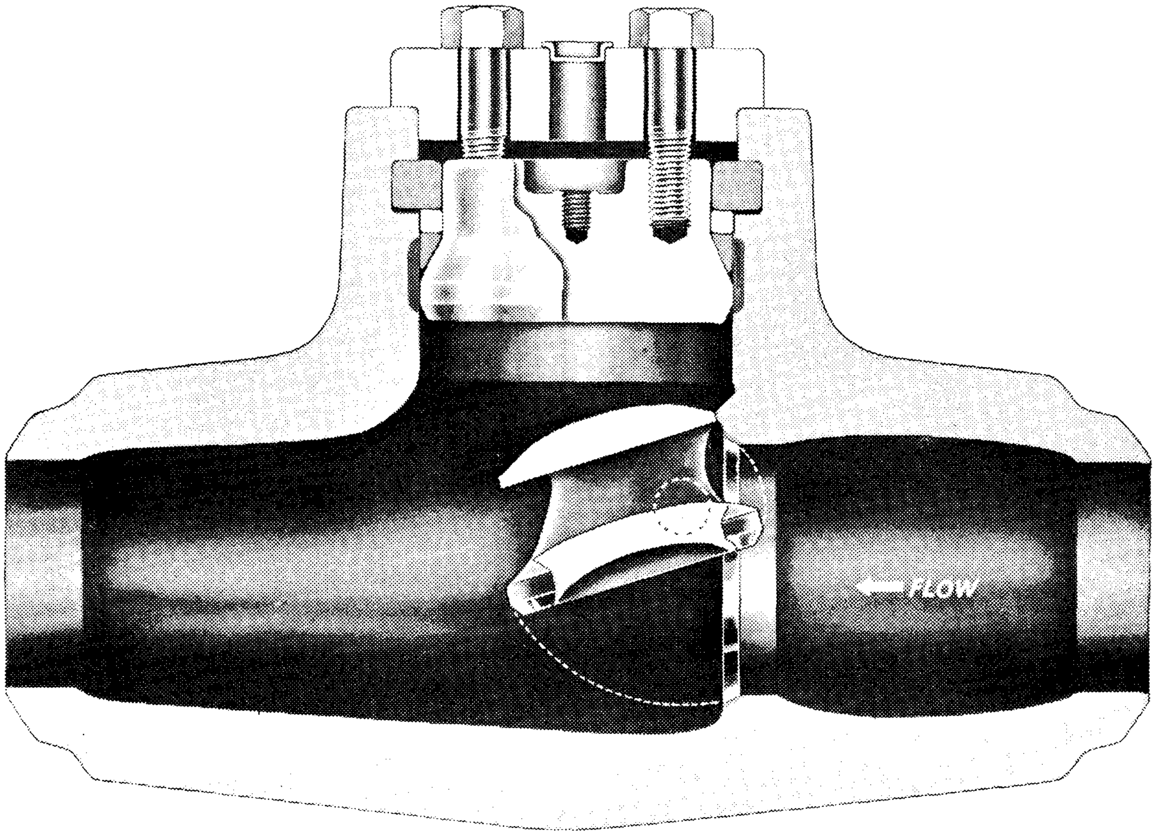


Fig. 2.11. Tilting Disc Check Valve.
(Courtesy Rockwell Manufacturing Company)

loss through swing check valves is relatively low because a high cross sectional area is available for flow when the disc is fully open and the fluid passes straight through the valve without a change of direction.

The seating surface in swing check valves is flat and mates with a corresponding flat surface on the disc. As in a gate valve, the contact area is relatively broad. The design is necessarily a compromise between the tolerances required to maintain good disc-seat alignment and the clearances necessary to insure freedom of motion. Emphasis should be placed on pin alignment and materials to retard pin bearing wear and corrosion insofar as possible. The seat tightness characteristics of the swing check valve are generally poorer than those of the lift check valve because of the type of seating (broad area contact versus line contact) and because the design inherently has a lesser capability for guiding the disc to a good contact with the seat. Swing check valves may also have external counterweights to improve the disc closure performance, as illustrated in Fig. 2.12. These counterweights are also used to suppress disc chatter in valves which experience frequent flow reversals and low-flow conditions.

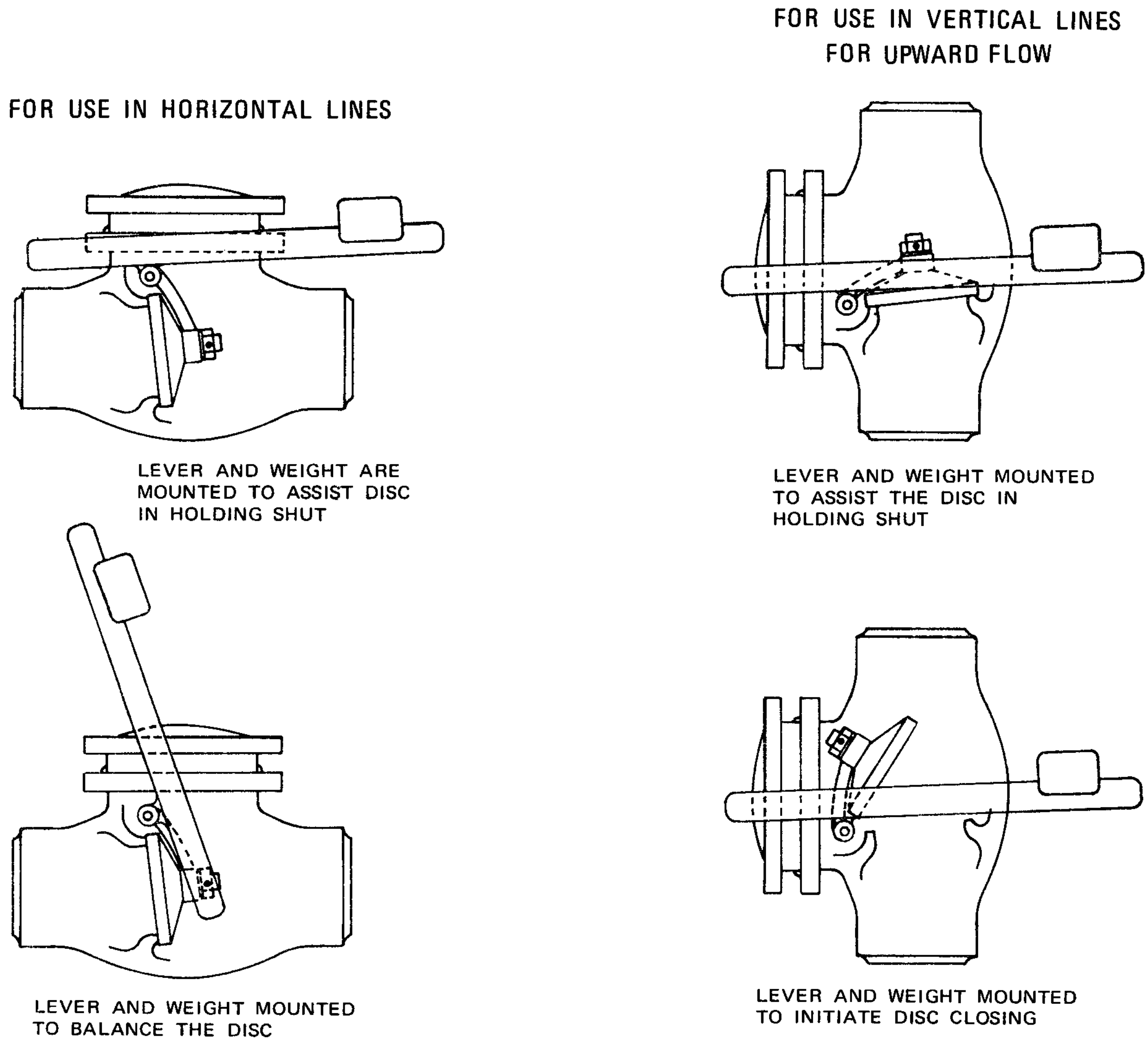


Fig. 2.12. Use of External Counterweights on Swing Check Valves in Horizontal and Vertical Lines.

2.3.2 Lift Check Valves

The lift check valve has a body and seating arrangement similar to that of a globe valve, and it may be designed in the three equivalent body styles of the globe valve: the in-line body, angle body, and the Y-pattern body. A typical globe lift check valve is illustrated in Fig. 2.13, and a typical angle lift check valve is illustrated in Fig. 2.14.

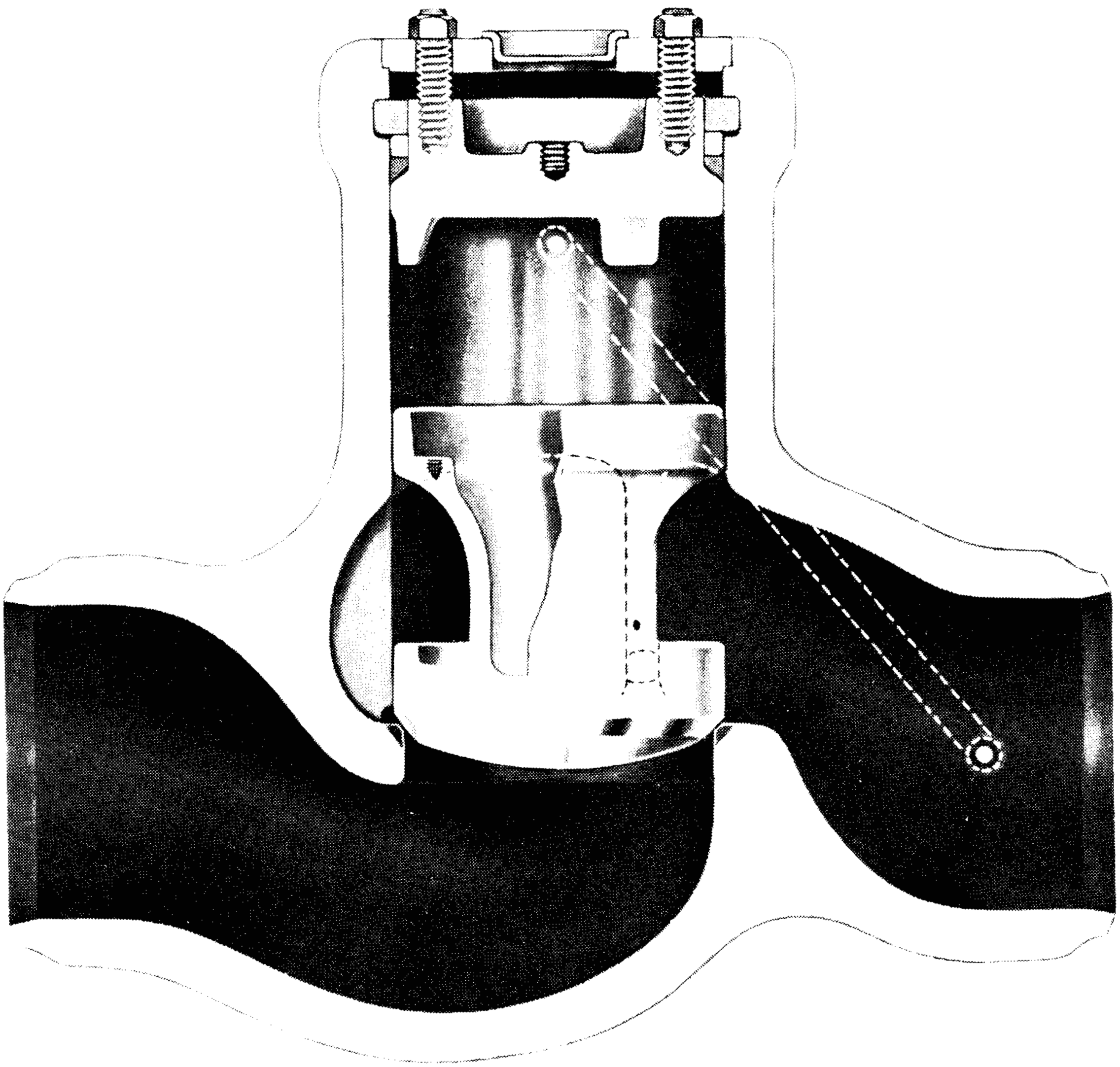


Fig. 2.13. Typical Globe Lift Check Valve.
(Courtesy Rockwell Manufacturing Company)

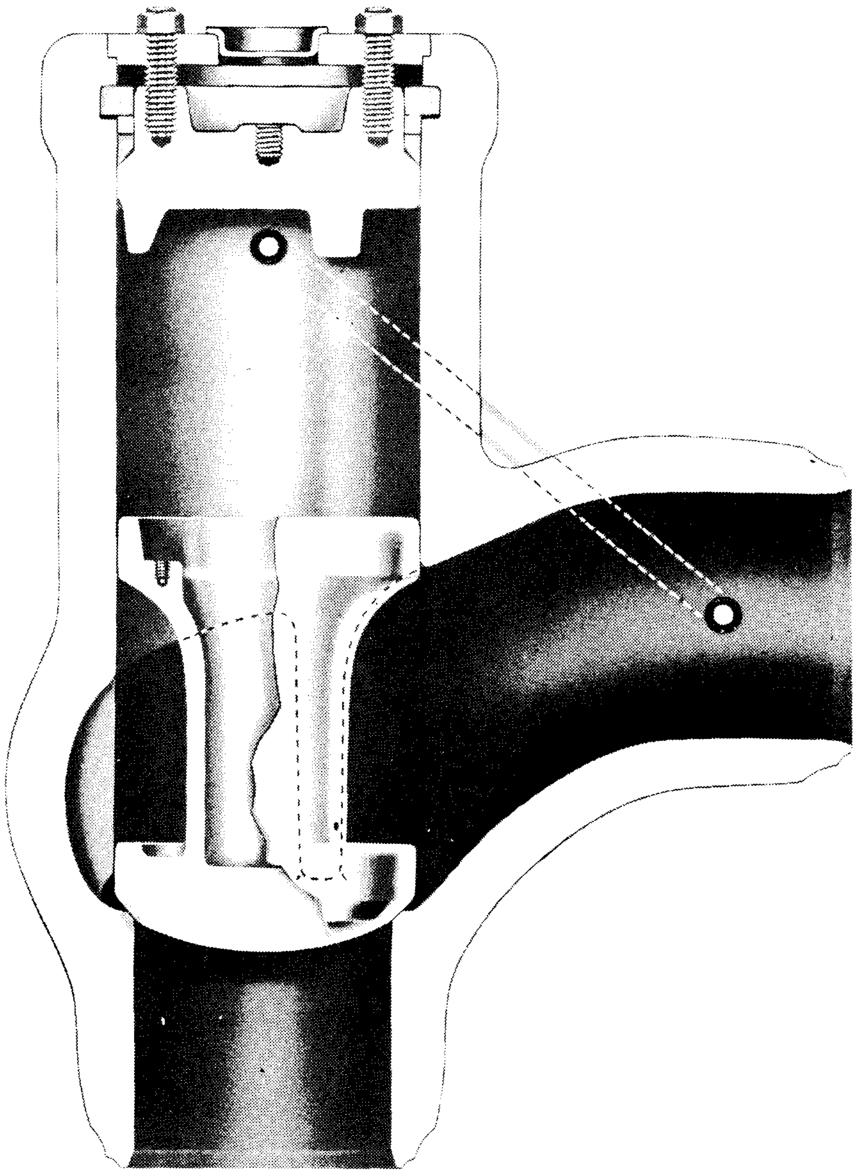


Fig. 2.14. Typical Angle Lift Check Valve.
(Courtesy Rockwell Manufacturing Company)

The lift check valve has disc and seat features similar to those of a conventional globe valve. The seats are shallow and tapered, and the discs have a mating taper. Line-contact seating is used, and this generally results in less seat leakage than encountered in the swing check valve. The lift check valve has no stem, and the disc must be guided from the valve body. Consequently, the discs are fitted with guide extensions which are basically abbreviated stems acting as a piston with a sliding fit in a cylinder above the seat. Many designs also include fluted guide extensions that extend from below the disc with the seat hole used as an additional centering guide, as illustrated in Fig. 2.15. This arrangement also aids self-centering of the disc by lowering its center of gravity.

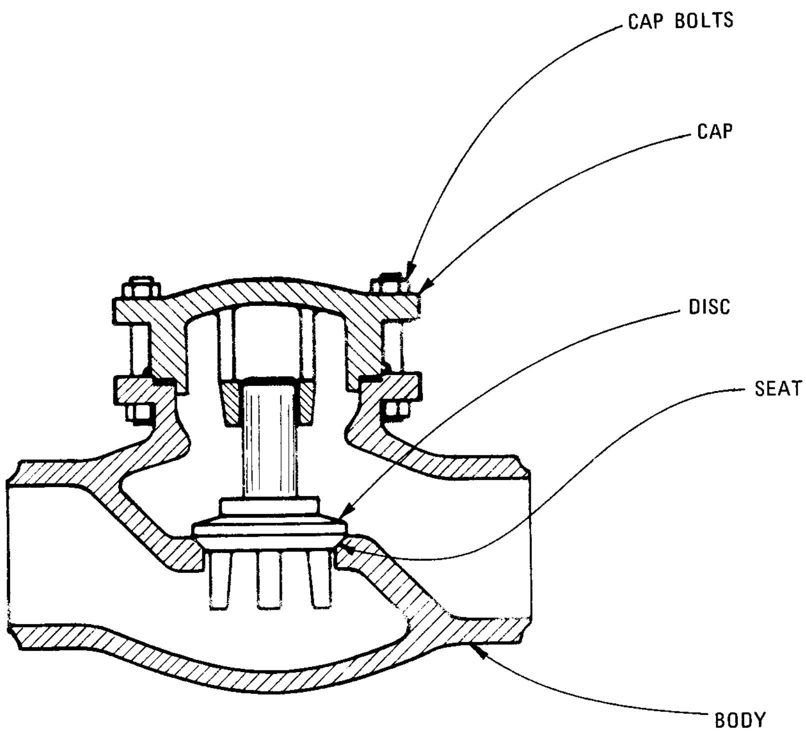


Fig. 2.15. Lift Check Valve With Lower Disc Guide.

The lift check valve is prone to cocking of the disc in the guide and consequent sticking in the open position. There is also the potential for corrosion product buildup in the guide cylinder that also leads to sticking in the open position. This sticking tendency can be overcome to some extent by providing a bleed passage from the downstream side of the valve to the guide cylinder above the disc "stem". This exerts a seating force to close the valve when reverse flow is initiated.

The pressure drop through the lift check valve is greater than that for the swing check valve because of a lesser flow area and the changes in the direction of flow required for the fluid to pass through the lift check valve. In the full-open position, the flow resistance of the lift check valve is approximately equal to that of an equivalent globe valve. However, the use of fluted guide extensions below the disc will significantly raise the flow resistance of the valve.

2.3.3 Stop Check Valves

The stop check valve is physically similar to a globe valve with the stem-disc connection modified to permit the disc to fall closed with the stem in the valve open position. When the stem is raised, the valve functions as a lift check valve; and when the stem is down, the valve functions as a closed globe valve. The stop check valve may also be designed in the three equivalent body styles of the globe valve: the in-line body, angle body, and the Y-pattern body. A typical angle stop check valve is illustrated in Fig. 2.16. The physical similarity between globe valves and stop check valves is so great that manufacturers furnish many stop check valves in the same body pattern used for their globe valves, and it is difficult to distinguish between them from outward appearances.

As in both the globe valves and the lift check valves, a line contact seal is used in the stop check valve. The disc is generally guided by its relatively close but sliding fit on the stem, and fluted guide extensions on the underside of the disc are seldom needed in stop check valves. The sliding fit of the disc on the stem is critical to the automatic functioning of the valve upon loss of fluid flow. A bleed line is also used in stop check valves to initiate disc closure in the event of a flow reversal. The positive closure feature is an advantage in the event the valve sticks open if the time delay in closing of the valve can be tolerated.

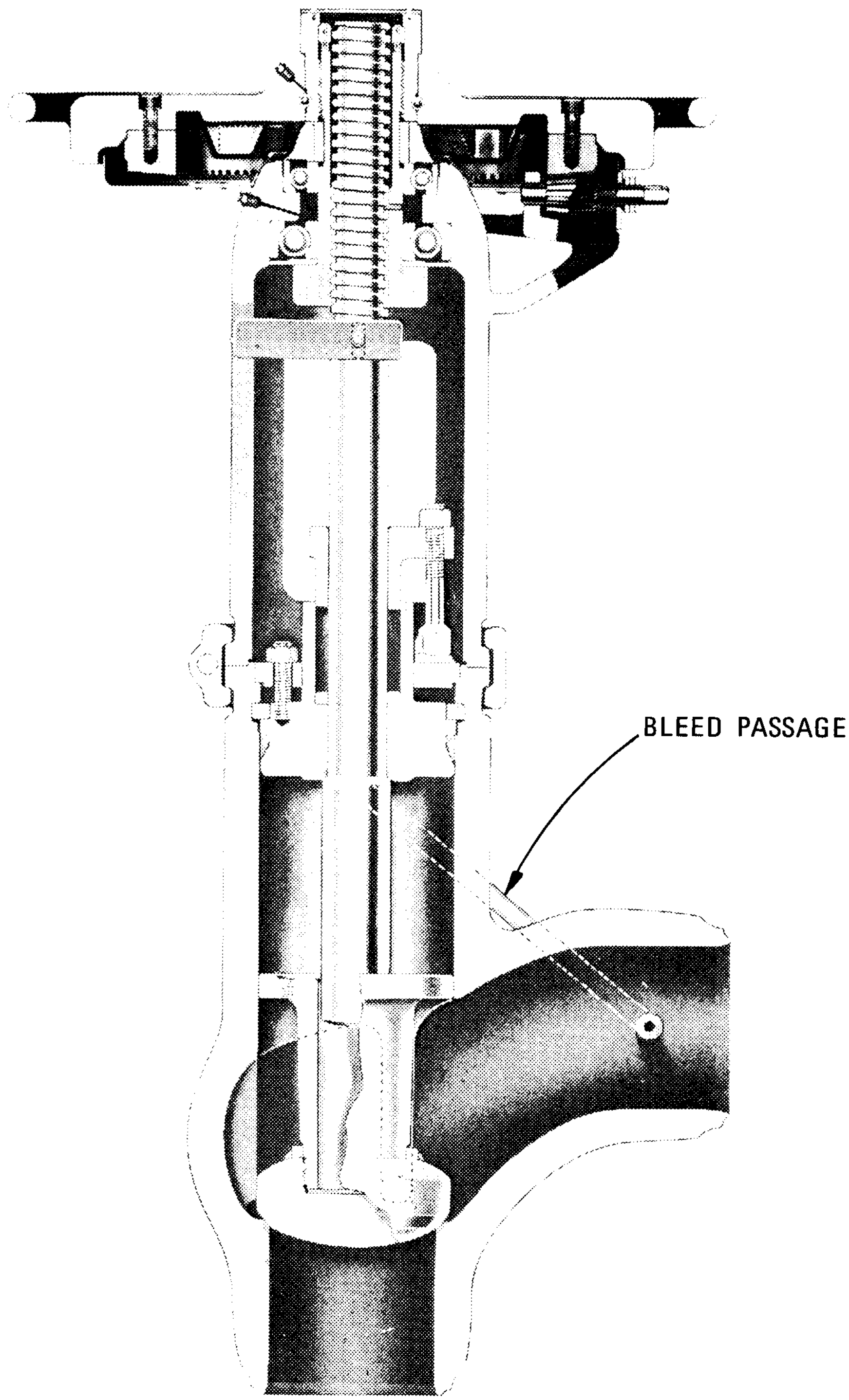


Fig. 2.16. Typical Angle Stop Check Valve.
(Courtesy Rockwell Manufacturing Company)

3. GENERAL DESIGN CONSIDERATIONS

The general design considerations for reliable nuclear valves discussed in this chapter include codes, standards, and specifications; flow design; materials; structural design and analysis; fabrication; and installation, maintenance, and inservice inspection.

3.1 Codes, Standards, and Specifications

The primary purpose of the codes is to assure safety. To this purpose, the codes set forth requirements for design, fabrication, and quality control that assure that the piping system, vessel, or valve to which they are applied will not rupture in service. However, codes are not design handbooks and do not eliminate the need for the engineer or competent engineering judgment.[1] The codes do give a complete listing of computational methods and work practices that experience has shown to be necessary for reliable service.

The primary purpose of standards and specifications is the establishment of uniformity in materials, dimensions, and manufacturing processes, and quality control procedures. The scope of standards is far more narrow than that of codes. Generally, a standard covers only a component, such as a flange, or a larger entity or a procedure, such as a dye penetrant inspection procedure, that represents only one step in the total effort of supplying the entity.

Codes and standards are interdependent. The codes refer to standards for documentation of industry-accepted parameters. Thus, the standards and specifications serve as a short method of describing materials and processes that will be satisfactory when used in the manner specified in the codes. However, the use of industry-accepted standards and specifications does not by itself insure a suitable valve. Satisfactory performance of the valve is assured only when the use of these standards and specifications is restricted to carefully defined conditions, such as an allowable temperature and pressure range. Using the standards and their resultant properties, the codes present design formulae and fabrication techniques that are intended to insure a safe design.

Codes and standards are promulgated under the concept that invoking the requirements therein is voluntary; that is, voluntary in the sense that they are originated and self-imposed by an amalgam of manufacturers, consumers, professional, technical, and trade associations. In some cases, local and state governments have enacted certain codes into law. For example, certain portions of the ASME Boiler and Pressure Vessel Code have become law in many states. The Atomic Energy Commission encourages the adoption of industry-accepted codes and standards in all nuclear power plants. Adoption of standards has resulted primarily from competitive needs without force of law. The common mode of imposition of both codes and standards is by contractual obligation; that is, a vendor is obligated to supply a component manufactured under the requirements of the codes and standards identified within the contract between the supplier and purchaser. As a consequence, these codes and standards acquire the force of law.

3.1.1 Codes Applicable to Nuclear Valves

There are presently four codes which are applicable to the design, fabrication, and application of nuclear valves. These are the

1. Draft ASME Code for Pumps and Valves for Nuclear Power issued for trial use and comment in November 1968;
2. ANSI Standard Code for Pressure Piping, ANSI B31.7-1969, Nuclear Power Piping;
3. ASME Boiler and Pressure Vessel Code, Section XI, Inservice Inspection of Nuclear Reactor Coolant Systems, issued in 1970; and the
4. ASME Boiler and Pressure Vessel Code, Section III, Nuclear Vessels, issued in 1968.

The ASME Code for Pumps and Valves for Nuclear Power is specifically applicable to nuclear valves. The convention of rating valves by their design pressure and temperature conditions is followed. Tables are included that establish seven pressure-temperature ratings (150, 300, 400, 600, 900, 1500, and 2500 lb) for valves made of 17 different steel materials. Design pressures are listed in the tables for each material for temperatures from -20°F to +700°F for ferritic steels and from -20°F to +800°F for austenitic steels. These pressure-temperature rating tables are identical to those used by the industry for conventional steel valves. This code also includes a table of minimum wall thicknesses listed by pressure-temperature rating and the inside diameter of the valve body.

The ANSI B31.7 Code for Nuclear Power Piping is directly applicable to the installation of valves in the piping system, and it is applicable by reference to the design of the body-to-bonnet seal and the bypass lines. The relationship between the ASME Code for Pumps and Valves for Nuclear Power and the ANSI B31.7 Code for Nuclear Power Piping is illustrated in Fig. 3.1.

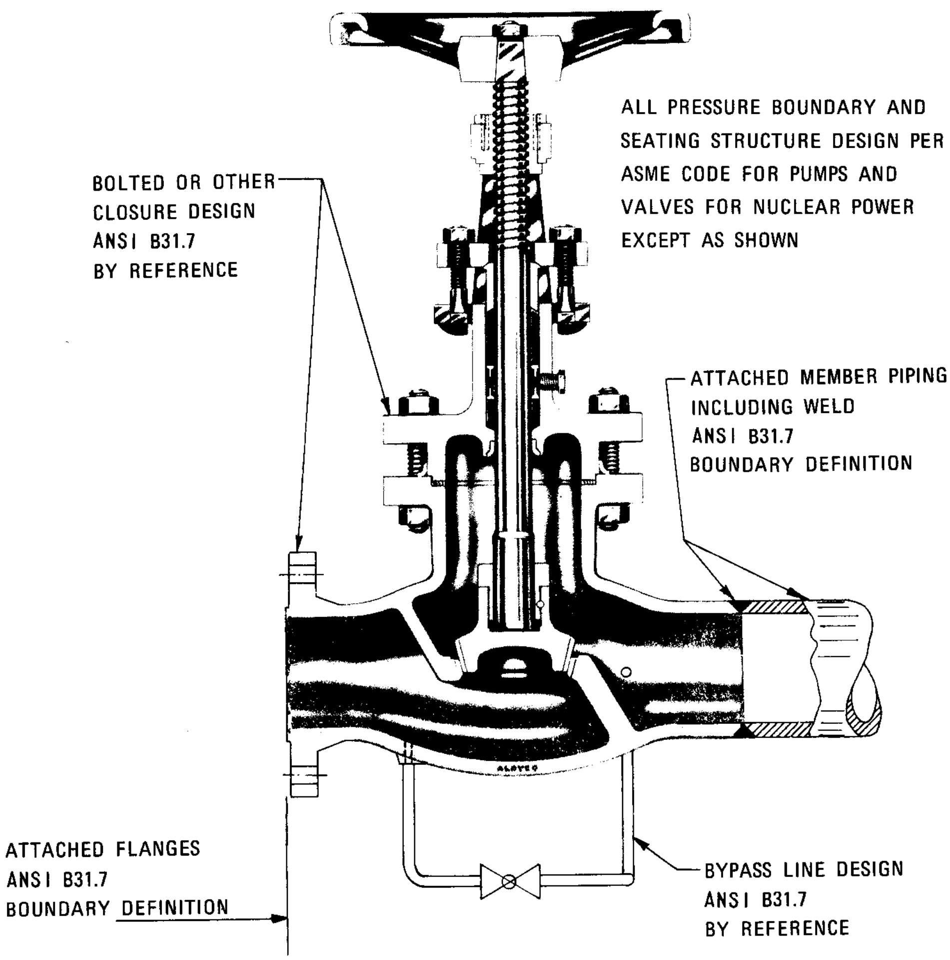


Fig. 3.1. Relationship Between Nuclear Valve and Power Piping Codes.

Section XI, Inservice Inspection of Nuclear Reactor Coolant Systems, of the ASME Boiler and Pressure Vessel Code is applicable to inspections which must be performed at stated intervals throughout the lifetime of the piping system. This document is of interest because certain methods of valve fabrication will influence the number and types of inspections required over the lifetime of the plant.

Section III, Nuclear Vessels, of the ASME Boiler and Pressure Vessel Code is the oldest of the four codes applicable to nuclear valves and the most firmly established from an enforcement viewpoint. It also set forth the design criteria and rigorous quality requirements deemed suitable for nuclear power plant application. Both the ASME Code for Pumps and Valves for Nuclear Power and the ANSI B31.7 Code for Nuclear Power Piping closely parallel the requirements set forth in the earlier document.

3.1.2 Standards Appropriate to Nuclear Valves

There are many industry-accepted standards and specifications that are appropriate to nuclear valves. The American Society for Testing and Materials (ASTM) is the most important source of specifications related to materials and their inspection. The ASTM specifications related to the testing and inspection of materials are designed to provide the desired material quality and have won wide-spread acceptance by industry. Many materials and alloys applicable to valve components are described in these specifications. The selection and understanding of a suitable specification is greatly simplified in that the materials are classified by their end use and service conditions.

The American National Standards Institute (ANSI) issues a large number of standards related to the standardization of dimensions for components and fittings used in the piping industry. Of particular relevance to valves is the ANSI B16.5 Steel Pipe Flanges and Flanged Fittings (including Reference to Valves), which has become the primary standard for steel valves sold in the United States of America. The ANSI B16.5 standard will in all likelihood remain the primary standard for valves intended for conventional (non-nuclear) power piping. The

ANSI B16.5 standard controls steel valve design by prescribing

1. minimum wall thickness,
2. dimensions of flanges and bolting for flanged-end valves,
3. dimensions of welding ends for welding-end valves (by reference to ANSI B16.25, Butt-Welding Ends),
4. face-to-face or end-to-end dimensions (by reference to ANSI B16.10, Face-to-Face and End-to-End Dimensions of Ferrous Valves),
5. pressure-temperature ratings, and
6. minimum hydrostatic test pressure for valve body (not a seat test).

Valves furnished under ANSI B16.5 are classified under seven pressure-temperature ratings of 150, 300, 400, 600, 900, 1500, and 2500 lb. Tables in this standard give the required pressure-temperature rating for any combination of design pressure and temperature for flanges and valves made of 17 different steel materials. These rating tables are identical to those incorporated in the ASME Code for Pumps and Valves for Nuclear Power. The tables in the ANSI Standard B16.5 also define for each rating the minimum hydrostatic test pressure for the valve body. Hydrostatic pressure testing of the seat and bridgewall is not required by ANSI Standard B16.5, nor does this document specify any leakage test for the valve seat. Minimum wall thicknesses for valves ranging from 1/2- to 24-in. nominal pipe size are listed in this standard with the caution that "Additional metal thickness needed for assembly stresses, valve closing stresses, shapes other than circular, and stress concentrations must be determined by individual manufacturers since these factors vary widely." It is difficult to compare the minimum wall thickness requirements of the ANSI B16.5 Standard with those of the ASME Code for Pumps and Valves for Nuclear Power because the two tables are based upon different valve dimensions. It should be noted that both the ANSI Standard B16.5 and the ASME Code for Pumps and Valves for Nuclear Power refer to the previously mentioned ANSI standards for flange dimensions, welding-end dimensions, and valve face-to-face or end-to-end dimensions. The relations of the various dimensional standards to a nuclear valve are illustrated in Fig. 3.2.

The Manufacturers Standardization Society (MSS) of the Valve and Fittings Industry issues standards related to both dimensional standardization and inspection testing of valves and fittings. The relation of

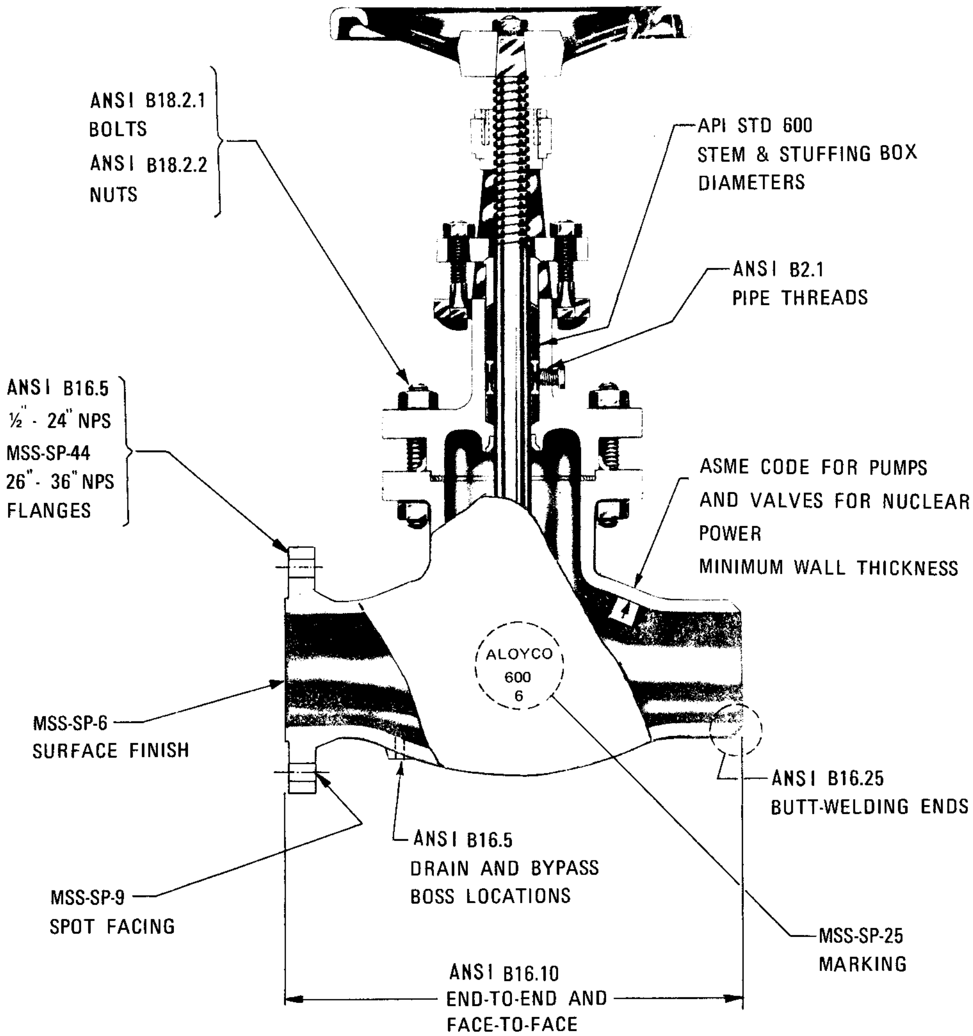


Fig. 3.2. Relation of Various Dimensional Standards to a Nuclear Valve.

these dimensional standards to nuclear valves is also illustrated in Fig. 3.2. It is interesting to note that one of these standards, MSS-SP-66, Pressure-Temperature Ratings for Steel Butt-Welding End Valves, provides an alternate method for rating steel butt-welding end valves by describing an alternate way of calculating the minimum wall thickness for steel valves. This method is often used for conventional (non-nuclear) power plant valves made with butt-welding ends. Another of these standards, MSS-SP-61, Hydrostatic Testing of Steel Valves, has become industry accepted for the hydrostatic testing of valves. It specifies a hydrostatic test of the valve shell at 1.5 times the design pressure adjusted to the hydrostatic test temperatures with a further requirement that the stem packing not show visible leakage during the hydrostatic test. The MSS-SP-61 standard also specifies a seat leakage test with ambient water at design pressure adjusted to the test temperature plus an optional seat leakage test with air at a pressure of 80 psig. Both of these seat leakage tests permit limited leakage across the seat where the permissible leakage is specified as a function of the nominal valve size.

The American Petroleum Institute (API) Standard 600, Flanged and Butt-Welding-End Steel Gate and Plug Valves for Refinery Service, is significant because in incorporates more detailed requirements for the design of the valve than the ANSI and MSS-SP standards. This standard specifies the diameters of the stem and stuffing box of valves. The API Standard 600 also provides pressure-temperature ratings for valves that are almost identical to the ratings prescribed by ANSI B16.5, and the hydrostatic test pressures prescribed are almost identical to those given in ANSI B16.5 for the respective pressure-temperature ratings. The API Standard 600 also specifies a seat leakage test with air at a pressure of 80 psig and with ambient water at design pressure adjusted to the test temperature with the requirement that the seat show no signs of leakage during these tests.

3.2 Flow Design

Those features of valve design that affect the flow characteristics of valves and things which may be done to optimize these features from a flow standpoint are discussed in this section. As a corollary, the possible deleterious effects on valves resulting from not giving due weight to flow considerations are included. As a preliminary to the above coverage, the objectives of flow design are outlined, the flow characteristics of the several types of valves are described, and some of the fluid flow theory used in analyzing flow characteristics of valves is presented.

The greatest applicability of this coverage of valve flow will be to globe valves. When operating in the full-open position, globe valves represent a larger resistance to fluid flow than either gate or swing check valves. When the fluid flow rate must be regulated, the globe valve design is the most relied upon for flow control. However, even the basic globe design must be modified with special designs to achieve any degree of good flow control. Gate valves cannot be recommended for any throttling or flow control other than very light-duty flow control. The principles of enhancing pressure recovery and minimizing erosion tendencies as described in the ensuing pages are equally applicable to the gate and swing check valves although these valves may not be referred to in the discussion.

3.2.1 Objectives of Flow Design

There are three general objectives to be attained in flow design. First, the valve should be designed with as low a flow resistance as practicable. Second, the valve should be designed to minimize erosion and/or cavitation problems. Third, the valve should be designed so that there is no excessive vibration or instability during its operation.

The reduction of flow resistance is a major objective in the design of valves. Valves often represent a significant fraction of the total flow loss in a process system, and reductions in the flow resistance of these valves can result in worthwhile savings in pump costs and in the operating cost of the process system. Further economy will be realized if the flow resistance can be reduced to a value which will allow the use of pipe with a smaller diameter.

The major savings in flow reduction are achieved by selecting the most appropriate valve for each application. For example, the flow resistance of the gate valve will be much less than that of a globe valve. In this case, the requirements for throttling and/or long life may override the use of the gate valve, but it would be the preferred valve from a flow reduction standpoint. If the throttling and long life requirements are overriding, the flow resistance may still be reduced by selecting an angle valve, a Y-body valve, or an offset valve, in order of ascending resistance.

Once the valve type has been selected, the flow resistance with the disc wide open is fixed within fairly broad limits. Obviously, careful attention to the internal arrangement and flow pattern of the valve can reduce this flow resistance significantly within the broad limits characteristic of that valve type. The various parameters and how they may be adjusted to reduce the flow resistance will be discussed in later portions of this section.

The next important objective of flow design is that of reducing the erosive effects of fluid flow that result from the impingement of the fluid on the valve surfaces. The surfaces affected are both the body and the trim. The erosive effect increases with an increase in velocity. As discussed in Section 3.3, high-velocity flow can erode away the valve base material or an oxide film that has formed.

Erosion is often accompanied by cavitation when the liquid being throttled undergoes a large pressure drop and the pressure drops to the vapor pressure of the liquid being pumped. The liquid will then flash into the vapor phase momentarily and will collapse back to the liquid phase as pressure recovery occurs downstream of the throttling location. Although interrelated and often occurring simultaneously, erosion and cavitation are distinct effects. Erosion almost always occurs with cavitation, but cavitation only occurs in the cases where the subcooled liquid flashes to vapor and subsequently collapses back to liquid again.

Erosion can be general or localized. A general surface erosion may not be obvious to visual inspection. Localized erosion is rather easily detectable and is the more dangerous because it will lead to leakage earlier. In a system which has high flow velocities, both general and localized erosion may occur. General erosion tends to predominate in systems carrying steam with a high percentage of moisture carryover, and localized erosion tends to predominate in systems carrying subcooled liquid where whorls and vortices occur downstream of the throttling region. Cavitation tends to cause general erosion downstream of flow restrictions and valve seating surfaces where the vena contracta pressure has dropped to the vapor pressure of the subcooled liquid. Cavitation can be quite destructive and should be eliminated.

It should be noted that no matter how well valves may be designed from the standpoint of flow, cavitation and erosion can be produced simply by installing the valve in a severe flow environment. The valve designer cannot guarantee that the valve design alone will eliminate erosion and cavitation. Rather, the emphasis should be placed on providing design features in valves that will optimize flow characteristics.

The third objective of valve flow design is to minimize vibration and instability. Vibration and instability are most likely to occur near the valve-closed position where the disc has its maximum flow stream effect and the flow velocity is maximized by the low flow area. When throttling, a large percentage of the pressure energy is being converted to kinetic energy, and if not properly designed, the valve can absorb some of this energy through disc vibration or rotation. The effects of throttling could be severe, and an objectionable side effect is a very high noise level. A difficult aspect of vibration is its unpredictable nature. It is an instability which arises out of flow effects in a complicated disc-seat geometry.

There are other secondary considerations which are desirable characteristics in specialized applications. Throttle valves should have good throttling linearity. This means that for each increment of stem travel there should be a corresponding percentage increment of flow increase. Most globe valves not designed for throttling have a disproportionate amount of flow increase in the first increment(s) of stem travel from the closed position. Linearity is desirable in a control system. For many applications, it may not be necessary to achieve strict linearity but only to reduce the excessively nonlinear characteristics of the standard globe valve. It should be noted that the disc-seat geometries which improve throttling linearity normally increase the fully-open resistance coefficient of the valve. Thus, valves with good throttling characteristics normally have high resistance coefficients when wide open.

Check valves should be designed to minimize water hammer effects when sudden flow reversals occur. In both plug and swing check valves, sudden valve closure may initiate vibration and pressure surges. This "slam" can be reduced by valve design. In swing check valves, it is desirable to eliminate disc movement while in the open position during normal system operation. The movement that may occur is a "fluttering" or "chattering" of the disc against its stop if the flow is insufficient to keep the disc firmly against its stop. Such cyclic motion of the disc can lead to advanced wear of disc bearings.

3.2.2 Flow Resistance Characteristics of Valves

The flow resistance of a valve may be designated in several ways. The three common measurement criteria are the resistance coefficient (K), the flow coefficient (C_v), and a measure known as "equivalent length in pipe diameters" (L/d). These flow measurement parameters with their formulas[2] and principal applications are described as follows.

The resistance coefficient K is the resistance coefficient associated with each particular valve, and it is a measure of the reduction in static head expressed in terms of velocity head loss of the fluid in passing through the valve. Thus, K is a dimensional constant in velocity head loss formulas that indicates the relative measure of flow resistance.

$$K = \frac{h_L}{\frac{V^2}{2g}}$$

or

$$K = f(L/d)$$

where

h_L = loss of static pressure head caused by fluid flow, feet of fluid,

V = mean velocity of flow, ft/sec,

g = acceleration of gravity = 32.2 ft/sec·sec,

f = friction factor, and

(L/d) = equivalent length in pipe diameters.

The flow coefficient C_v is a relative measure of the flow conductance of a valve. It is defined as the flow rate in gallons per minute of water at a temperature of 60°F and at a pressure drop of 1.0 psi across the valve.

$$C_v = Q\left(\frac{\rho}{62.4\Delta P}\right)^{1/2}$$

or

$$C_v = \frac{29.9d^2}{(K)^{1/2}}$$

where

Q = rate of flow, gal/min,

ρ = weight density of fluid, lb/in.^{3}, and

ΔP = pressure change, psi.

The "equivalent length in pipe diameters" (L/d) is defined as the length of pipe which, if substituted for the valve, would provide an equivalent amount of flow resistance. This parameter is generally expressed in non-dimensional form by defining the pipe length in terms of pipe diameters. This definition of flow resistance is not as accurate as the other parameters but is frequently used to provide an easily visualized method of describing flow resistance. Some valve vendors use it to express the flow resistance of a model or class of valves because it is approximately constant over the size range for the same design.

$$(L/d) = \frac{h_L}{\frac{fV^2}{2g}} \quad .$$

(a) Gate Valves. Properly applied, a gate valve is used either shut or full open. In the full-open position, the gate valve offers little resistance to flow. On page A-30 of Ref. 2, the statement is made that the flow resistance for a conventional gate valve will be approximately equal to a length of pipe 13 times the diameter of that pipe. For example, for a pipe with a diameter of 6 in., the corresponding gate valve would cause the same amount of flow resistance as a 6.5-ft length of the pipe. This amount of flow resistance represents only a small part of the total flow resistance of most process system loops. The low flow resistance characteristic of the gate valve is a major advantage of this type of valve.

(b) Globe Valves. In comparison with a gate valve, a globe valve offers a very high flow resistance even in the full-open position. The following values for the flow resistance, stated in equivalent length in pipe diameters, of several types of globe valves in the full-open position are given in Ref. 2.

Valve Type	Equivalent Length in Pipe Diameters
Conventional in-line globe	340
Conventional in-line globe with guided disc	450
Conventional angle	145
Conventional angle with guided disc	200

Y-pattern globe with 60° stem	175
Y-pattern globe with 45° stem	145

Globe valves provide from 11 to 26 times as much flow resistance as is provided by gate valves. Thus, the flow resistance of one globe valve frequently represents a significant part of the total flow resistance of a process loop, and efforts to reduce the flow resistance in a globe valve often benefit the design of the process loop. In minimizing the loop flow resistance or pumping power, the first consideration should be directed toward selection of the valve. The angle valve represents the smallest flow resistance if it can be located in an elbow position. If the valve must remain at an in-line section of piping, the use of a Y-pattern globe valve will offer a significant reduction in flow resistance. The guided-disc design given in the preceding list refers to the use of fluted guide extensions or pin guide extensions below the disc. These obviously increase the flow resistance of the valve when they are required.

(c) Check Valves. Because of their wide differences in geometry, check valves have widely differing flow resistances. The following values for the flow resistance, stated in equivalent length in pipe diameters, of several types of check valves in the full-open position are given in Ref. 2.

Valve Type	Equivalent Length in Pipe Diameters
Conventional swing check	135
Globe lift check or stop-check	340
Globe lift check with guided disc	450
Angle lift check or stop-check	145
Angle lift check with guided disc	200

If minimizing the flow resistance is important, prime consideration should be given to selecting the proper type of check valve. The swing check valves offer less flow resistance, but the lift-check valves generally have tighter sealing characteristics. The lift check valves are substantially equivalent to their respective body styles in globe valves, and use of guided discs significantly raises the valve flow resistance.

The inherent flow characteristics of the check valve may present a special problem in a process system which has a "low-flow" operational mode. Functioning of the check valve depends upon the energy of the flowing fluid to hold the disc open. Most check valves are designed to hold the disc full open even when being operated with only a partial flow. In the turbulent flow range, the pressure drop or head loss through the valve will be roughly proportional to the square of the fluid velocity. This relationship of pressure drop to velocity of fluid is valid for check valves only if there is sufficient flow to hold the disc in a wide open position. As the disc closes, the valve flow resistance coefficient (K) rises very rapidly and it becomes very difficult to predict the relationship between valve pressure drop and fluid flow. Extended operation of the check valve in the partially open condition is highly undesirable anyway because this results in noisy operation and premature wear of the moving parts. In specifying a check valve, it is preferable to undersize the valve rather than let it see much service in the partially open condition.

3.2.3 Flow Theory

Valve flow, per se, is not amenable to strict mathematical analysis. Thus, the solution of many such problems is at least partially dependent upon experimentally determined coefficients. When using empirical formulas, one must be sure that the conditions of the problem approximate the conditions of the experiments from which the formulas were obtained.

Complicated effects along with some of the techniques useful in correcting them can be described without analysis. The valve flow theory discussed here involves consideration of the effects of sudden change in direction, sudden contraction, and sudden expansion of incompressible fluids. These effects are then combined to make up an approximate description of flow through valves. In addition, some special flow topics, such as compressibility effects, laminar flow effects, instabilities of the vena contracta, sonic effects, and two-phase flow effects, are included. Valve flow is also influenced by viscosity effects, but viscosity is not important for a nuclear power plant where the fluid is normally either high-pressure steam or water.

It will be more clearly pointed out later that the treatment for incompressible flow through a valve generally holds for compressible fluids such as steam as long as the flow velocity of the steam is small when compared with the velocity of sound. This is generally true in power plant steam systems. Elementary flow theory for incompressible flow through curved pipes, sudden contractions, and sudden expansions is treated in many textbooks on the subject. The following excerpt on fluid flow in curved pipes and channels from one such text[3] is indicative of the treatment of elementary flow considerations.

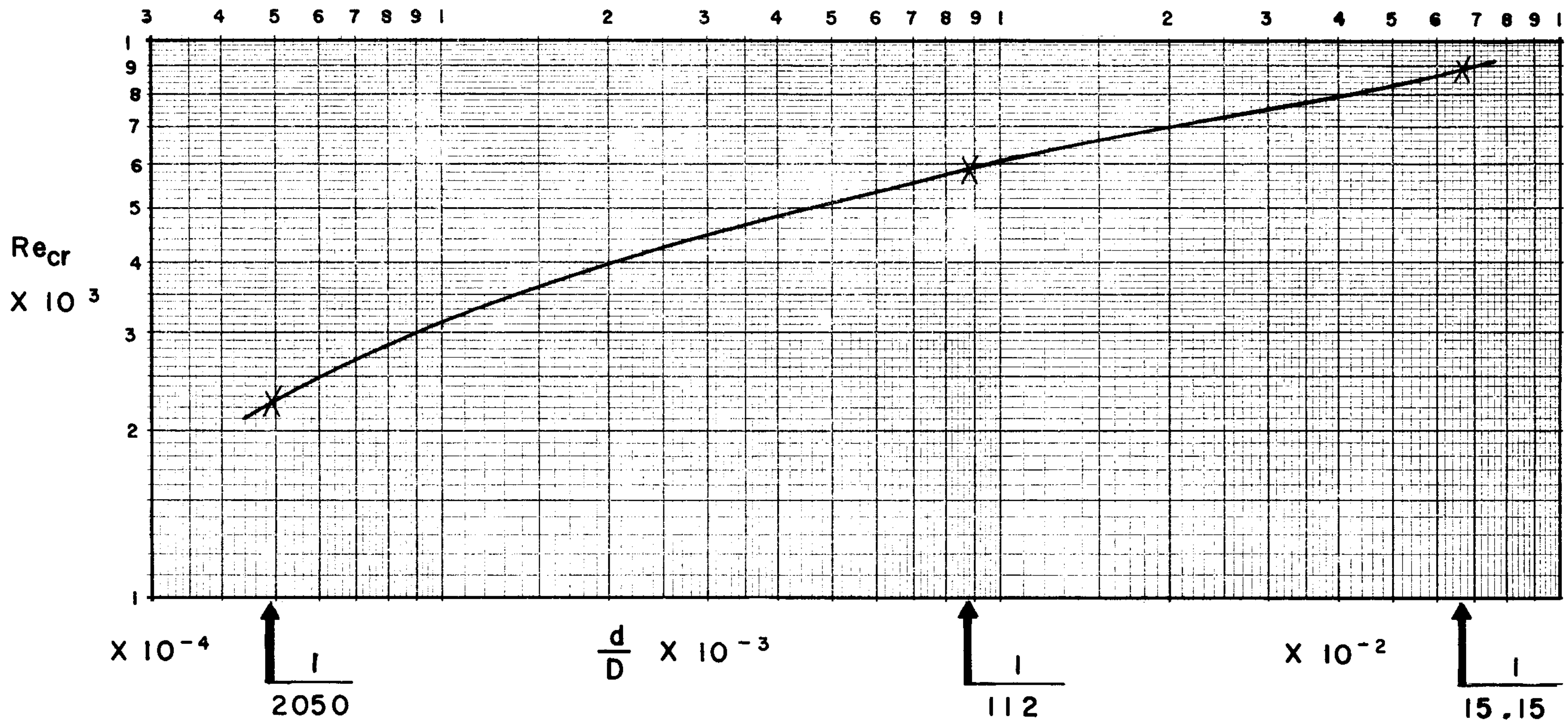
> "Let us first consider the flow pattern in a curved channel. During flow along a curved trajectory, centrifugal forces, proportional to the square of the velocity, act on the fluid particles. Due to frictional resistance, particles near the walls have lower velocities. Particles in the central part of the fluid, which have higher energy, displace particles located at the outer wall of a curved pipe and force them to move toward the inner wall of the bend. The displacement of particles produced by the centrifugal forces causes transverse motion. This secondary flow, superimposed on the main flow parallel to the channel axis, gives the streamlines a helical shape which is maintained in the straight portion of a pipe with a bend in it until it essentially disappears 10 to 15 diameters beyond the bend.
>
> "The flow pattern and the frictional resistance in a curved channel are determined by three things: the formation of an eddy region near the inner wall of the bend, the formation of a similar region near the outer wall, and the generation of a vortex pair in the channel cross section. These phenomena determine the resistance of a curved channel and the deformation of the velocity field throughout the entire straight portion beyond the bend.
>
> "The energy loss during fluid flow in curved channels depends on the Reynolds number, the relative roughness of the walls, the angle of the bend, the relative radius of curvature R/d (where d is the pipe diameter and R is the radius of curvature of the bend), the ratio of entrance area to exit area, and on the pipe shape."[3]

The text[3] goes on to point out that a phenomenon noticed in curved flow is that the critical Reynolds number increases. The critical Reynolds number, which indicates the transition from laminar flow to turbulent flow, is normally about 2000 but increases to 9000 for a curve flow diameter to a radius-of-curvature ratio of 1 to 8. This means that a higher curvature increases the stability of the laminar flow regime, as is illustrated in Fig. 3.3.

Further details pointed out[3] include the fact that for low Reynolds numbers, the maximum velocity streamline is displaced toward the outer wall of the bend and as the Reynolds number increases the maximum velocity streamline displaces toward the inner wall of the bend. The fluid pressure along the outer wall of the bend first increases and then decreases. Along the inner wall of the bend, it first decreases and then increases. The roughness of the wall of the bend has a large effect on flow resistance. This is especially true at higher Reynolds numbers where the maximum flow velocity tends to displace toward the inner wall of the bend. The short distance between the maximum velocity streamline and the wall of the pipe bend tends to amplify the effect of wall roughness. This is why in valves which have been cast, it is important to minimize the as-cast surface roughness of the internal flow passages.

The overall flow resistance of a pipe bend also depends on the angle of the bend. For a given radius of curvature, the flow resistance increases approximately proportionately with the angle of bend up to about 20°. Beyond this, the increase in flow resistance remains proportional only for gradual, smooth bends where the ratios of radius curvature to flow diameter are about 2 to 4. At lower ratios (sharper bends), the increase is greater. For instance, at an R/d ratio of 1, the flow resistance at 40° may be three times that at 20°. Conversely, for higher ratios, the flow resistance increases at a rate less than proportionately. For shallower bends, the maximum flow velocity is not displaced far from the center line of the pipe (pp. 154-156 in Ref. 3).

There are several ways that flow resistance can be minimized in bends. Basically, these measures are an attempt to minimize the displacement of the maximum flow velocity from the center line of the pipe. One such technique is to provide concentric vanes inside the bend so that the flow is captured and cannot displace radially. Another technique is to follow one bend immediately with another in another direction. The flow resistance in this case would be less than that of having two equal bends separated by some straight pipe.



$\frac{d}{D} = \frac{1}{2050}$ ..... Re_{cr} 2250

$= \frac{1}{112}$ ...... Re_{cr} 6000

$= \frac{1}{15.15}$ ...... Re_{cr} 9000

D = DIAMETER OF CURVATURE = 2 TIMES RADIUS OF CURVATURE (SAME UNIT AS d).

d = PIPE DIAMETER (ANY UNITS)

Fig. 3.3. Critical Reynolds Number as a Function of d/D.

For a sharp bend, as opposed to a gradual bend, it has been found by correlation that the total flow resistance is approximately proportional to

$$\frac{5}{2\sin^2\frac{\delta}{2}},$$

where δ = angle of the sharp flow bend. In general, this means that in comparison with the smooth bend, the sharp bend offers more flow resistance per angle of bend and the rate of increase of flow resistance with bend angle is higher. The flow resistance of a 30° sharp bend is about the same as the flow resistance of a 90° smooth bend for which the R/d ratios are greater than two. Beyond 30°, the flow resistance of a sharp bend increases quite rapidly. In forged globe valves where there has been an attempt to reduce the end-to-end dimensions, the angles of the sharp bends in the valves are often greater than 30°.

Flow energy loss in valves is caused by sudden widening effects. Sudden widening occurs when flow from a small-diameter flow path enters a large-diameter flow path with little or no transition zone. At a sudden widening in the flow path, the stream expands suddenly, a jet is formed, and the jet is separated from the remaining part of the slower moving liquid by an eddy surface. The eddies promote mixing and turbulence and absorb energy from the flow stream. This energy loss can be computed from straightforward momentum considerations. The resulting equation, called the Borda-Carnot theorem, can be expressed as follows.

$$\Delta E = \frac{1}{2}(V_1 - V_2)^2,$$

where

ΔE = energy loss with changes in fluid velocity and

V = mean velocity of flow, ft/sec.

This theorem stated that the energy loss is equal to one-half the square of the velocity loss in going through the sudden widening. The energy loss is then directly convertible to head loss per unit density.

Because of pipe friction losses not considered in the preceding analysis, the above equation can be expressed as follows.

$$\Delta E = \frac{K}{2}(V_1 - V_2)^2,$$

where K is normally very close to unity. Pressure losses caused by sudden enlargements can be quite high. Minimizing such losses requires either that the size of the enlargement be reduced or that the enlargement be made gradual. The amount of saving in head loss that results when the enlargement is made gradual is illustrated in Fig. 3.4.

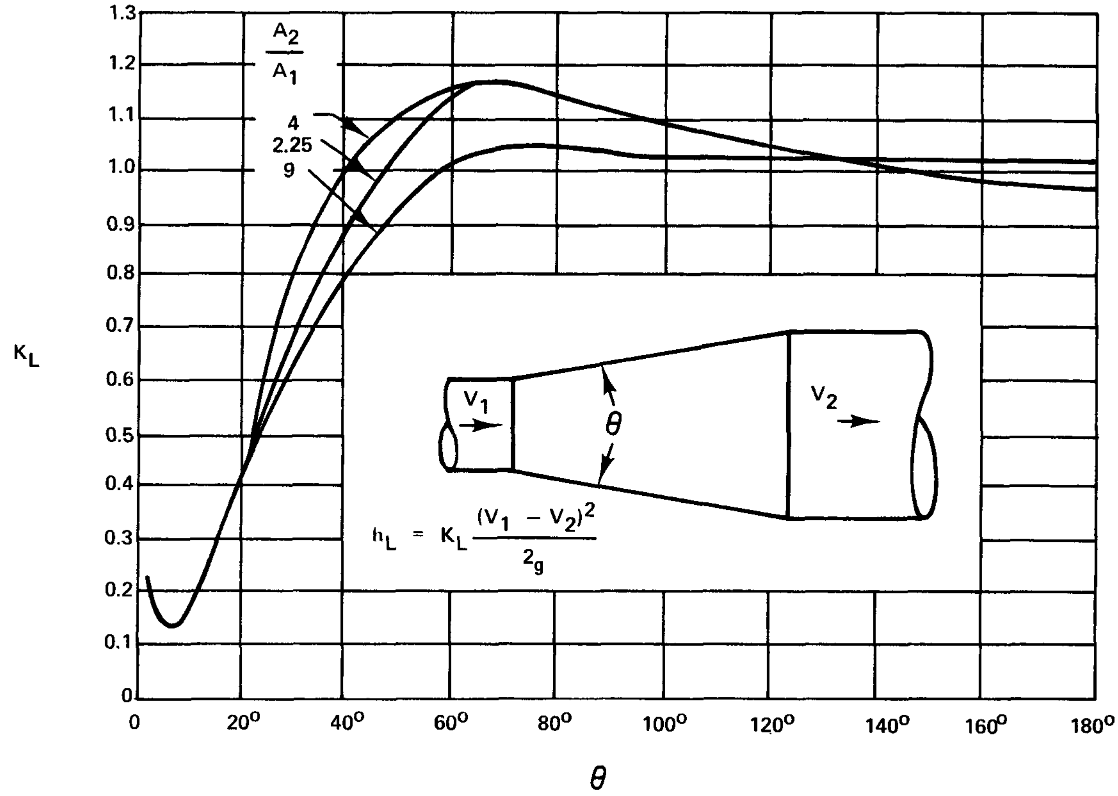
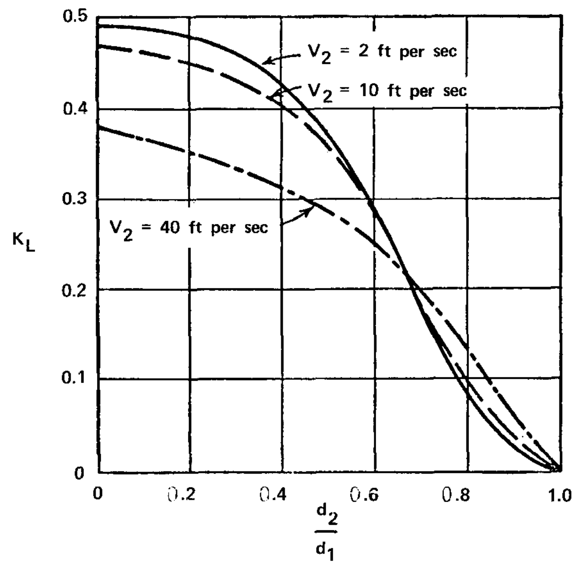


Fig. 3.4. Loss Coefficients for Conical Enlargements (from Ref. 4).

The gradual enlargement would take the form of a cone-shaped piece between the smaller flow area and the larger flow area. As indicated in Fig. 3.4, the angle is the included angle of the cone. It may be seen that starting with the zero-included angle, the loss first decreases until about 7°. This indicates that the effects caused by pipe friction of the cone section are reduced at first, reducing the overall energy loss. It should be noted that if the cone-shaped piece is substituted for equivalent length of pipe, such a savings will not be realized. Further, it is better to use a sudden enlargement than one of a cone angle of around 60° since K_L is smaller for the former. It is not clear why this is the case, but it can be conjectured that at 60° there is a small vena contracta formed at the sudden flow boundary change entering the cone. The vena contracta can aid in restricting flow at angles of about 60°, but this is not the case at greater angles. The graph illustrated in Fig. 3.4 indicates clearly that in most cases, the enlargement must be very gradual with a cone angle of less than 50° for any effect whatsoever. This may not be achievable in valves where flow channel space is limited. Emphasis should therefore be directed toward eliminating flow area changes.

The energy loss caused by sudden narrowing is interesting not so much from the standpoint of energy loss as from the standpoint of an interesting phenomenon called the vena contracta, which is the contraction of the free jet of liquid passing through an orifice. The energy loss caused by sudden narrowing is always less than the energy loss caused by sudden enlargement of an equal amount. However, it is important to note that losses can be high at restrictions because any narrowing results in an energy loss at a subsequent enlargement back to the original diameter. The loss factor versus the ratio of area contraction is illustrated in Fig. 3.5. It is given in terms of $V_2^2/2g$. As may be



d_1 = inlet diameter
d_2 = exit diameter
V_2 = exit fluid velocity

Fig. 3.5. Loss Coefficients for Sudden Contractions (from Ref. 4).

seen, the loss is greatest when the ratio of exit diameter over inlet diameter is smallest. However, in no case does its value exceed 0.5. The loss decreases to zero as the amount of contraction diminishes. Fluid flow velocity has some effect on this flow resistance, as is indicated in Fig. 3.5.

The loss mechanism in fluid contraction is the same as in sudden fluid expansion. It is a jet effect with separation between a high-velocity center flow and the flow boundary. As the fluid contracts in entering the reduced flow area, it is subject to momentum which causes the minimum flow area to occur not at the entrance to the reduced flow area but a little farther beyond, as is illustrated in Fig. 3.6. Thus, the flow area at this point is actually less than the flow area of the reduced area section. This phenomenon, the vena contracta, is responsible for the losses suffered by the flow in a sudden contraction. Up to the vena contracta, the losses consist of pipe friction losses only.

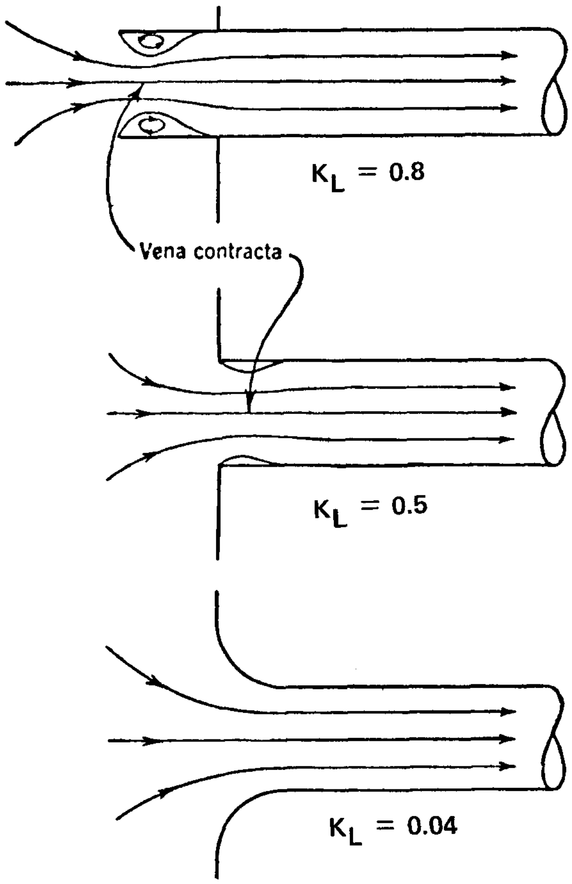


Fig. 3.6. Loss Coefficients for Pipe Entrances (from Ref. 4).

Beyond the vena contracta, the loss is due to the same effect as losses in sudden enlargements since expansion occurs. This is brought about by the generation of eddies and turbulences as the central jet of the flow causes flow shear with the slower moving fluid closer to the boundary.

The losses caused by sudden enlargements and sudden contractions are compared in Fig. 3.7. The resistance coefficient has been standardized and compared. Standardization of the resistance coefficient consisted of basing this coefficient on the fluid velocity in the smaller pipe. It may be seen that losses caused by sudden enlargements are roughly twice those caused by sudden contractions.

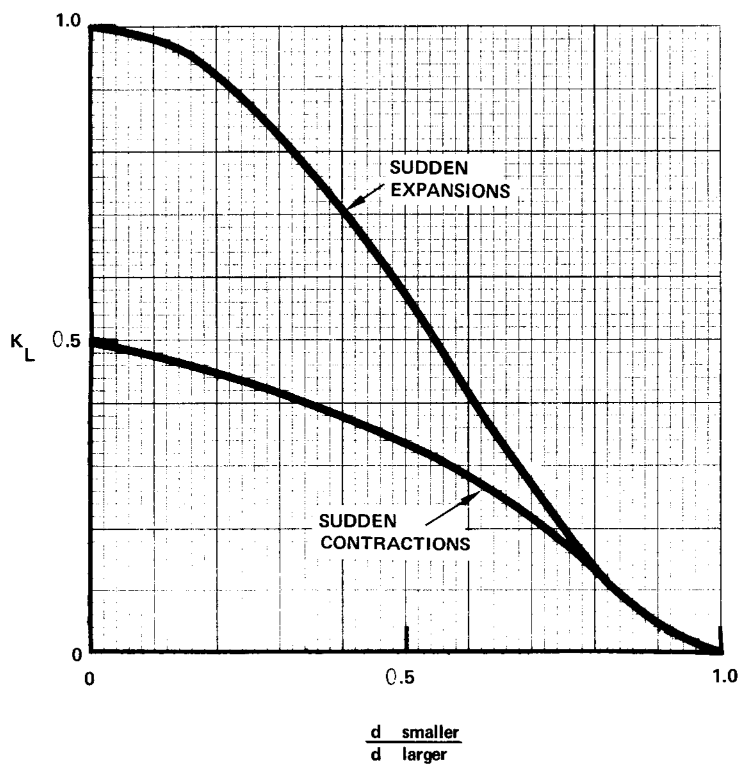


Fig. 3.7. Flow Resistance Caused by Sudden Expansion Compared With That Caused by Sudden Contraction (from Ref. 2).

Losses caused by sudden contractions are easier to eliminate or reduce than those caused by sudden expansions. If the sharpness of the edge at the entrance is reduced even by a small amount, the vena contracta is largely eliminated. On the other hand, if the sharpness of the entrance is accentuated, the loss coefficient can increase above 0.5.

It will be seen later that in terms of valve design, the vena contracta can play a large part in valve flow resistance through the seat and disc area. Suffice it to say here that for reduced flow resistance, the vena contracta should be reduced by using a smooth-flow inlet design. Further, instabilities in flow resistance can be the result of an unstable vena contracta, which may exhibit cyclic motion. Instabilities in the vena contracta can be reduced by providing a well-defined edge to "anchor" it. Thus, it can be seen that a smooth entrance to a reduced flow rate area is not the complete story. If a vena contracta forms at higher flow velocities, such an area may suffer the disadvantage of an unstable vena contracta since there is no well-defined edge to "anchor" it.

The flow resistance brought about by the location of an orifice in the flow path is simply a matter of combining loss caused by sudden contraction and the loss caused by sudden enlargement. There are some deviations to this general rule brought about primarily by the configuration of the edge of the orifice, as is illustrated in Fig. 3.8. The degree and type of tapering provided can increase or decrease flow resistance. In valves, flow constrictions that are similar in geometry are provided by the seat bridgewall. If the diameter of the seat is substantially less than the diameter of the pipe in the system, flow resistance arising from orifice-type flow losses will result.

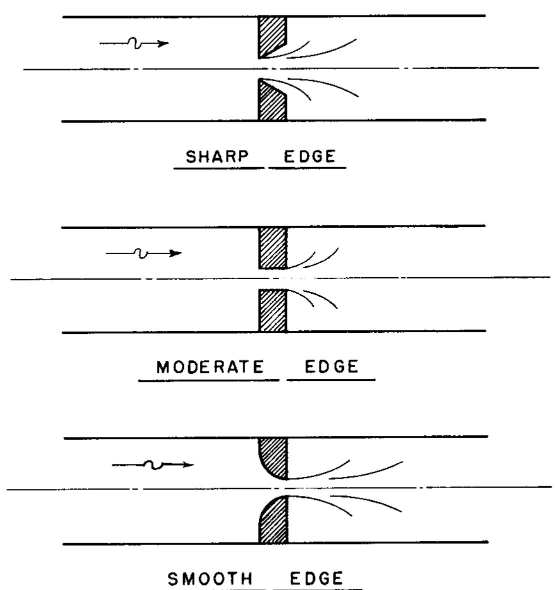


Fig. 3.8. Orifice Edge Configurations.

As pointed out earlier in this section, other aspects of fluid flow as they pertain to valves must be covered. These include laminar flow, fluid compressibility, two-phase flow, vena contracta dynamics, sonic effects, vapor effects, and determination of hydraulic diameter.

Laminar flow results in an increase in valve flow resistance over turbulent flow resistance. It is unusual for laminar flow to occur in piping systems for power plants. Most systems are designed for turbulent flow although laminar flow must be considered in the system design, including pipes and pumps, if such flow is to occur in a system. Laminar flow considerations see much more application in systems where the fluid is more viscous than water. The only comment which need be made here from the standpoint of valves for water and steam systems is that where laminar flow occurs, the valve fluid flow resistance can be much higher than normally is the case because of the increased friction effect of wall surfaces.

The discussion to this point has concerned incompressible fluids. Since steam is compressible, some comment must be made on deviations from incompressible theory. Fluid flow under compressible fluid conditions has been well analyzed, and much information on the subject[2,3,5] is available. However, compressible flow theory is not required here since it is applicable to flow which has velocities approaching the speed of sound (Mach 1). Compressible fluid flowing at velocities much less than Mach 1 exhibit behavior very close to incompressible flow. It should be carefully noted that the flow considered should be the maximum flow velocity in the pipe sections. For instance, the flow of steam in a system where the globe valves are fully open will be fairly uniform throughout the system at velocities much less than the speed of sound. If a globe valve in the system is gradually closed, however, there will be a point reached where the throttling effect will increase the local velocity of the fluid to the speed of sound and there will be flow characteristic of compressible flow for a short length of flow path at this location.

Choking occurs in compressible flow. When the local flow velocities at the location of greatest flow constriction in the valve approach the speed of sound, either through valve closing or increase in pressure drop across the valve, the flow begins to "choke". This means that the flow and mass transfer through the valve reach a maximum value and cannot be increased further. It becomes meaningless at this point to talk about valve flow resistance since the limiting condition is the maximum flow rate of compressible fluid through the valve at the given conditions. Compressible sonic flow is not a desirable situation. Shock waves are produced at the valve and downstream where the flow again reverts to subsonic flow. The shock waves cause vibration of the system, and the high fluid flow velocities advance internal erosion of the system. It is usually unwise to locate a pipe bend or another valve immediately downstream of a valve where such flow can occur. When designing valves for such systems, it is important to look not only at valve flow resistance but at valve flow restrictions where higher than normal velocities can occur, and these should be carefully reviewed to assure that local flow velocities remain appreciably below Mach 1.

In subcooled water systems, two-phase (liquid and vapor) flow must be considered. If the local flow pressures at valve restrictions can fall below the vapor flash point, local two-phase flow will occur. This has several effects on valve operation. First, cavitation occurs. The damaging effects of cavitation have been mentioned previously. An unstable and fluctuating pressure occurs at the point of maximum flow restriction. Since the pressure is dependent upon the temperature of the fluid, slight variations in fluid temperature result in corresponding variations in fluid pressure at this point. Although pressure "bottoms out" at this point, as is illustrated in Fig. 3.9, the velocity increases because of

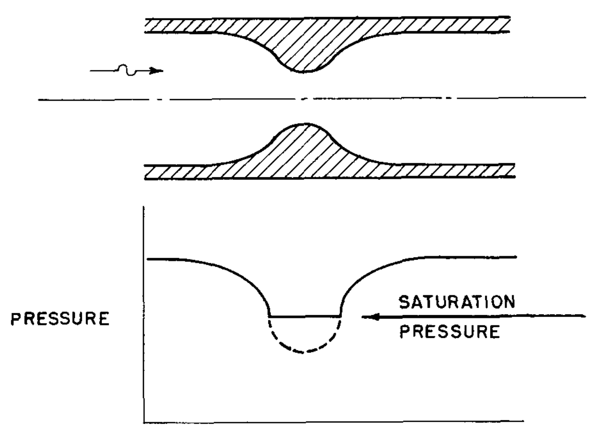


Fig. 3.9. Saturation Effects.

vapor formation. High flow velocities, approaching the speed of sound, can occur. Thus, the unlikely situation arises where flow conditions can be such that vapor formation and sonic effects can occur even in subcooled incompressible fluid systems. This is doubly undesirable in subcooled liquid systems since cavitation as well as supersonic highly erosive flow is involved. The feedwater systems in steam plants are locations where such effects can occur, and this possibility should be considered in the design.

A note should be made here about cavitation. In incompressible fluid systems where cavitation occurs because of vapor formation and restrictive flow, one can expect severe damage to fluid boundaries downstream of the valve. The local decrease in pressure to the flash point of the subcooled liquid at the valve flow restriction results in the generation of voids or vapor bubbles in the flowing fluid. These are then carried downstream, and as pressure recovery begins to take place, the fluid remains slightly supersaturated with steam because of momentum considerations. When the voids collapse downstream, the pressure at the cross section is a little higher than saturation pressure. This adds energy to the collapse of the voids that is then converted to small pressure pulses throughout the flowing stream. These pressure pulses destroy protective corrosion layers on fluid flow boundaries and help reduce their corrosion resistance.

Claims are made on the part of valve vendors to the effect that damaging effects from cavitation and supersonic flow can be reduced or minimized through the use of devices to separate flow. The theory behind such devices is to promote mixing of the flow downstream of the valve. Since flow energy disturbed by the valve must be converted to either erosion or increased fluid enthalpy, any device which can promote flow enthalpy at the expense of erosion damage is considered to be beneficial. A typical device of this type is illustrated in Fig. 3.10. It promotes mixing by separating the main high-velocity flow stream into a grid network of smaller flow streams separated by relatively slow moving liquid. Since the dimensions are about the size of the mixing length of the fluid, vigorous local mixing occurs that absorbs the energy of the high-velocity flow stream along with void energy.

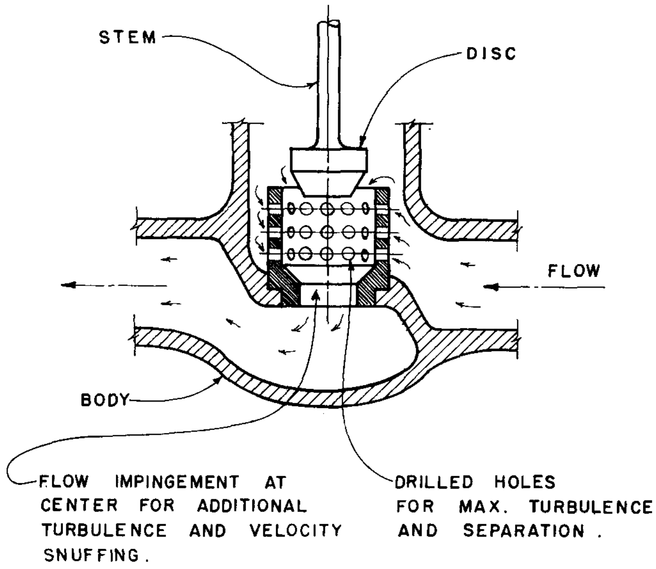


Fig. 3.10. Typical Cavitation Reducing Device.

It is sometimes true in valve design that the effect of flow area of the valve in any particular section is not clear. While the area of any section may be taken as the flow proceeds through the valve, the angle of orientation of the section may affect the effective flow area, as is illustrated in Fig. 3.11. The best rule of thumb in determining the effective flow area or the effective hydraulic diameter at any

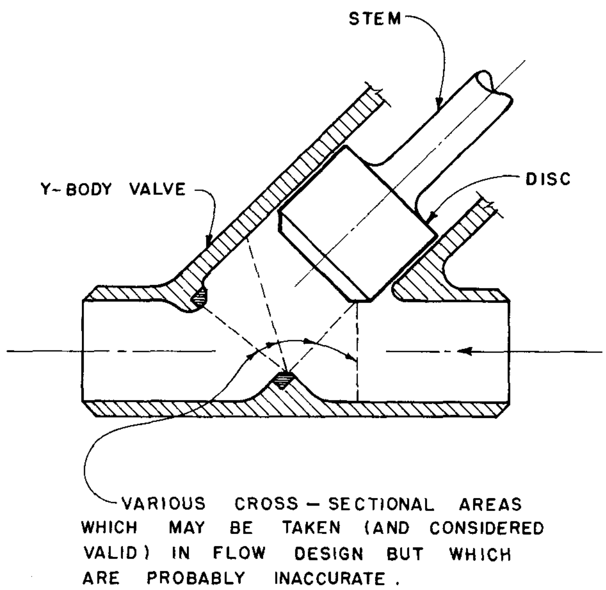


Fig. 3.11. Effective Hydraulic Diameter.

particular location is to find the highest velocity streamline within the valve at that point and take a section perpendicular to it. It can be seen at once that where bends are involved, the highest velocity streamline may follow different paths through the valve as the flow velocity is changed, as is illustrated in Fig. 3.12. This means that the effective valve hydraulic diameter may change and be dependent upon the valve flow rate.

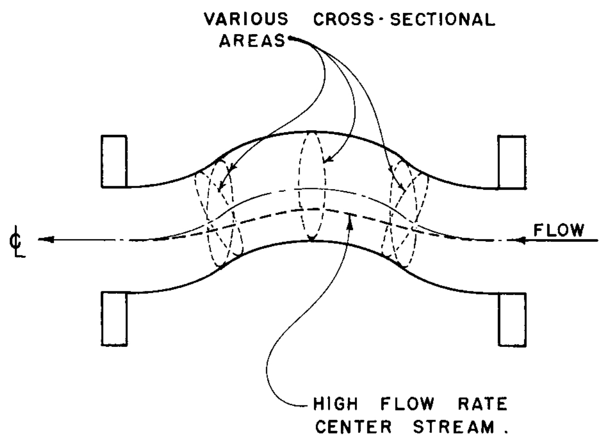


Fig. 3.12. Variation of Cross-Sectional Area Caused by Increase in Flow Velocity.

One further note may be made about the highest velocity streamline within the valve. Once the path of the highest velocity streamline through the valve is located, it is desirable to design valve details to be compatible. Various restrictions which are necessary for flow control purposes may be designed to reduce the overall flow resistance by arranging their geometry to be parallel to the path of this high-velocity streamline. This is also true for determining the angle of taper enlargements or contractions. How this is done is illustrated more clearly in Fig. 3.13.

The type of flow downstream of a valve differs from that upstream. Regardless of whether the valve is fully open or partially closed, there is a certain amount of flow velocity separation downstream. This means that the difference between the highest velocity and the lowest velocity is greater in this area than in the upstream area. The higher velocity usually occurs in the center of the flow stream. Throttling and valve

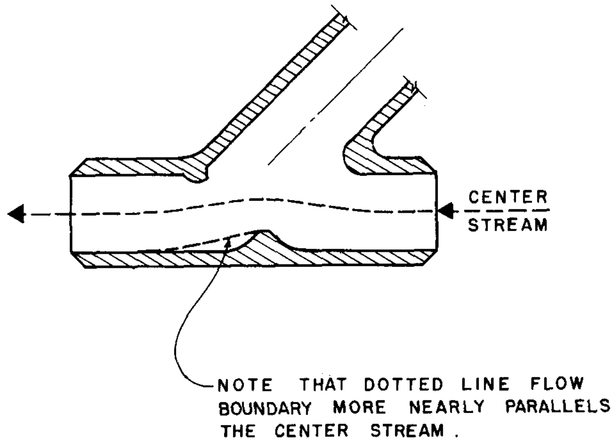


Fig. 3.13. Center Stream Considerations.

closure accentuate this effect. The high-velocity center stream of the flow can have an erosive effect. The erosive effect of this center stream at a flow bend is illustrated in Fig. 3.14. Since the center flow

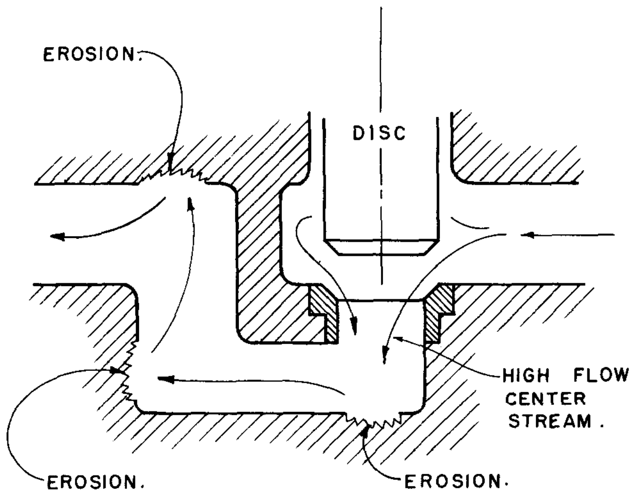


Fig. 3.14. Erosion Effects of High-Flow Center Stream.

stream has a higher momentum than other parts of the flow stream, its path will deviate to the walls of the valve body when passing through flow bends. The cross-sectional velocity profile of the flow stream should be allowed to reach equilibrium before introducing severe flow bends, as is illustrated in Fig. 3.15.

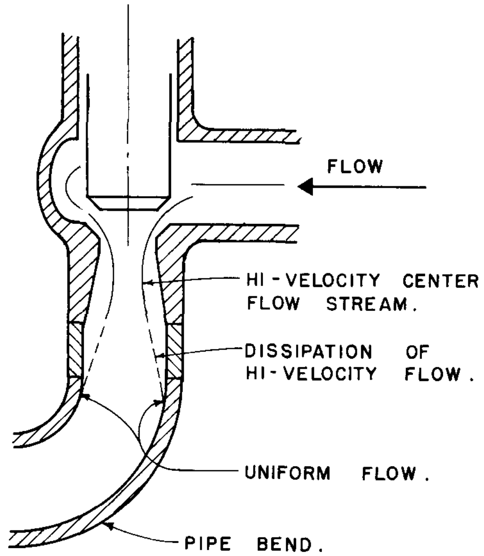


Fig. 3.15. Dissipation of High-Velocity Flow Downstream of a Valve.

To summarize the preceding discussion, the greatest effect in minimizing the valve flow problems of cavitation, high resistance, high turbulences, and erosion can be attained by reducing the number and sharpness of flow bends and by providing gradual flow-area changes.

3.2.4 Valve Design Techniques for Flow Control

As previously pointed out, there are several objectives to valve flow design, some or all of which may be important to a particular valve application. Flow design may reduce flow resistance, reduce cavitation or erosion, reduce valve vibration, increase throttling linearity, and provide high-pressure-drop flow-throttling capabilities. The various design techniques to achieve these flow qualities are discussed in this section.

(a) Reduction of Flow Resistance. The various techniques which can be applied to minimize valve flow resistance are illustrated in Fig. 3.16. These techniques all employ three basic principles: reduction of flow bends, rounding of inlet corners, and tapering of the outlet flow area.

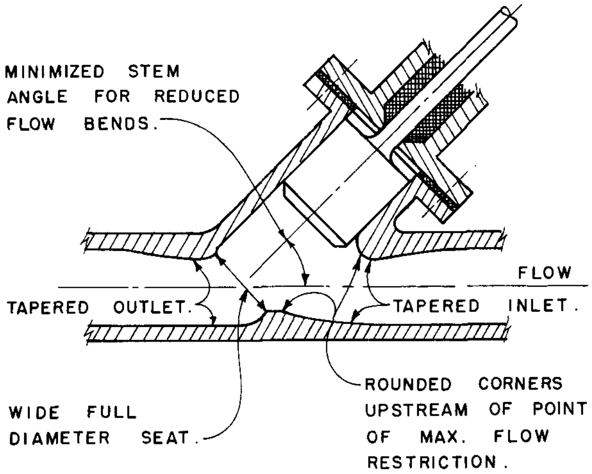


Fig. 3.16. Valve Designed for Minimum Flow Resistance.

When a globe valve is open, the largest share of the valve flow resistance is caused by the changes in direction of the fluid as it passes through the valve. As the disc approaches the seat during valve closing, valve flow resistance begins to increase because of the reduction in the flow cross-sectional area. As the flow decreases, the major share of valve flow resistance then appears across the seat of the valve. At this point, seat and disc geometries become important in influencing the way in which fluid is controlled in the gap between them.

In some globe valves, the flow changes direction within the valve body in 90° turns as many as six times as it passes through the valve. These often are sharp bends or, at best, radius bends with a small radius of curvature, as is illustrated in Fig. 3.17. However, it should be kept

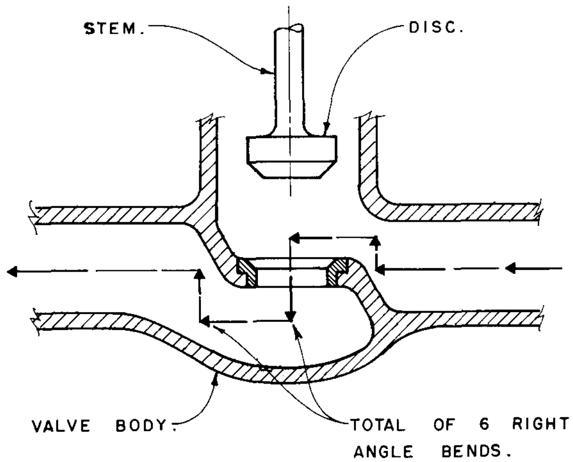


Fig. 3.17. Globe Valve Flow Bends.

in mind that since the bends are adjacent to each other, the total flow resistance will be somewhat less than the sum of all of the individual flow resistances caused by each individual bend. Globe valve flow bends can be reduced in both size and number. Typical techniques for reducing the number of bends in globe valves are illustrated schematically in Fig. 3.18. As illustrated, angle globes, non-in-line globe valves (inlet and outlet connections offset from each other), and Y-body globe valve designs can be used for this purpose.

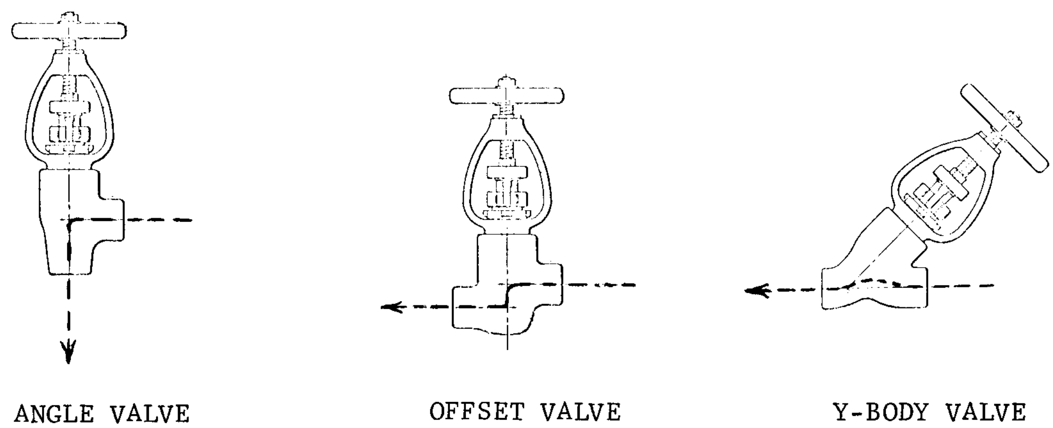


Fig. 3.18. Globe Valves With Reduced Bends.

The effect of reducing the number or degree of bends in a valve flow geometry can be deduced from the preceding discussion of valve flow theory in Section 3.2.3. It should be remembered that in all these cases, the disc should be free and clear of the flow path. Uneven flow and lateral pressure differences across the disc can put lateral forces on the valve stem, increasing stem resistance and valve operating force. This can also result in bending and overstress of the stem similar to the condition of a gate valve without disc guides being used to throttle flow. Therefore, it is important that the geometry of the disc and body be such that the flow can easily get around to the backside of the disc when it is in position above the seat, as illustrated in Fig. 3.19.

The major flow restriction occurs at the seat. Up to and including this point, the flow theory rules of sudden contraction or tapered contraction apply. The seat geometry is more complicated because there are some angular flow boundaries, a flow split, and some rotational effects.

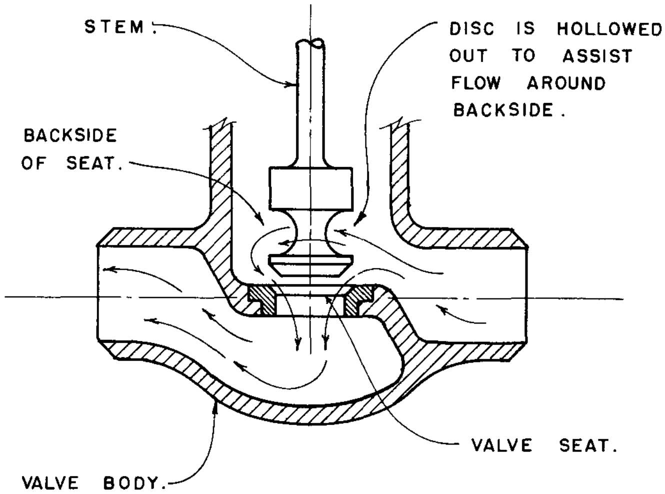


Fig. 3.19. Flow Around Disc.

In some valves where the disc is designed to be fully opened under most flow conditions, the approach region should be designed to be compatible with the flow resistance requirements of the valve in the wide-open condition. Normally, some slight reduction in the flow diameter occurs in the seat area. If this occurs, the approach region should have a slight narrowing taper. If flow bends occur in the approach region or seat area, the approach region should be configured to provide flow bends of reduced sharpness and large radii of curvature. This usually means starting the flow bend as far upstream as practicable. It should be noted here, also, that flow bends should be as uniform as practicable. A gradual flow bend with a sharp entrance or exit may have a higher flow resistance than a sharper bend with no such entrance or exit restrictions. These measures are applied more to Y-body globe valves than in-line globe valves because the Y-body design has the purpose of minimizing flow restrictions when the valve is in the fully open position, and other measures such as proper approach geometry are also utilized as much as possible. Little, if any, approach geometry is used in the in-line type of globe valve. One may see sudden changes in cross-sectional geometry, increase in cross section rather than tapered decrease, and sharp flow direction changes.

If the valve is to be used for throttling purposes, normally the approach geometry during throttling operation does not have a large effect. However, there is one important consideration here. Although they are considered optional, approach geometry measures are sometimes taken to assist in the reduction of disc vibration during throttling operation. As the disc is closed to reduce flow, a large effective reduction in the flow cross-sectional area usually occurs at the point where the flow encounters the disc above the seat. This means that an effective sudden contraction occurs at this point, and it assists in the throttling action but can also cause an unstable vena contracta at this point. As an optional measure in a valve designed for throttling operation, a deep groove may be provided around the disc above the seat area to permit a more gradual reduction in the flow cross-sectional area at this point. This is illustrated in Fig. 3.20. There is a certain amount of "good

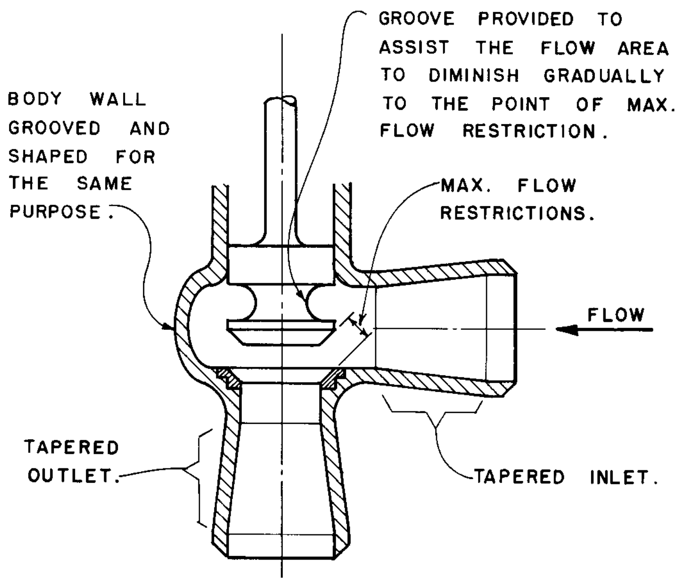


Fig. 3.20. Disc Groove for Flow Area Tapering.

design practice" in this measure since, effectively, the design permits a gradual narrowing of the cross-sectional area of the flow as it progresses from the inlet to the valve seat regardless of the position of the disc. This means that any tapered construction design in the valve flow inlet is not wasted on the disc when the disc is in any position other than fully opened.

After leaving the approach or inlet region of the valve, the flow proceeds to the point of maximum flow restriction. When a globe valve is wide open, there may be no point of maximum flow restriction since flow resistance is normally distributed throughout the entire flow path of the valve. Rather, the point of maximum flow restriction is defined here as that point where maximum flow restriction occurs while the valve is being throttled even though it may normally be wide open. In most globe valves, this point is usually just above the seat as the valve is closed and it proceeds to just below the seat when the valve is almost fully closed, as is illustrated in Fig. 3.21. When the valve is fully open, this area should have an effective flow cross section equal to that of the upstream and downstream pipe. Thus, the disc should be fully nested in the bonnet so it does not provide any obstructions, interferences, and orificing effects in the flow stream. This is particularly applicable to Y-body globe valves which are designed to minimize flow restrictions.

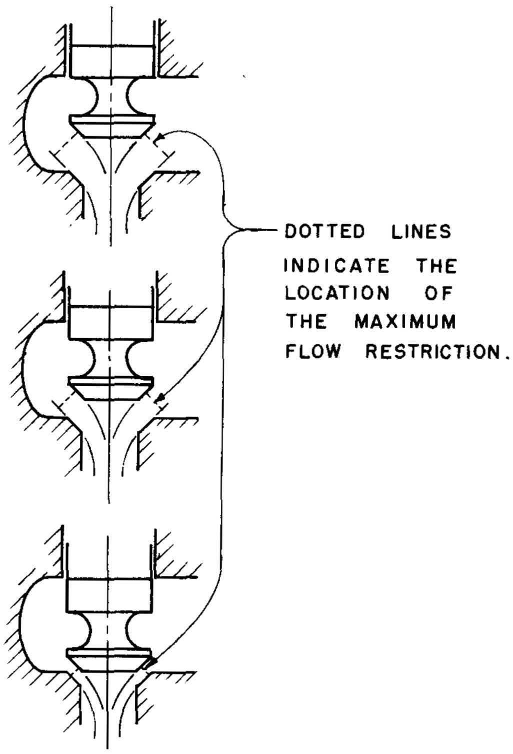


Fig. 3.21. Movement of the Point of Maximum Flow Restriction.

During throttling, it is important to provide for a uniform reduction in the flow cross-sectional area without excessive instability or stem forces. As the flow cross section is reduced, large vena contracta form on the surfaces of the disc and the inside surfaces of the seat, gradually moving downward until well within the seat. It is normally considered good design practice to "anchor" the beginning edge of the vena contracta to some sharp corner of the disc and the seat to aid in the reduction of instabilities caused by axial fluctuation of the vena contracta. This is illustrated in Fig. 3.22.

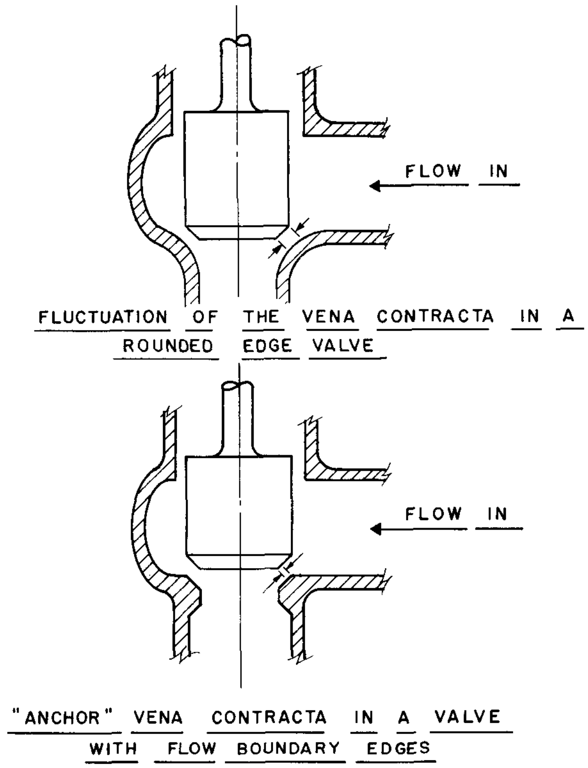


Fig. 3.22. "Anchoring" the Vena Contracta.

(b) <u>Reduction of Cavitation and Erosion</u>. The reduction of cavitation and erosion effects is obtained by helping the flow downstream of the vena contracta to reach velocity equilibrium. Since the velocity of the center stream just downstream of the vena contracta is very high, this velocity should be reduced to that of the flow stream surrounding it before it is allowed to contact the walls of the valve body. It should be remembered throughout this discussion that the primary method of reducing cavitation and erosion problems is through system design. The flow and pressure drop should not be such as to cause abnormally high velocities or abnormally high pressure reductions within the valve.

From a valve design standpoint, there are some measures that can be taken to assist in minimizing cavitation where there is a likelihood that it will occur. One such measure discussed in Section 3.2.3 is a device to promote mixing of the flow downstream, and this device is illustrated in Fig. 3.10 (page 55). As previously stated, another method of reducing the effects of cavitation is to assist the system in reaching equilibrium as soon as possible. Cavitation occurs because of the collapse of vapor bubbles within the flowing fluid during pressure recovery. Since this collapse occurs somewhat farther downstream of the location of the saturation pressure, there is an attendant amount of kinetic energy applied to the collapse that is dissipated in a pressure wave added to the fluid. Theoretically, cavitation effects can therefore be reduced if the downstream surfaces are tapered so gradually that essentially equilibrium conditions occur at all points downstream, as is illustrated in Fig. 3.23. This means that vapor collapse occurs at or near the location of attainment of saturation pressure. The erosion effects of high throttling velocities can also be minimized through the use of these same two measures.

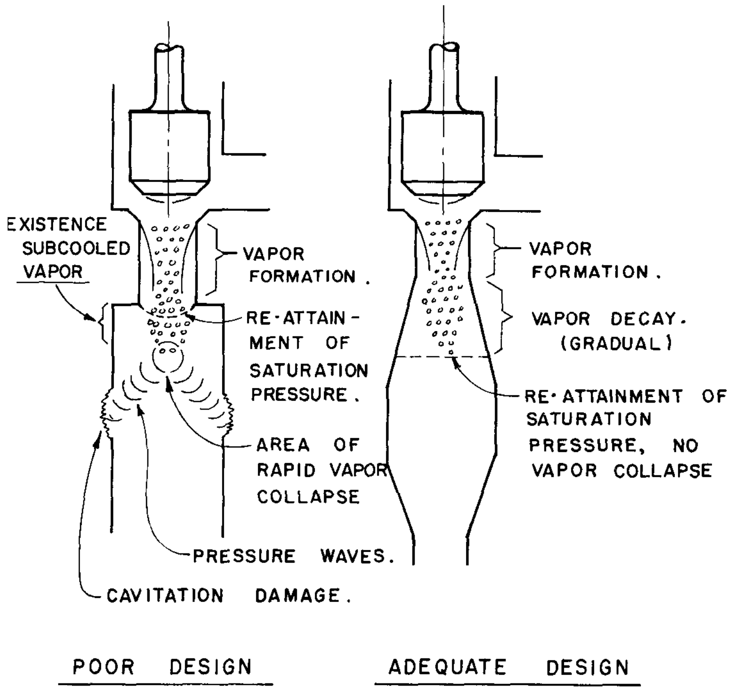


Fig. 3.23. Minimizing Cavitation Damage by Flow Area Tapering.

(c) Reduction of Disc Rotation. A disc which is prone to rotate if the valve is used for throttling purposes is illustrated in Fig. 3.24.

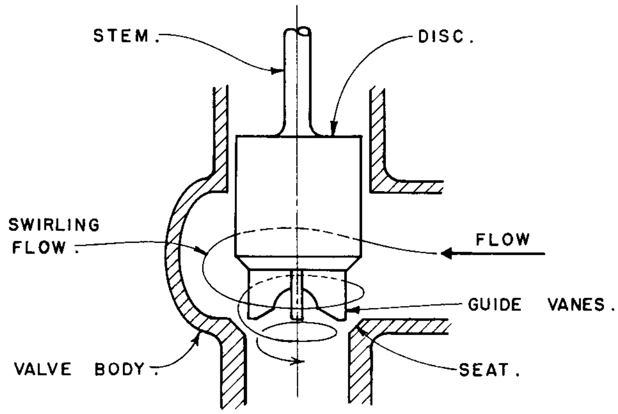


Fig. 3.24. Disc Rotation.

As may be seen, the vanes under the disc, while useful as disc guides, will act as paddles to propel the disc as the flow swirls around them. On the other hand, a valve disc configuration which probably will not allow disc rotation is illustrated in Fig. 3.25. In this design, fluid swirling is not encouraged as it is in the annular flow area between the

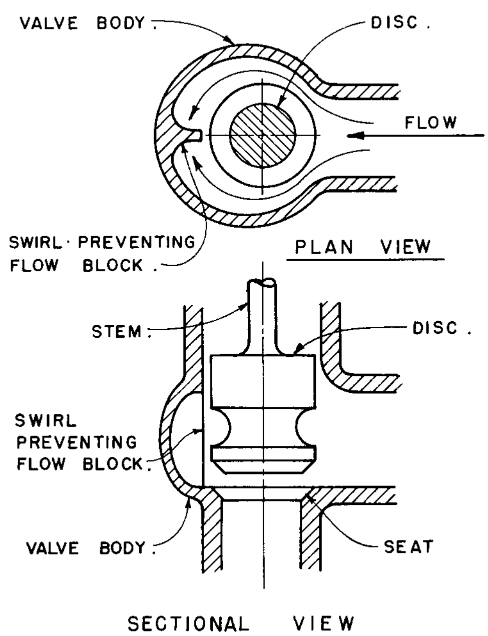


Fig. 3.25. Design to Eliminate Flow Vortex.

disc and seat in the design illustrated in Fig. 3.24. Other valve designs that tend to encourage a flow vortex with resultant disc rotation during throttling operation are illustrated in Fig. 3.26.

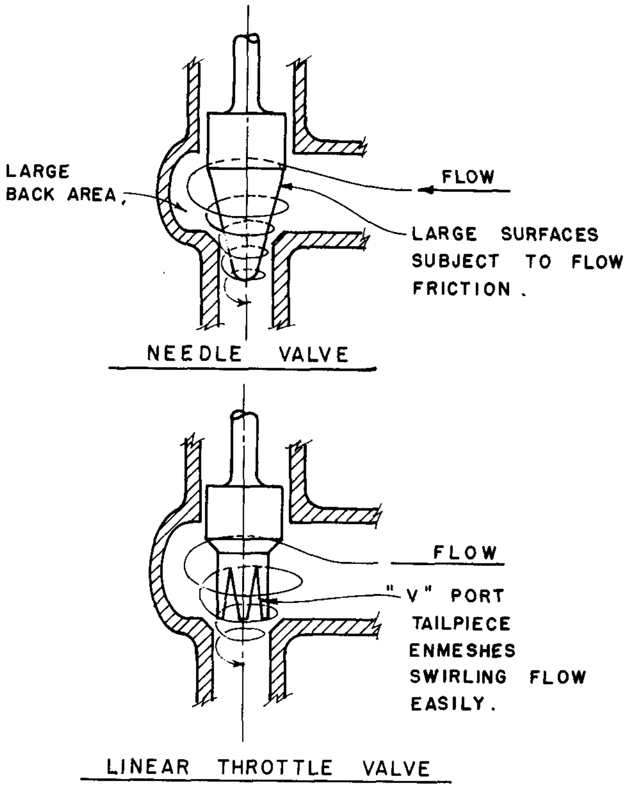


Fig. 3.26. Configurations Which Tend to Encourage Flow Vortex.

The possibility of disc rotation may sometimes be hard to predict for high-pressure and high-temperature applications. An otherwise smooth round disc with no vanes or other protuberances may sometimes be caught up in the swirling motion simply by surface friction. The disc-to-stem connection must be firm enough to resist fluid-induced rotation but not firm enough to resist rotation when the disc contacts the seat and the handwheel is turned further to tighten the closure.

Disc vibration is somewhat more difficult to predict. Disc vibration may sometimes be caused by oscillation in the location of the vena contracta. This can be corrected by changing the seat design to incorporate a slight break that will "anchor" the vena contracta, as previously discussed. It is usually desirable to provide a generous amount of guidance for the disc and make the tolerances very small. Some typical guidance designs are illustrated in Fig. 3.27. The rules governing guide surface wear should be followed, especially for close tolerance design.

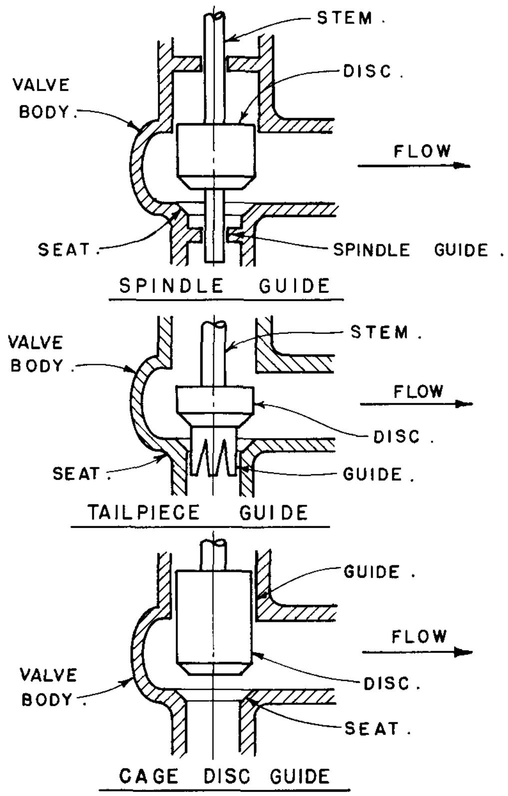


Fig. 3.27. Typical Disc Guides.

(d) Throttling Linearity. Linearity (equal increments of valve opening resulting in equal increments of valve flow increase) is generally only necessary in those valves which perform control functions where the response of the valve in terms of flow to a given control signal that moves the stem must be known. The linearity is achieved by designing the disc-seat geometry so that the incremental cross-sectional areas of disc-seat opening are equal for equal increments of stem travel. Since the forces on the stem vary with disc position (the closer the disc is to being seated, the greater are the flow forces acting on it), the valve actuating mechansim must be capable of developing the required movement forces without affecting the linearity.

Stem force and position characteristics are determined by disc design. Various disc designs intended to achieve various types of stem position characteristics are illustrated in Fig. 3.28. Complicated and

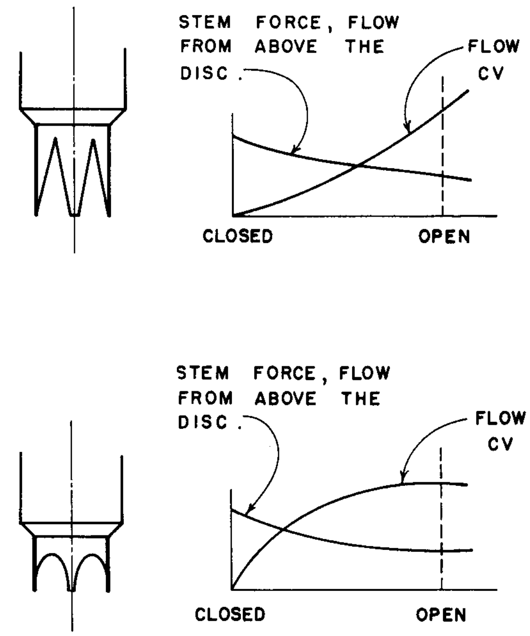


Fig. 3.28. Throttle Valve Flow and Stem Force Characteristics.

precision geometries are used in some valves to reduce non-linearity. A disc-within-a-disc feature sometimes used for this purpose is illustrated in Fig. 3.29.

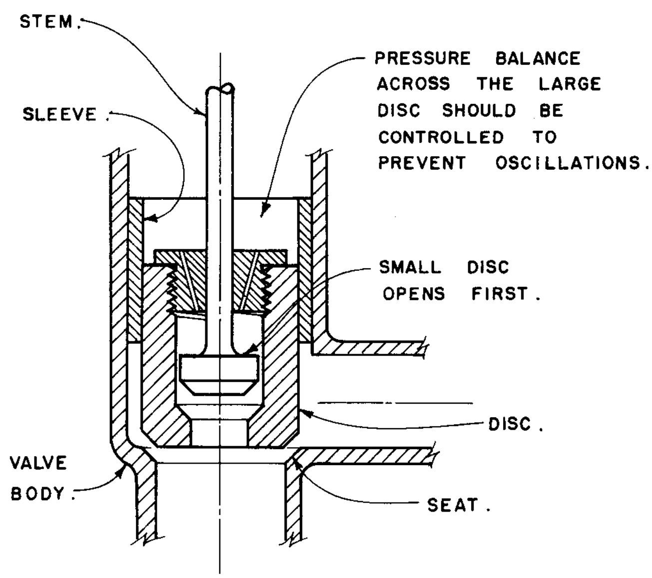


Fig. 3.29. Disc-Within-a-Disc Design.

(e) High-Pressure Flow Throttling. The terminology "high-pressure throttling" is intended to denote the situation where a throttle valve is required to throttle flow at very low flow rates and very high differential pressures for long periods of time. High-pressure flow throttling poses a particularly severe problem with respect to the design of throttle valves. One technique used is to provide a balanced pressure disc, as illustrated in Fig. 3.30.

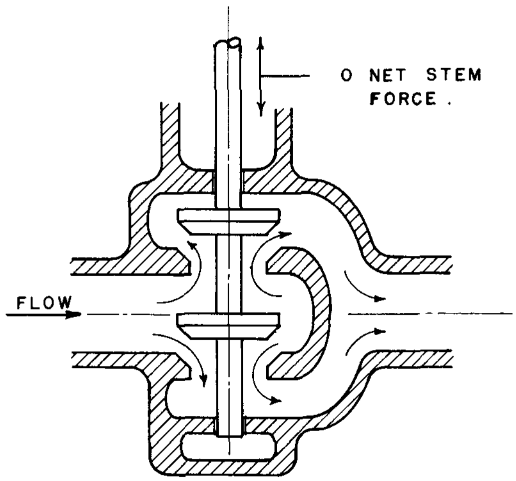


Fig. 3.30. Balanced Pressure Disc Throttle Valve.

In smaller valves, the disc is often of a needle design, as illustrated in Fig. 3.31. It is recognized that the design for this type of valve is not desirable for system shutoff purposes. Use of this nature

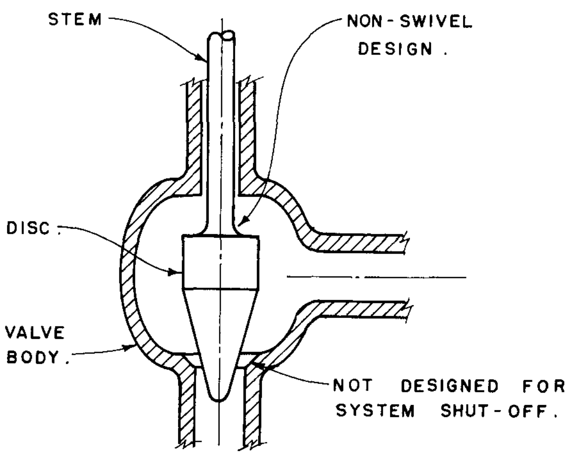


Fig. 3.31. Needle Valve Design.

would cause disc wear and could result in the disc seizing within the seat.

The disc-within-a-disc feature illustrated in Fig. 3.29 (page 69) is sometimes provided for high-pressure throttling purposes. In these cases, the nested disc is designed with good disc guidance and other features to provide good throttling properties. However, in this type of valve, there sometimes is a danger of severe disc oscillation as the large disc is lifted from its seat if pressure balance across the disc is not controlled.

3.2.5 Surge Pressure Characteristics of Swing Check Valves

The large swing check valves used in primary coolant loops of nuclear power plants have a special flow-related characteristic which has been studied analytically. These studies have been directed toward the minimization of surge pressure caused by check valve "slam". Such slams can arise from a number of different flow conditions and can be quite destructive to other components such as differential pressure cells that are wetted by the primary coolant. The phenomenon arises when the pump or pumps in one piping loop are stopped while there is still flow in another connected loop. The flow from the operating loop rapidly closes the check valve in the nonoperating loop, causing a high pressure pulse that is brought about by the kinetic energy absorbed.

The problem is a transient one solvable by differential equation analysis of various moments and initial conditions of the system. Essentially, a solution is sought in which the check valve would reach the closed position at exactly the same time that the flow velocity of the system reaches zero so that reverse flow does not start. This usually means that the check valve should be designed to close with a reduced total angle of disc travel and that the weight of the disc should be increased to the point where the valve begins to close even with forward system flow.[6] Analyses available on this subject provide detailed information which can be used to design check valves from this standpoint.

One such analysis[7] is based on a typical pressurized-water reactor system with multiple primary loops joining at the reactor, with each loop containing a pump, check valve, and heat exchangers. It was assumed that all pumps were operating when the power to one was cut off. A differential analysis of the system is presented in which a basic second-order differential equation requiring inputs of flow velocity versus time, valve flow coefficients as a function of angular disc position, and various constants is solved. This equation provides angular position of the disc as a function of time. This can then be compared with the flow velocity as a function of time to determine whether the disc closes at the same time the flow velocity drops to zero. If not, further analyses, also presented,[7] can be used to determine the surge pressure transmitted through the fluid when the disc closes. The analyses presented were confirmed by experimental evidence that showed a close agreement. It was concluded that it is possible to predict accurately from flow characteristics and basic constants of various pivoting check valves the surge pressure which will be generated in any system for which the coast-down curve (velocity versus time) and valve flow coefficients are known.[7]

If an analysis indicates that the valve requires corrective measures to decrease closing time, there are several valve design techniques available for this purpose. The total angle of disc travel can be reduced by providing an inclined seat, as illustrated in Fig. 3.32. This may not be

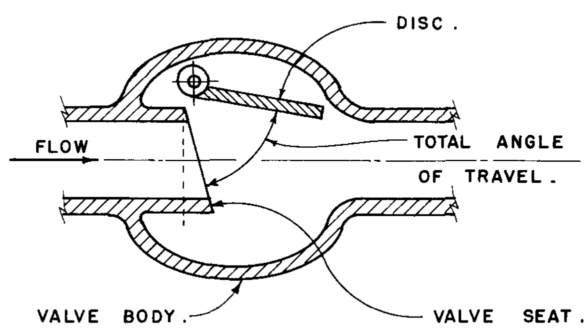


Fig. 3.32. Check Valve With Inclined Seat.

satisfactory if the system is required to be capable of having some natural circulation flow. Normally, the provision of natural circulation requires minimization of valve flow resistances, and a negatively inclined angle is sometimes used with the disc hanging slightly open to minimize flow resistance, as is illustrated in Fig. 3.33. A drawback of

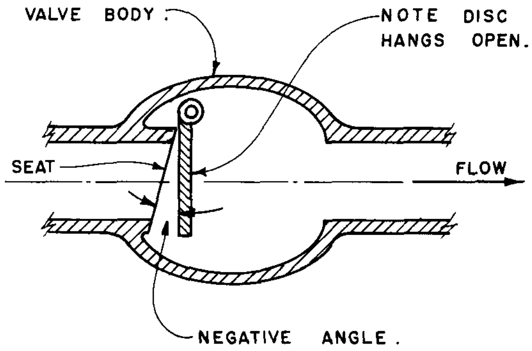


Fig. 3.33. Check Valve With Negative-Angle Inclined Seat.

the negative seat angle design is that it is subject to slamming whenever the first pump in another loop is started. Another way of reducing total disc travel is to restrict the fully open disc position to a position less than that of a horizontal disc completely removed from the flow stream, as is illustrated in Fig. 3.34. This design will result in more flow restriction than other designs.

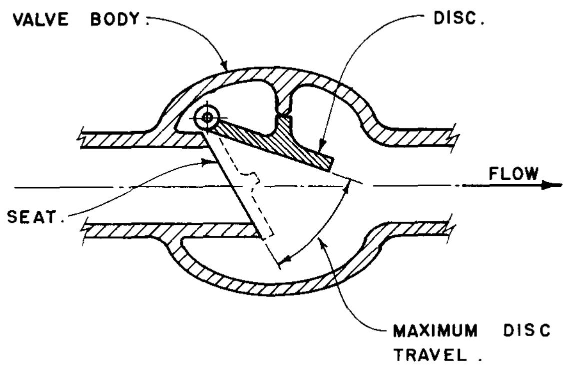


Fig. 3.34. Restriction of Fully Opened Disc Position to Reduce Disc Travel.

Other measures have been tried. Since it is desirable to increase the valve closing moment due to gravity, a large closing moment arm or distance from the pivot to the disc is sometimes used. Such a design is usually made more feasible by articulating the disc arm, as illustrated in Fig. 3.35. This allows the long-armed disc to reach a fully open

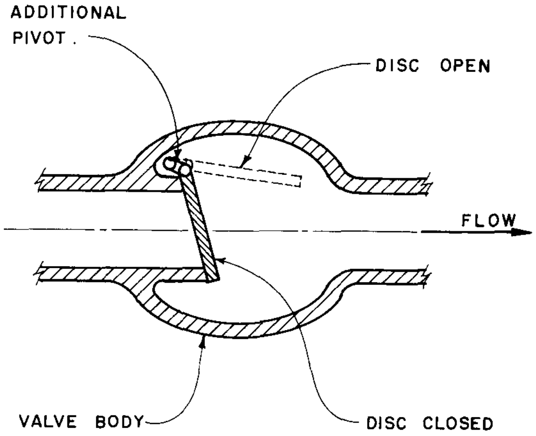


Fig. 3.35. Articulated Check Valve.

position more easily. However, the addition of the second pivot complicates the valve design and increases the chances for small valve fastener pieces to be eroded off the valve and swept downstream.

Additional mechanical means are sometimes provided to decrease valve closing time. A pivot spring may be added, as illustrated in Fig. 3.36.

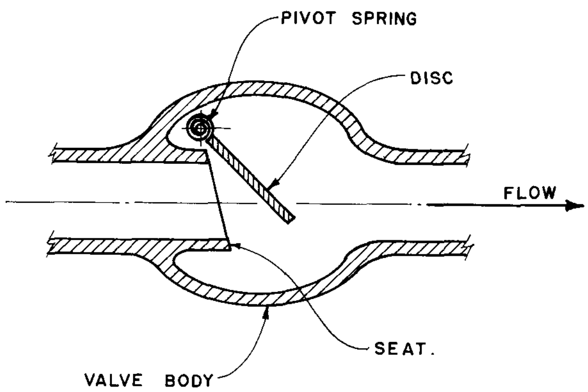


Fig. 3.36. Check Valve With Pivot Spring.

This spring can be sized to provide a boost to disc closure at the fully opened position where it is most needed. However, the spring and disc combination may be induced to oscillate under normal fully open flow conditions, advancing pivot wear and introducing undesirable flow oscillations to the system. If a shaft penetration can be tolerated, an outside counterbalancing weight can be provided, as is illustrated in Fig. 3.37.

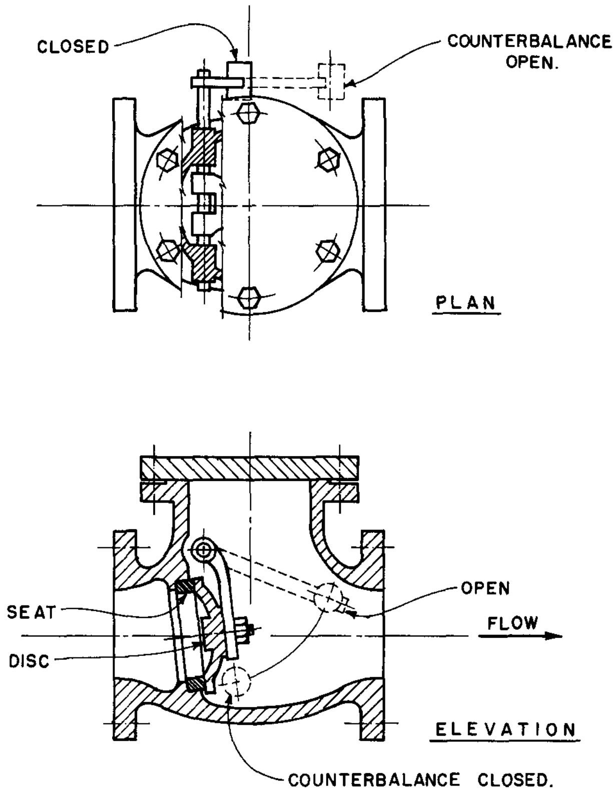


Fig. 3.37. Check Valve With Outside Counterbalancing Weight.

The outside counterbalance is not advisable in nuclear applications because of penetration packing leakage and the possibility that the pivot arm might become a containment-piercing projectile.

3.3 Materials

Considerations related to materials are a large part of valve design. However, this is not unique to valves inasmuch as other components of piping systems have corrosion, wear, and strength problems that must be solved through material design. Since valves are not unique in this respect, much of the information on materials presented in this section will be applicable for other system components such as pumps, strainers, pipe, and instrumentation.

3.3.1 General Material Requirements

Valves have wear surfaces, bearings, and rotating shaft seals similar to those of pumps. Large valve bodies take on the appearance of small pressure vessels with several nozzles, flanged covers, and mechanical penetrations. Therefore, the materials of valve bodies must simultaneously have good weldability, corrosion resistance, mechanical seal capabilities, and castability and machineability properties suitable for being configured into a complicated geometry. The selection of materials for valve internals must include consideration of the various problems related to crevice corrosion, seat leakage, stem leakage, and wear properties of contacting materials. Since machine design and vessel design problems are combined within the same envelope, the materials of valves must satisfy machine design considerations of fastening, sealing, and providing good bearing surfaces while satisfying vessel design considerations of strength, ductility, weldability, and corrosion resistance. In effect, valve materials must achieve a balance among properties that represent the best possible compromise in characteristics.

Since the balance of material properties plays a large part in valve design, it would appear that most valve problems could be solved by the use of appropriate materials. This is true for many parts of the valve. Problems of valve leakage or sticking can perhaps be solved by upgrading materials of the disc, disc guide, or seat. On the other hand, there are cases where the use of appropriate materials alone cannot solve the problems. A case in point is the problem of severe seat-disc wear in certain throttle valves. In many such cases, even the best available materials cannot withstand the continued exposure to severe throttling.

From the above example, it can be seen that materials, in general, set a practical upper limit on valve capabilities. The best example of this is hard-faced seating surfaces where some materials are used simply because none better exist. Recognizing this, one must accept the limitations of such materials and provide for them in other areas of valve and system design. Since mechanical wear cannot be eliminated entirely, valve leakage will continue to recur. The purpose of this section on materials is to show how such problems can be reduced to a practical minimum and, hopefully, to practical insignificance through the use of better materials.

Consideration of materials must go beyond merely investigating their hardness, surface finish, and corrosion resistance as this constitutes only a portion of a thorough evaluation. While it is beyond the scope of this section to go into all facets of all materials, a few examples of additional considerations will serve to illustrate the point being made. These additional considerations include (1) fabrication, (2) radiation, and (3) heat treatment.

1. Fabrication. While colmonoy is very good as a seat material, good weld deposits are difficult to make when this material is used, thereby resulting in a significant fabrication problem. In addition, this material is extremely brittle and therefore difficult to machine.

2. Radiation. The cobalt in stellites causes high after-shutdown plant radiation levels because of the activation of the corrosion product in the core. However, until suitable alternates are developed, the hardness and wear resistance of stellite make it a widely used seating material.

3. Heat Treatment. To be suitable for use in valve stems, the heat treatment of 17-4 PH alloys must be at a temperature of approximately 1100°F. Heat treatment in the temperature range of 850 to 900°F leaves this material subject to stress corrosion cracking.

In summary, a thorough evaluation of the acceptability of a given material for a particular application must include consideration of the complete gamut of usage from the constituents and treatments of the base materials through possible fabrication problems to effects which are brought about by the type of usage. The requirements placed on valve materials for both pressure boundary parts and valve internals are considered in this section. Pressure boundary (body and bonnet) materials for valves of 4-in. size and above in radioactive fluid service are limited almost exclusively to carbon steel, low-alloy steels, and stainless steel. On the other hand, several materials are used for valve internals. This presentation is directed toward establishing the most important material qualities necessary while simultaneously outlining material characteristics and providing criteria upon which selection can be based. Using these techniques and knowing the important material qualities required, materials can be readily selected from the many alloys now available and yet to be developed.

3.3.2 Corrosion of Valve Materials

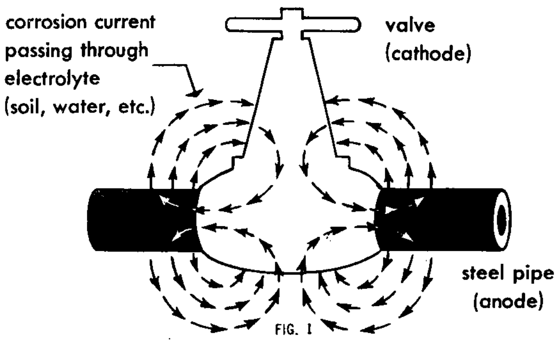
The common types of corrosion important to valve materials[8] are illustrated and described briefly in Fig. 3.38. Reference to this figure will facilitate a grasp of the corrosion concepts discussed herein. In addition, many of the terms used in this discussion are defined in the Glossary of Terms provided in this document. The materials discussed with respect to corrosion include carbon and low-alloy steels and stainless steel.

(a) Carbon and Low-Alloy Steels. The most common type of carbon steel corrosion is simple surface oxidation. The oxide is not very adherent and cannot be relied upon to inhibit the progress of corrosion. Once set in motion, this type of corrosion covers the entire surface of the exposed metal and proceeds at a relatively constant rate with perhaps some tapering off as the lifetime of the metal progresses. Various chemical factors are important in advancing or retarding this type of corrosion. The most important of these factors affecting the corrosion rate are the potential of hydrogen (pH), oxygen concentration, and galvanic coupling.

The nature of general surface corrosion of carbon steel is described in detail by H. H. Uhlig on pages 79 through 96 of his book Corrosion and Corrosion Control.[9] It appears that by the electrochemical theory, corrosion relates to a network of short-circuited galvanic cells on a metal surface where metal ions go into solution at anodic areas in amounts equivalent to the reaction at cathodic areas. The rate of iron corrosion is usually controlled by the cathodic reaction, which can be accelerated by dissolved oxygen. Within a freshwater pH range of about 4 to 10, the corrosion rate is independent of the pH and depends only on how rapidly oxygen diffuses to the metal surface. Uhlig[9] concludes that as long as oxygen diffusion is the controlling factor, any variation in the

Corrosion is nature's wasteful way of returning metals to their ores. The chemistry of corrosion emphasizes the basic corrosion reaction $M^{o} \rightarrow M^{+} + \text{electron}$, where M^{o} is the metal and M^{+} a positive ion of the metal. As long as the metal (M^{o}) retains possession of its electrons, it retains its identity as a metal. When it loses possession of them by any means whatever, it has experienced corrosion. Physical forces sometimes join with chemical forces to cause a valve failure. We now list the many common varieties of corrosion, which show up in many ways and forms and mostly overlap each other. The mechanism of corrosion resistance is attributed to the formation of a thin protective corrosion film on the metal surface.

GALVANIC CORROSION



When two dissimilar metals are in contact and are exposed to a corrosive liquid or electrolyte, a *galvanic* cell is formed and the flow of current causes increased corrosion of the anodic member. The corrosion is usually localized near the point of contact. One method to minimize corrosion can be achieved by plating the dissimilar metals.

GALVANIC SERIES ANODIC, CORRODED ENDS

Magnesium
Magnesium alloys

Zinc

Aluminium

Cadmium

Mild Steel
Alloy Steel
Wrought Iron
Cast Iron

Stainless Steel (active)

Ni-resist

Soft solders
Lead
Tin
Brasses
Copper
Bronze
Copper-Nickel Alloy

Nickel
Inconel

Stainless Steel (passive)
Hastelloy

Titanium

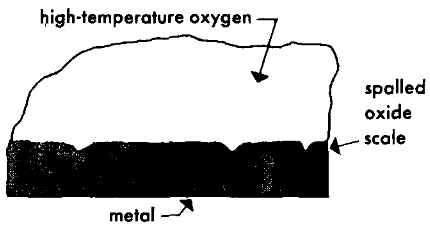
Silver

Graphite
Gold
Platinum
Cathodic protected end

Metals near the top act as anodes and suffer corrosion when coupled with one nearer the bottom. Those close together corrode more slowly. Couples of metals within one of the group corrode the least.

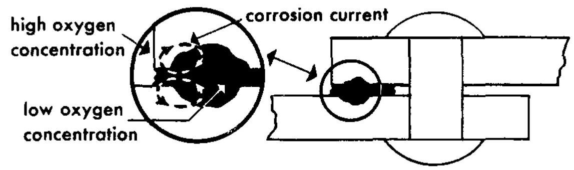
Fig. 3.38. Common Varieties of Corrosion That Affect Valve Materials (from Ref. 8).

HIGH-TEMPERATURE CORROSION



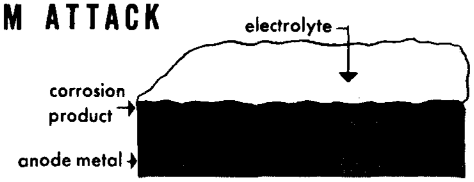
To predict the effect of high-temperature oxidation we need data on: (1) metal composition (2) atmosphere composition (3) temperature and (4) exposure time. So it's not easy to tell what will happen in one case, from the results of another. But we do know that most light metals (those lighter than their oxides) form a non-protective oxide layer that gets thicker as time goes on. The layer forms, spalls and reforms. Other forms of high-temperature corrosion include sulfidation, carburization and decarburization.

CREVICE CORROSION



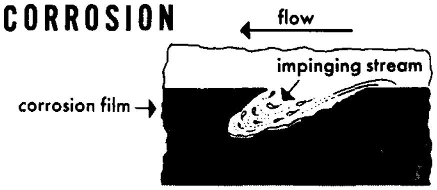
You'll recognize this condition by its presence in crevices. It sets up differences in solution concentration. The crevice, above, for example, hinders diffusion of oxygen. Results: High and low oxygen areas which are anodic, cause concentration cells. Metal-ion concentration cells, much like their oxygen counterparts, also strive to balance out concentration differences. Thus, when the solution over a metal contains more metal ions at one point than another, metal goes into solution where ion concentration is low. Crevice corrosion can be minimized by avoiding: accumulation of deposits on metal surfaces, sharp corners, gasket joints or other conditions favoring stagnate areas of solution.

UNIFORM ATTACK



Look for a general wasting away of the surface. You'll find this condition all too common where metals are in contact with acids and other solutions. The corrosion product may form a protective layer on the metal, slowing down corrosion. Or, as in the case of direct chemical attack, the corroded material easily dissolves in the corrosive material. The problem can be solved by selecting a more corrosion resistant metal.

EROSION CORROSION

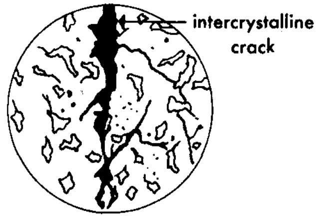


Similar in their method of attack and net effects are impingement corrosion and cavitation corrosion. Here's how they do their damage:

An impinging high velocity stream breaks through corrosion scale, dissolving the metal. The effect depends mainly on liquid speed. Excessive vibration or flexing of the metal can also have similar effects. Cavitation, a common form of corrosion in pumps and sometimes in valves depends on the hammer-like effect produced by collapsing air bubbles. Bubbles break down when they pass through a pressure drop area.

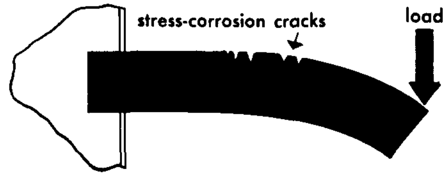
Fig. 3.38. (continued)

INTERGRANULAR CORROSION



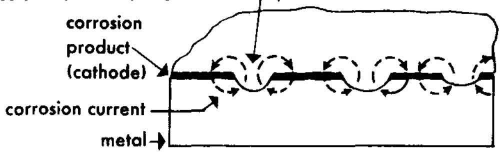
You may be faced with intergranular corrosion for a variety of reasons. But the result is almost the same—selective attack along the metal's grain boundaries, destruction of mechanical properties, intercrystalline cracking such as we picture in the photomicrograph above. Austenitic stainless steels are subject to intergranular corrosion by many corrosives if not properly heattreated or exposed to the sensitizing temperatures of 800 - 1500°F. This condition can be eliminated by preannealing and quenching from 2000°F, by using low carbon (LC) stainless steels C-0.03 max) or stabilized types with Columbium or Titanium.

STRESS CORROSION CRACKING



Team up high tensile stresses with a corrosive atmosphere and you're likely to be in for trouble. Here's how it develops. Tensile stresses build up at metal surface under static loading. Corrosive action concentrates stresses and causes them to exceed metal's yield point. Result shows up as a local failure. Under continued exposure, metal alternately corrodes and builds up high stress concentrations. Eventually the part may fail. Avoiding this failure can be achieved by early stress relief annealing, proper selection of alloys and design.

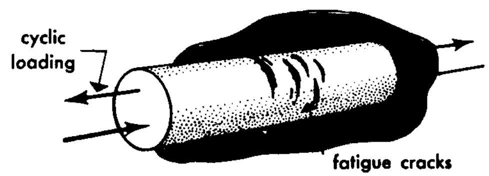
PITTING CORROSION



When protective films or layers of corrosion product break down we get localized corrosion—pitting. An anode forms where the film breaks, while the unbroken film or corrosion product acts as the cathode. Thus we've in effect, set up a closed electrical circuit.

Some stainless steels in presence of chloride are susceptible to pitting attack. The breakdown occurs because of some inhomogenity in the metal surface or rough spots.

CORROSION FATIGUE



In much the same way that static stresses link up with corrosion to produce stress-corrosion cracking, cyclic loads work hand-in-hand with corrosion to cause corrosion fatigue. Metal failure takes place substantially below the fatigue limit for non corrosive condition.

Surprisingly enough, the combined deteriorating effect of these two bed brothers—corrosion and fatigue—is greater than the sum of their individual damages. So it pays to apply best possible corrosion protection when dealing with metals under alternating stresses.

Fig. 3.38. (continued)

composition of a steel and its heat treatment or method of working has no bearing on corrosion properties. The oxygen concentration, temperature, and velocity of the water determine the reaction rate.[9]

It should be clearly noted that even with a minimal amount of oxygen present, there is always some general corrosion of carbon steel. This is brought about mainly by slight galvanic coupling of carbon steel in contact with more noble metals such as stainless steel valve trim. The valve trim must be more noble to eliminate the possibility of galvanic pitting in critical trim contact areas.

The corrosion rates[10] given in Table 3.1 were obtained from tests made where static fluid conditions existed and where apparently no corrosion loss was caused by galvanic effects. These rates therefore represent absolute minimum corrosion rates. In actual practice, when applying the rules on corrosion allowance set forth in the Draft ASME Code for Pumps and Valves for Nuclear Power, one must consider experience and test results on corrosion loss. Summaries of corrosion rate testing given in Ref. 11 are partially given in Table 3.2 for informational purposes.

Carbon steel is known to have the disadvantage of being subject to a certain amount of erosion damage. As the velocity of the fluid in contact with the carbon steel is increased, more oxygen is brought in contact with the diffusion layer of oxide on the surfaces of the carbon steel. This diffusion layer consists of three separate layers of oxide products through which the oxygen diffuses to come into contact with the metal surface. If the flow velocity is increased to the point where the oxide diffusion layer is beginning to be swept away, the erosion rate will begin to increase substantially. At even greater flow velocities where high turbulences and cavitation occur, the fluid penetrates through and removes the entire local oxide layer. This causes rapid cavitation attack. Clearly, as discussed in Section 3.2.4(b), it is important to valve design that low average and local fluid velocities be maintained.

With respect to seal design, it is well known that carbon steel is subject to wire drawing. This means that if small paths of micro-leakage occur at a carbon steel body seal, the leakage flow velocities are likely to build up to high values. Even though an oxide diffusion layer forms on the flow boundaries of the leakage path, it is not strong enough to

Table 3.1. Oxidation of Common Steam Plant Steels in Air and Steam (from Ref. 10, p. 265)

Oxidizing medium	Temperature, F	Penetration, mils — Time, hr. 100	1,000	10,000	100,000
ASTM A216, Grade WCB					
Air	850	0.12	0.22	0.4	0.74
	1000	0.18	0.52	1.6	4.90
	1150	0.88	3.70	15.5	66.0
Steam	850	0.16	0.2	0.34	0.6
	1000	0.29	0.86	2.6	8.0
	1150	3.1	11.6	46.0	150
ASTM A105, Grade 11 (AISI 1029)					
Air	850	0.12	0.22	0.57	1.55
	1000	0.38	0.68	2	6
	1150	0.68	1.61	14.0	105
Steam	850	0.10	0.21	0.43	0.90
	1000	0.22	0.50	1.2	2.7
	1150	1.25	5.8	28	102
ASTM A108, Grade 1035 CD (AISI 1035)					
Air	850	1.81	0.32	0.58	1.1
	1000	0.26	0.64	1.5	3.7
	1150	0.88	3.3	11	40.1
Steam	850	...	....	....	...
	1000	0.19	0.84	3.8	17
	1150	4.3	13	39	101
AISI 1116 CD					
Air	850	0.15	0.42	1.3	3.8
	1000	0.35	1.0	2.8	8.6
	1150	0.70	3.2	14.8	68.0
Steam	850	...	....	....	....
	1000	0.23	0.72	2.2	7.0
	1150	2.7	8.8	29	96.8
ASTM A193, Grade B7 (AISI 4140)					
Air	850	0.09	0.17	0.44	1.1
	1000	0.2	0.54	1.5	4.1
	1150	0.9	2.9	9.5	30
Steam	850	...	....	....	....
	1000	0.3	0.96	3.2	10.2
	1150	6.8	13.8	27	55
ASTM A217, Grade WC1					
Air	850	0.12	0.23	0.44	0.82
	1000	0.216	0.58	1.58	4.4
	1150	1.4	3.5	10.4	40.1
Steam	850	....	....	....	....
	1000	0.175	0.57	1.9	6.6
	1150	1.3	6.2	30	124
ASTM A182, Grade F6 (410 Stainless)					
Air	850	0.04	0.027	0.052	0.15
	1000	0.037	0.068	0.125	0.23
	1150	0.074	0.18	0.42	1
Steam	850	0.036	0.16	0.62	0.95
	1000	....	....	....	....
	1150	0.019	0.29	8	1.3
ASTM A217, Grade WC6					
Air	850	0.095	0.19	0.53	1.9
	1000	0.15	0.45	1.4	4.3
	1150	0.55	2.5	11.5	53
Steam	850	0.099	0.18	0.36	0.62
	1000	0.2	0.62	1.88	5.8
	1150	1.2	3.6	9.4	26.5
ASTM A217, Grade WC9					
Air	850	0.07	0.22	0.58	1.6
	1000	0.1	0.41	1.75	7.4
	1150	0.27	1.09	4.1	16
Steam	850	...	....	....	....
	1000	0.8	2.1	5.4	14
	1150	1.3	3.6	8.2	21

100,000-hr data extrapolated from 10,000 hr.

Table 3.2. Corrosion of Carbon and Low-Alloy Steels in Pressurized Water (from Ref. 11, pp. 26-31)

ALLOY	TEMPERATURE	SPECIMEN PREPARATION	DISSOLVED GAS (ml/liter) O_2	DISSOLVED GAS (ml/liter) H_2	pH	ADDITIVES	FLOW RATE (fps)	CORROSION RATE (mg/dm^2 - month) NONDESCALED	CORROSION RATE (mg/dm^2 - month) DESCALED
a	600°C	Electropolished	Degas	Degas	11	LiOH	30		-15
a	600°C	Ground	Degas	Degas	11	LiOH	30		-25
b	600°C	Electropolished	200-1000		9-10	NH_4OH	30		-25
b	600°C	Pickled, sanded, and electropolished.		500	11	LiOH	30		-25
c	600°F	Machined	<0.1	20	9.5	102 ppm NH_3 NH_4OH	20		220
c	600°F	Machined	<0.1	70	9	10 ppm NH_3 NH_4OH	21		180
c	600°F	Machined	<0.1	30	10	5 ppm HN_3 LiOH	21		130
c	600°F	Machined	<0.1	20	7		15		430
c	600°F	Pickled		100	d		30	-150	-238
c	600°F	Surface ground		100	d		30	-189	-260
c	600°F	Sanded		100	d		30	-160	-242
e	500°F	Machined	Degas	Degas	7		25		
e	500°F	Machined	30		7		25	-320	
e	500°F	Machined	240		7		25	-44	
e	500°F	Machined		18	7		25	-350	
e	600°F	Machined	Degas	Degas	7		25	+16	
e	600°F	Machined, sanded, and electropolished	<0.1		11	LiOH	30		30
e	600°F	Machined	<0.3	Equilibrium	7-10	LiOH	30		320
e	600°F	Sanded, electropolished, and pickled	<0.2	20	7-10	1 ppm NH_3 NH_4OH	30		360
e	600°F	Sanded, electropolished and pickled	<0.1	500	11	100 ppm NH_3 NH_4OH	30		85
e	540°F	Sanded, ground	<0.1	500	7		30		270
e	600°F	Sanded, electropolished, and pickled	400	low	7-9	2 ppm NH_3 NH_4OH	30		85
e	600°F	In-pile specimen	0-30	100	8-10		30		120 min 150 av 180 max

Table 3.2 (continued)

ALLOY	TEMPERATURE	SPECIMEN PREPARATION	DISSOLVED GAS (ml/liter) O_2	H_2	pH	ADDITIVES	FLOW RATE (fps)	CORROSION RATE (mg/dm^2 - month) NONDESCALED	DESCALED
f	600°F	Sanded, electropolished, and machined	<0.1	Degas	11	LiOH	30		30
f	600°F	Sanded, electropolished, and machined	<0.3	500	11	LiOH	30		80
f	600°F	Sanded, electropolished, and machined	<0.2	20	7-10	NH_4OH	30		350
f	600°F	Sanded, electropolished, and machined	<0.2	500	11	NH_4OH	30		70
f	600°F	Sanded, electropolished, and machined	<0.2	20	7-10	100 ppm NH_3 NH_4OH	30		450
f	200-600°F	Sanded, electropolished, and machined	400	low	7-9	NH_4OH	30		145
g	600°F	Machined	<0.1	20	9.5	NH_4OH	20		220
g	600°F	Machined	<0.1	70	9	NH_4OH	21		140
g	500°F	Machined	<0.1	20	9.5	NH_4OH	12		95
h	600°F	Machined	<0.1	20	9.5	NH_4OH	20		220
h	600°F	Machined	<0.1	20	11	LiOH	12		140
h	600°F	Machined	<0.1	21	10	LiOH 5 ppm NH_3	21		290

[a] Nominal composition: 0.15% C; 0.3-0.6% Mn; 0.03% P; 0.03% S; 0.5% Si; 2.0-2.5% Cr; 0.9-1.1% Mo (Croloy 2 1/4).

[b] Nominal composition: 0.15% C; 0.3-0.6% Mn; 0.03% S; 0.03% P; 0.5-1.0% Si; 1.0-1.5% Cr; 0.45-0.65% Mo (Croloy 1 1/4)

[c] Nominal composition: 0.27% C; 0.62% Mn; 0.26% Si; 0.012% P; 0.04% S (ASTM 212 A).

[d] Cationic corrosion products continuously removed by ion-exchange purification system.

[e] Nominal composition: 0.35% C; 0.90% Mn; 0.15-0.30% Si; 0.04% P; 0.04% S (ASTM 212 B).

[f] Nominal composition: 0.04% C; 0.41% Mn; 0.009% P; 0.036% S; 0.01% Si (carbon steel)

[g] Nominal composition: 0.25% C; 0.14% Mn; 0.012% P; 0.015% S; 0.17% Si; 0.10% Al; 0.46% Ti. (Carbon Steel)

[h] Nominal composition: 0.06% C; 0.08% Mn; 0.008% P; 0.015% S; 0.09% Si; 0.40% Cu; 0.31% Zr.

resist the erosion of the fluid. Consequently, no self-plugging occurs and no resistance is provided to inhibit advancement of the erosion. Once beyond a certain point, this erosion feeds upon itself and rapidly causes a substantial leak to form. The so-called wire drawing across gasketed surfaces is the result. Considerations in achieving proper gasketed seals and alterante seal designs are discussed in Section 4.4. Other than providing better geometry and sealing techniques, there is no way of reducing the tendency of carbon steel to wire draw. As previously pointed out, material design and alloying can help only within limits. Some help can be obtained through chemical control, notably through the reduction of oxygen concentration. By increasing corrosion rates, higher oxygen concentrations increase the possibility of leakage in gasketed flanges.

The conclusions which can be drawn from this discussion for the case of carbon and low-alloy steel valves operated in high pressure and high temperature power plant environments are as follows. First, oxygen concentrations must be reduced and maintained at a low or negligible level. Any increase in oxygen concentration is reflected immediately in a higher corrosion rate. Next, pH control probably has little or no effect on general corrosion of carbon steel. Further, unlike the case for stainless steel, formation of a passive film on carbon steel surfaces is highly undesirable. Although carbon steel passivity reduces general corrosion, its instability and proclivity to local breaking down into pitting mean that instead of being beneficial, it merely substitutes crevice pitting for general corrosion. This is not good in valves where crevices exist.

Last, increasing the flow rate generally increases both general corrosion and the likelihood that a passive film will form. Since both are undesirable, they provide a reason over and above fluid friction and turbulence for avoiding high flow velocities in the design of carbon steel systems and components. Since local flow velocities within valves can be quite high, it is also important to design carbon steel valves to reduce turbulence, eddies, and nozzling inasmuch as preventing local velocities from becoming much higher than the average velocity results in better corrosion control. Even where there are small galvanic effects, erosion-enhanced corrosion in a poorly designed valve can be quite high.

(b) Stainless Steel. The corrosion of stainless steel is retarded because of the thin tenacious film on its surface that is called the "passive" film. A definition and explanation of passive films and passivity is included on pages 57-58, 67-68, and 258-259 of Ref. 9. Uhlig[9] defines a passive metal as one that is active in the electromotive series but corrodes at a very low rate. According to Uhlig, there are two definitions of passivity still in force today. These are as follows.

> "Definition 1. A metal active in the Emf Series, or an alloy composed of such metals, is considered passive when its electrochemical behavior becomes that of an appreciably less active or noble metal.
>
> "Definition 2. A metal or alloy is passive if it substantially resists corrosion in an environment where thermodynamically there is a large free energy decrease associated with its passage from the metallic state to appropriate corrosion products ... ."[9]

Uhlig[9] goes on to state that there are two commonly expressed points of view regarding the nature of the passive film. Sometimes called the "oxide-film theory", the first holds that the passive film of Definition 1 or 2 is a diffusion barrier layer of reaction products that separates the metal from its environment to slow down the corrosion reaction rate. The second point of view, called the "absorption theory of passivity", holds that metals passive by Definition 1 are covered by a chemisorbed film (O_2 or passivating ions) that displaces absorbed water molecules and slows down the rate of anodic dissolution involving hydration of metal ions.[9]

Metals included within Definition 1 are chromium, nickel, and the stainless steels, which are all naturally passive in air. In the case of stainless steel, the passive tightly adherent protective film generally means that the extra thickness of material provided for corrosion protection need be much less than for carbon steel. (General stress design and safety factors usually require a relatively thick body wall to begin with.) Test results indicate that for primary coolant use under normal circumstances, Type 304 stainless steel corrodes at a negligible rate, generally less than 5 mg/dm^2 per month after the passive film is established. Summaries of corrosion testing of stainless steel are provided in Ref. 11 and partially given in Table 3.3 for informative purposes. However, the

Table 3.3. Corrosion of Stainless Steel in Pressurized Water (from Ref. 11, pp. 9-12)

AISI TYPE	TEMPERATURE (°F)	SPECIMEN PREPARATION	TEST DURATION (hr)	DISSOLVED GAS (ml/liter) O_2	DISSOLVED GAS (ml/liter) H_2	pH	ADDITIVES	FLOW RATE (fps)	CORROSION RATE (mg/dm^2-month) DESCALED	CORROSION RATE (mg/dm^2-month) NONDESCALED	POSTTEST APPEARANCE
304	500	Machined	1350	0-5		7	a	30	-3	-6	Steel gray
304	500	Machined	1000	Degas	Degas	7	a	30	-8	-8	Thin, gray film
304	500	Machined	1500	Degas	Degas	7	a	30	-4	-11	Thin, gray film
304	500	Machined	1000		24-77	7	a	30		-1	Loose, thin straw film
304	500	Machined	336	30		7		30		+9	Brown tarnish
304	500	Machined, sensitized	A	Degas	Degas	7-10	LiOH	30	-10	-5	
304	500	Machined	A	1-5		7		30	+5	-5	
304	500	Welded, sensitized	B	0-5		10	LiOH	30		-10	
304	580	Electro-polished	2500		500			15	-2		
304L	500	Machined	401	30		7		30		+19	Lustrous tarnish
304L	600	Machined	500[b]	1-5		7		30	-4	+4	Thin, tight film
304L	500	Machined	1000[b]	1-5		7		30		+2	Thin, tight film
304L	500	Machined	1500[b]	1-5		7		30	-3	+1	Thin, tight film
304L	500	Machined	1500[b]	Degassed	Degassed	7		30		-4	Light blue film
304L	500	Machined	1000[b]		24-77	7		30	-1	0	Very thin, loose film
304L	500	Machined	500[b]		104-654	7		30		-5	
316	600	Machined	334	Degas	Degas	7		25		+52	Powdery black oxide
316	500	Machined	500	0-5		7		30		-17	Steel-gray color
316	500	Machined	1500	1-4		7	c	30	-2	+2	Glossy, thin loose film
316	500	Machined	1000		39-78	7	c	30	0	0	Very thin, loose film
316	500	Machined	500		104-654	7	c	30	0	+2	Very thin, loose film
316	500	Welded, sensitized		0-5		7-10		30	-5	+5	
316	500	Machined	500		25	7		30	-3		
316	500	Machined			0-500	7		30	-5	-5	

[a, b, c] Cationic corrosion products continuously removed by ion-exchange purification system.

burden of design falls more on control of the base material. Stainless steel can be subject to various difficulties caused by lack of proper fabrication or the use of an improper type of stainless steel for the fluid conditions of a particular system. Generally, the most likely difficulties in the applications considered here (high-pressure, high-temperature reactor coolants) are stress corrosion, intergranular corrosion, and pitting.

Stress corrosion is a well-known potential weakness of stainless steel. Stress corrosion cracking is transgranular in nature and occurs in conjunction with applied or residual tensile stress. It is generally considered that the rate of attack is dependent upon the level of stress, with no practical minimum level below which attack does not occur. This type of corrosion is largely dependent upon the chemical nature of the fluid environment and can be practically eliminated with proper chemical control. An indication of the chloride and oxygen concentrations needed for stress corrosion cracking to occur in Type 304 stainless steels is illustrated in Fig. 3.39.

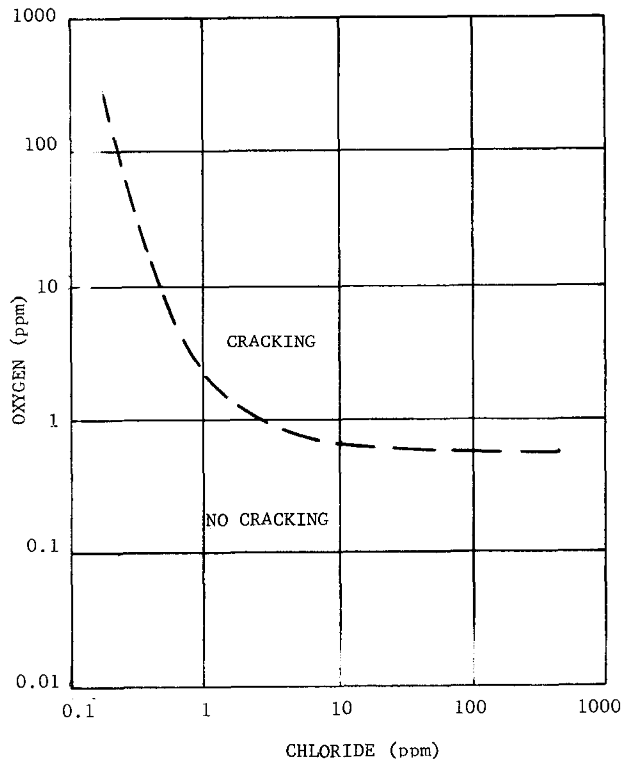


Fig. 3.39. Relation Between Chloride and O_2 Content of Boiler Water on Stress Corrosion Cracking of Austenitic 18-8 Type Stainless Steels (from Ref. 9, p.277).

Different types of stainless steel vary in their susceptibility to stress corrosion attack and in the chemicals to which they are susceptible. The Type 304 and the austenitic stainless steels are susceptible to OH^- and Cl^-, and the martensitic and ferritic stainless steels are susceptible to OH^- and H^+. Slightly acid conditions cause rapid stress corrosion failure of hardened martensitic stainless steel. Increasing the nickel content in austenitic stainless steel reduces susceptibility to some types of stress corrosion. For alloys of chromium-iron-nickel, stress corrosion is practically eliminated at 50% Ni and above, as illustrated in Fig. 3.40. Sometimes, varying the impurities in nickel and

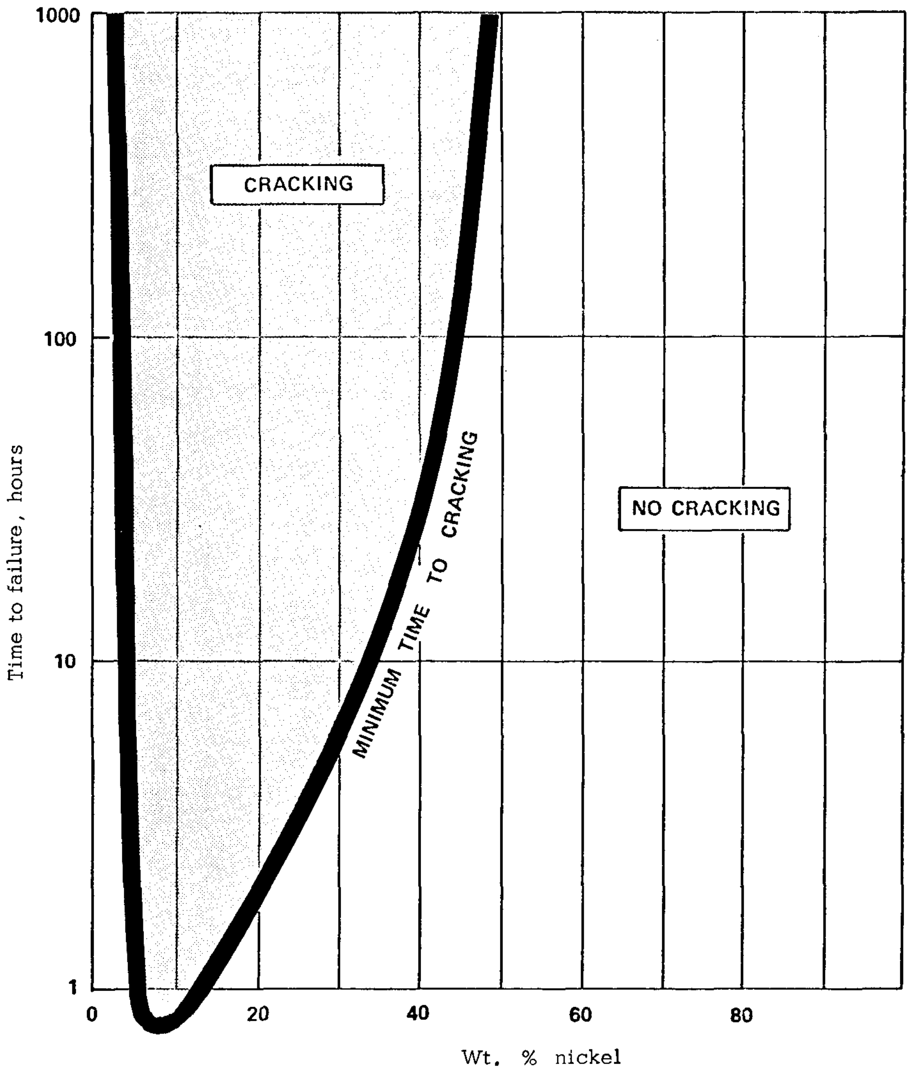


Fig. 3.40. Stress Corrosion Cracking of Cr-Fe-Ni Alloy (from Ref. 9, p. 279).

carbon have an effect, but in general, few other alloying elements have an effect on stress corrosion susceptibility.

For valve pressure boundary parts, the preceding information on stress corrosion cracking means basically that whatever care is taken in the design of piping systems to avoid stress corrosion must be taken in the design of valves with perhaps additional safeguards because of the naturally higher localized stresses in valves. For example, if Type 304 stainless steel is to be used for pipe, Type 316 stainless steel, which has a higher nickel content and greater stress resistance, might be judged to be preferable for use in valve bodies where local stresses brought about by thermal effects and geometry may be expected to be higher.

Intergranular corrosion, selective attack along the metal grain boundaries, is a type of corrosion that potentially can affect the weld joints of valves. The susceptibility to intergranular corrosion is associated with improper heating, termed "sensitizing", which can inadvertently occur during welding. Some valves, especially in larger sizes, are fabricated with welds which join various pieces of the valve. In these and in the attachment welds, care must be taken to reduce the conditions favorable for intergranular corrosion.

Ferritic and austenitic stainless steels are subject to such corrosion. The theory behind this is that the carbon within the alloy migrates to the grain boundaries at the sensitizing temperature, drawing adjacent Cr with it. The area adjacent to the boundary depleted of Cr then loses passivity and is susceptible to rapid electrochemical attack in the presence of an ionic solution. The temperature-versus-time dependence of the effect is illustrated in Fig. 3.41. As can be seen, temperature above or below the sensitizing temperature can be held without sensitization. If temperatures higher than the sensitizing temperature are held, passage must be made through the sensitization range quickly to avoid sensitization. In terms of welding technique, the welding time should be minimized by reducing the number of passes and minimizing the number of repair welds necessary.

Tests to check the immunity to intergranular attack may be performed on prototype valves as part of a purchase. Such tests would involve

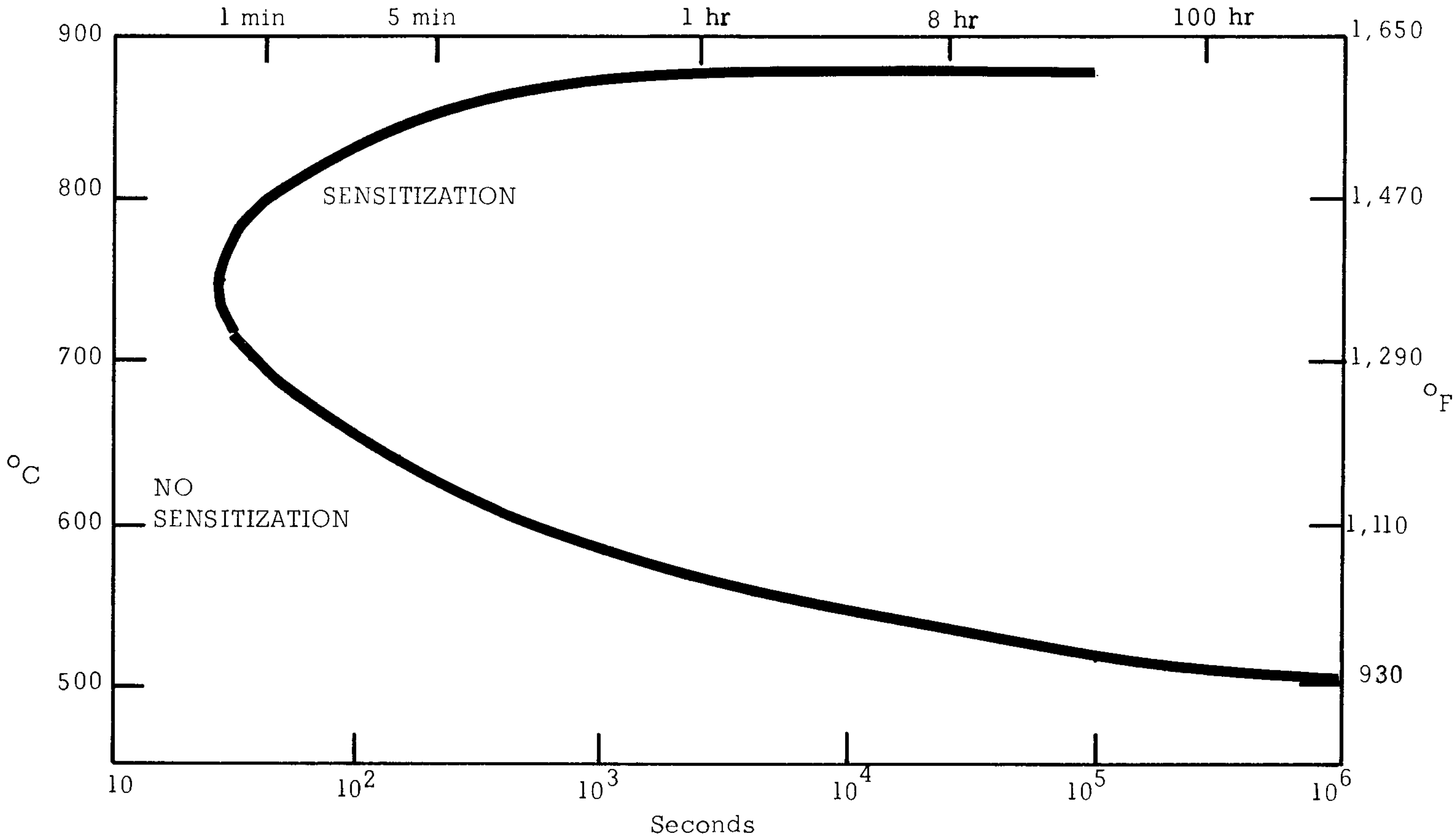


Fig. 3.41. Effect of Time and Temperature on Sensitization of 18.2% Cr, 11.0% Ni, 0.05% Cr, and 0.05% N Stainless Steel (from Ref. 9, p. 226).

welding the nozzles to short lengths of pipe under maximum heat input conditions possible prior to normal hot and cold life testing or qualification testing. The area adjacent to the weld would then be metallurgically checked after testing for evidence of such corrosion. If necessary, further precautions against intergranular corrosion may be taken by either using a low-carbon type of stainless steel, the L series, that reduces but does not eliminate sensitization or by specifying that the valve material or the weld rod or both be "stabilized"; that is, that some titanium or niobium be added to preclude sensitization.

Pitting is the third type of corrosion of any significance in stainless steel valves. Since stainless steel produces its high resistance to corrosion by the mechanism of passivation, it follows that spots where the passive film breaks down would result in deep local corrosion. This is known as pitting. Pitting is not a common or an important problem in valve design. It is mentioned here because it has a tendency to be associated with crevices in the base material that are unavoidable in valve design. The occurrence of pitting requires specific chemical conditions of significant concentrations of Cl^-, Br^+, or $S_2O_3^{--}$. If any such ions are introduced inadvertently during cleaning or decontamination of the system, subsequent flushing may not remove them from the areas where they may do the most harm: the valve crevices.

The easiest way to avoid pitting is to provide proper chemical control, especially of high pH and low oxygen. From the standpoint of valve design, the avoidance of crevices through generous fillets, large-radius corners, and proper design of threaded mating parts will reduce the possibility of pitting and also provide the additional advantage of reducing the number and size of particle traps within the valve.

3.3.3 Structural Characteristics of Steels

A comparison of carbon steels and stainless steels shows that their yield strengths are about equal at room temperatures, and the low-alloy steels generally have a significantly higher yield strength than either the carbon steels or stainless steels. At the higher operating temperatures of a water-cooled reactor (500 to 600°F), the yield strength of the stainless steels is considerably less than that of the carbon steels. The low-alloy steels have the highest yield strength at these operating temperatures. Thus, in selecting valve material from a yield strength viewpoint only, the materials would be low-alloy steels, carbon steels, and stainless steels, in that order.

On the other hand, the austenitic stainless steels are less susceptible to fatigue than the carbon and low-alloy steels. Therefore, the cyclic stresses imposed on the valve pressure boundary by fluctuations in fluid pressure and temperature can be slightly larger for a given number of cycles if the material is austenitic stainless steel.

The austenitic stainless steels are also more ductile than the ferritic or carbon and low-alloy steels. But more important, the ferritic steels often suffer a marked loss of ductility or fracture toughness at the lower temperatures experienced at nuclear reactor power plant sites. The ferritic steel materials should be protected to prevent their exposure to service temperatures below 30°F, or they must be selected and tested to ensure adequate fracture toughness at the lowest service temperature. Austenitic stainless steels are not subject to this marked loss of fracture toughness at the lower temperatures (Ref. 12, page 551).

(a) Carbon and Low-Alloy Steels. The principal strength requirements of carbon steel and other low-alloy steels are given in Table 3.4.

Table 3.4. Strength Requirements of Carbon and Low-Alloy Steels

Type Steel	Specification	Minimum Tensile Strength (psi)	Minimum Yield Strength (psi)	Minimum Percentage of Elongation in 2 in.
Carbon (cast)	ASTM A216 Grade WCB	70,000	36,000	22
Carbon (forged)	ASTM A105 Grade II	70,000	36,000	22
Chrome-moly (cast)	ASTM A217 Grade WC6 or WC9	70,000	40,000	20
Chrome-moly (forged)	ASTM A182 Grade F11 or F22	70,000	40,000	20

Since the percentage of elongation is generally high, the amount of deformation before rupture can be expected to be large. This has been borne out in testing where rupture of carbon steel welds brought about by gross over-pressurization was preceded by extensive deformation.

The strength properties of carbon steel vary with temperature. Actual strength and operating temperature are important factors in design calculations, as discussed in Section 3.4. For this reason, the strength reduction of the material with temperature must be taken into account in cold hydrostatic testing of valves during production. For example, in calculating the production hydro-test pressure for the body-strength test, 150% of the system design pressure is not enough. This value should be increased by a factor such as S_u(test temperature)/S_u(design temperature) to compensate for the reduced strength of the material at design temperature. It should also be kept in mind that other forces applied for cold test purposes, such as for seating tests, may have to be increased by some factor for the same reason.

While the strength of the low-alloy steels is generally greater than that of carbon steel, the use of low-alloy steels may contribute to difficulties in welding. In Ref. 12, pages 545-546, it is stated that weld metals of Cr-Mo compositions have limited toughness in the as-welded condition.

> "Preheating is widely used to keep the deposits from cracking, and post-weld stress relief heat treatment ordinarily is applied to improve the toughness of the weld metal and the base metal heat-affected zones. Sometimes for weldments where post-weld heat treatment cannot be applied, a weld metal possessing more suitable mechanical properties in the as-welded condition may be applied. Austenitic Cr-Ni stainless steel or a nickel base alloy filler metal generally is selected, e.g., E309 or E310 stainless steels. While these dissimilar weld metals possess good strength and toughness, their use does not eliminate hardening of the weld heat-affected zones, and further consideration must be given to whether this structural condition will affect performance and service ... ."[12]

The text goes on to say that use of stainless steel filler for low-alloy steel welds poses problems similar to those posed by any bimetallic weld joint; that is, thermal cycling problems.

The conclusions to be drawn from this are as follows. For steam-plant applications, welding of chrome-moly valve bodies requires both pre- and post-heat treatment. Chrome-moly steels offer no better or no worse corrosion protection than unalloyed carbon steel, and for many reactor-system applications, the increased strength differential at high temperatures may be too small to warrant any gains in the reduction of the thickness of valve bodies or parts. These considerations are a significant factor in the extensive use of carbon steel in power piping.

(b) Stainless Steel. The principal strength and ductility values of the more common stainless steels are given in Table 3.5. In general, the austenitic types of stainless steel are not hardenable by heat treatment and do not have exceptional strength. As with carbon steel, the percentage of elongation of stainless steel is high and the amount of deformation before rupture can be expected to be large. The deformation and temperature considerations applicable to carbon steel valve body materials are also applicable for stainless steel materials. Combinations of stresses can result in body deformations which can severely reduce the leak tightness of the valve.

Table 3.5. Nominal Mechanical Properties of Standard Stainless Steels (from Ref. 10, p. 414)

Grade	Condition	Tensile Strength (psi)	0.2% Yield Strength (psi)	Elongation in 2 in. (%)	Reduction of Area (%)	Hardness Rockwell	Hardness Bhn
304	Annealed	85,000	35,000	55	65	B80	150
304L	Annealed	80,000	30,000	55	65	B76	140
316	Annealed Cold drawn bar[a] to 300,000	85,000	35,000	55	70	B80	150
316L	Annealed	78,000	30,000	55	65	B76	145

[a]Depending on size and amount of cold reduction.

The fatigue strength of stainless steels is affected by many factors such as directionality, temperature, surface conditions, micro-structure, and fabrication techniques. Cast bodies do not show the fatigue directionality or wrought bodies. In any case, engineering design for notch toughness and fatigue strength follows the generally accepted principles and is the same as for any pressure vessel designed to similar atandards. Since stress design and material metallurgy go hand-in-hand in producing a product with adequate resistance to failure, all metallurgical techniques that will assist in achieving this goal should be considered along with adequate stress design. It follows that any additional margin of assurance that can be obtained by good metallurgical practices should be considered. Additional cold working of wrought bodies or internal polishing of cast bodies may help to enhance fatigue properties if not depended upon alone to provide fatigue resistance.

3.3.4 Effects of Radiation on Valve Materials

Corrosion takes on a greatly increased significance in radioactive fluid systems. The corrosion products, more familiarly known as "crud", become radioactive. The corrosion products are transported by the fluid in the system to the reactor vessel where they are deposited on the core and become radioactive. Subsequently, portions of this radioactive crud may be swept away and redeposited throughout the system. Accumulation of this radioactive crud in pockets or crevices results in localized "hot spots" and adds to the general radiation problem. Besides contributing to the deposition of radioactive crud throughout the system by corrosion, the valves are themselves subject to radiation effects either directly from the reactor core or from crud traps within them where radioactive crud deposits accumulate.

Radiation effects on the metals used to fabricate valves are negligible and do not require special consideration for all locations outside the pressure vessel. The embrittlement of metals results from their exposure to fast neutrons, and the reactor vessel absorbs the majority of neutrons at these energy levels. Any small embrittling tendency of components external to the reactor vessel can be overridden by self-annealing at reactor plant operating temperatures.

On the other hand, some nonmetallic materials may experience degradation of properties because of radiation. Nonmetallic materials are used in valves for gaskets, packing, and sometimes for parts in the operator. This degradation does not constitute a significant problem as it is a relatively long-term process, and routine replacement of packing or other seals for maintenance purposes may nullify it. In addition, the valve itself provides shielding to the seal materials. In general, all organic materials are subject to deterioration caused by severing of some molecular cross links, resulting in tensile reduction, and by the formation of other cross links, resulting in brittleness. The results of empirical tests with some qualitative descriptions of the effects various chemical conditions have on the results are given in Ref. 13, page 184. For some high-temperature plastics, resistance to radiation damage is lessened by oxygen atmospheres. As previously noted, when evaluating a material, it is essential to review the characteristics of the material under all operating conditions.

3.3.5 Valve Body and Bonnet Materials

Carbon and low-alloy steels and stainless steels are used almost exclusively for valve pressure boundaries, bodies and bonnets, in large valves (4 in. and over) for radioactive fluid service.

(a) Carbon and Low-Alloy Steels. A long history of experience supports the use of carbon steel for components of high-pressure systems. Carbon steel has high strength at elevated temperatures, is easily fabricated, and is readily available. Although carbon steel is not corrosion resistant, corrosion can be reduced to a very low value by controlling the oxygen entrained in the process fluid. Paragraph 442 of the Draft ASME Code for Pumps and Valves for Nuclear Power requires that carbon steel valve pressure boundaries designed for long plant lifetimes have a certain amount of added material for corrosion allowance. The carbon and low-alloy steels commonly used for valve bodies, bonnets, and flanges are given in Table 3.6, and oxidation data for some of these steels are given in Table 3.1 (page 83). The information given in Table 3.1 indicates that only small allowances are required for many of these steels at temperatures of 1000°F, but many of these steels oxidize too rapidly at a temperature of 1150°F to be useful.

Table 3.6. Plain Carbon and Low-Alloy Steels Commonly Used for Pressure Retaining Components in Steam Generating Plants (from Ref. 10, p. 264)

Common name	ASTM Designation	Pertinent nominal composition, %	Usual maximum service temperature, °F
Cast carbon steel.	A216-53T, grade WCB	0.35 C max	850
Cast carbon steel.	A216-53T, grade WCA	0.25 C max	850
Forged carbon steel.	A105-55T, grades 1 and 11	0.35 C max	850
Cast chromium-molybdenum steel	A217-55, grade WC6	0.20 C max, 1.25 Cr, 0.50 Mo	1050
Cast chromium-molybdenum steel	A217-55, grade WC9	0.18 C max, 2.50 Cr, 1.00 Mo	1050
Cast carbon-molybdenum steel (a).	A217-55, grade WC1	0.25 C max, 0.50 Mo	850
Forged carbon-molybdenum steel (a)	A182-55T, grade F1	0.25 C, 0.50 Mo	850
Forged 1.25% chromium-molybdenum steel.	A182-55T, grade F11	0.15 C, 1.25 Cr, 0.50 Mo	1050
Forged 1% chromium-molybdenum steel. . .	A182-55T, grade F12	0.15 C, 1.00 Cr, 0.50 Mo	1050
Forged 2.25% chromium-molybdenum steel.	A182-55T, grade F22	0.15 C max, 2.25 Cr, 1.0 Mo	1050

(a) Carbon-molybdenum steel may graphitize in high-temperature service, particularly if it has been welded.

(b) Stainless Steel. The resistance of stainless steel valve bodies to erosion, fretting, wire drawing, and other forms of wear is quite good in a primary coolant environment. This is particularly important for flow surfaces; mechanical wear surfaces; and sealing surfaces, such as gasketed joints, formed by parts of the valve body. The adherent film on the surface is strong and thin, rooted to the metal, and forms quickly to repair damage. Cases are known of in nuclear plants where high-pressure throttling caused severe damage to stems and discs made from materials harder than stainless steel but did not erode the adjacent stainless steel body walls which were coated with the passive film. There is practically no comparison between the cavitation-erosion resistance of stainless steel and that of carbon steel or non-ferritic materials, as is illustrated in Fig. 3.42.

This means that for stainless steel valves, the flow control and part fitup design techniques need not be any better than for carbon steel valves. The design consideration that is the most delicate for success in minimizing erosion is flow velocity. If flow areas in the valve are subject to velocities likely to be two or more times the average flow velocity for such reasons as throttling, etc., the flow passages downstream should be wide and tapered as insurance against erosion.

Fretting caused by contact of moving parts, such as valve latches, check valve bearings, top or bottom disc guides, and other hardware in speciality valves, can be a source of significant body wear over a period of many cycles. The use of hard-facing, such as stellites, should be a standard practice unless satisfactory results can be proved by life testing. Where intermittent loose contact, as in gate valve disc guides, is permissible; some fretting and wear of the body material may be acceptable as long as no reduction in the thickness of the body pressure boundary can occur.

Wire drawing does not seem to occur in stainless steel valves. Micro-leakage around the backside of seats or at body joints is probably plugged by renewed passivation. Consequently, seat leaks which form in stainless steel valves are usually associated with a poor finish, material upset, or deformation caused by overstress. In principle, the same finish requirements or thread tolerance requirements used for carbon

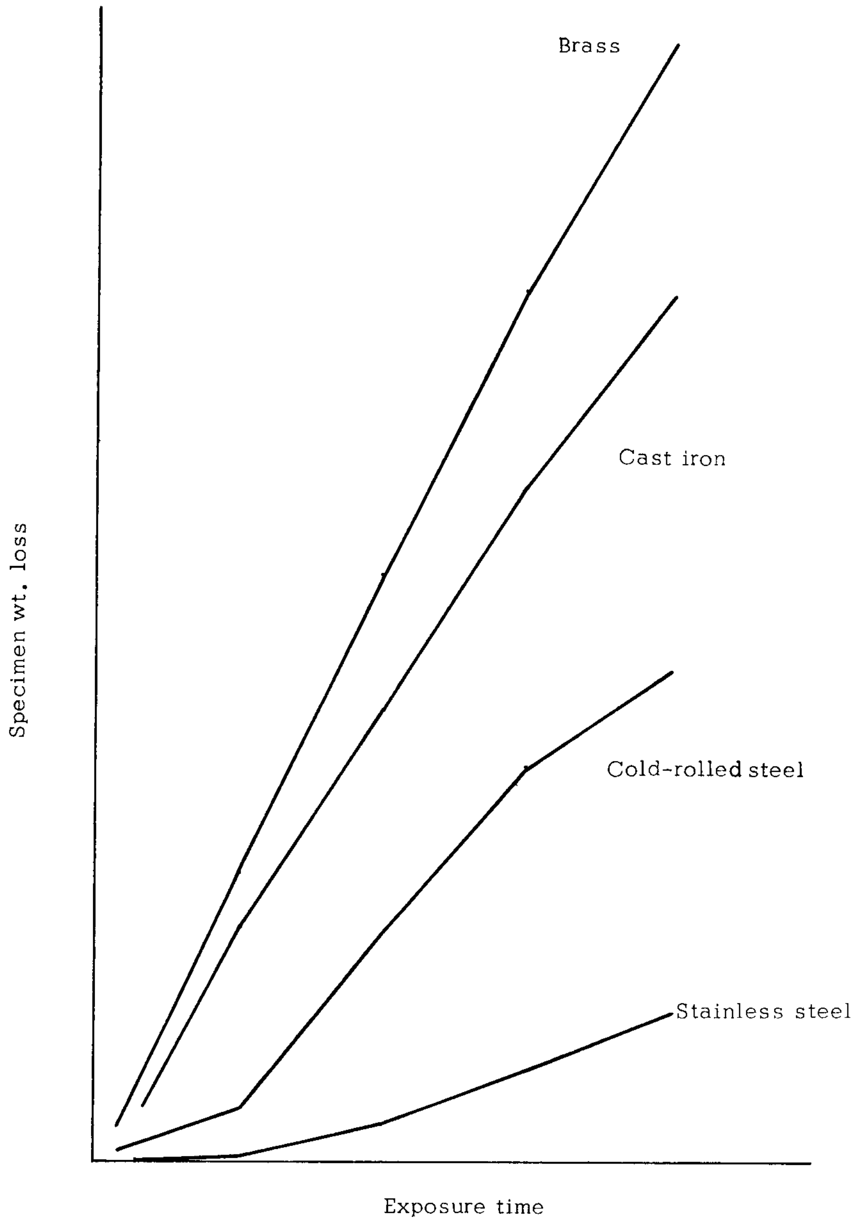


Fig. 3.42. Resistance of Metals to Cavitation-Erosion in Low-Pressure Water (from Ref. 9, p. 98).

steel pressure seal joints may be applied to stainless steel valves with good results.

3.3.6 Valve Trim Materials

Valve trim is comprised of those internal parts of the valve other than the body and bonnet that are wetted by system flow. This normally includes seats, stems, discs, and other miscellaneous hardware such as latches or guides sometimes used. All internal fasteners, such as fasteners for the disc and stem, are included. The valve trim pieces normally experience severe operating environments consisting of high flow erosion, mechanical wear, and thermal transients; and these pieces, especially the disc and seat, must maintain a high degree of integrity and dimensional stability. The various properties considered important in the selection of valve trim materials are discussed here.

Wetted internal parts of valves are normally made of at most three different materials. The body and bonnet are made of one material, the trim pieces are basically made of another material, and the hard-facing on several trim pieces constitutes the third material. Trim materials form a particularly sensitive area in the leak tightness of valve design, and it is in this area that the more exotic and expensive materials are usually used.

(a) Selection of Valve Trim Base Material. The most basic requirement of trim materials is corrosion resistance. Carbon steel is not used for trim pieces because it does not have good corrosion resistance properties. Flow-enhanced corrosion of carbon steel trim rapidly deteriorates the pieces and advances valve failure. Corrosion products also tend to cause binding and seizing of valve parts as well as interfering with good seat and disc contact for valve leak tightness. Therefore, most valves, including both carbon steel and stainless steel valves, have stainless steel trim.

Two basic types of corrosion should be checked when determining the basic corrosion resistance of trim materials. These are general corrosion and contact corrosion. (The penetrating types of corrosion, such as stress corrosion and intergranular corrosion, should also be checked as a matter of standard practice and should be compatible with the chemical conditions of the plant.) General surface corrosion should be controlled insofar as possible to reduce the possibility of forming a thick corrosion layer on the outside surfaces of valve trim materials. A material such as stainless steel, which forms a very thin passive film, is generally acceptable in this respect. If there is any doubt, basic corrosion-rate data for high-pressure flowing water systems can be obtained from several sources such as Ref. 10, pages 466 through 488; Ref. 14, pages 101 through 120; and Ref. 15, pages 626 through 631.

The other type of corrosion of general interest in the selection of trim material is crevice corrosion. Since all contact areas, gaps, indentations, and corners qualify as crevices, there are normally many locations in valves where trim crevice corrosion can occur. Corrosion can build up at the perimeter of contact areas, and this type of corrosion is caused by a metal-ion concentration cell. Pitting can occur in the area between two contact surfaces, and this type of corrosion is caused by an oxygen concentration cell. General surface or penetrating corrosion can occur in stagnant areas, and this is termed stagnant area corrosion. Metal-ion and stagnant area corrosion generally act to build up a corrosion layer and cause binding or seizing between pieces. Oxygen concentration corrosion and sometimes stagnant area corrosion act to cause penetrating pits or cracks in trim pieces, thereby weakening them.

As well as through material selection, there are several ways of avoiding crevice corrosion in valve trim. These include chemical control of the fluid and design control of geometry, especially of clearance between mating surfaces. A summary of test information, including coverage of all aspects of crevice corrosion, is given in Ref. 14, pages 147 through 185.

Valve trim material must also have high strength at operating temperatures. For the purposes of this document, it is noted here only that the higher the strength of the trim pieces, the smaller they need be made. For example, the stem must be strong enough to withstand high forces imposed on it to seat the valve and must not bend or warp. If the valve stem can be made of a smaller diameter, the size and weight of the valve can generally be reduced and advantages can be gained from missile impact considerations of flying stems. Because of transient stresses in the system, such as check valve slam or valve trip, the valve trim pieces must be strong enough to resist deformation and subsequent leakage.

However, high strength should not be attained at the expense of ductility. Highly hardenable martensitic stainless steels, for example, might be subject to shattering if used in a trip valve in the fully hardened condition. For applications involving severe environments, exotic nonferritic alloys are usually required. Nickel- and cobalt-base metal alloys have been used in such applications. Some of the more commonly used valve trim materials and their properties are given in Ref. 14.

(b) Valve Trim Hard-Facing Materials. Valve trim hard-facing materials must have high hardness and wear resistance characteristics. The two are not necessarily synonymous; some very hard materials in contact may have high wear rates. The lifetime of valve trim hard-facing materials is controlled primarily by their wear characteristics, and erosion is included under wear characteristics in this discussion. It should be noted that wear is a generic property which is influenced by other material properties. These other properties include corrosion resistance, temperature effects, mechanical considerations, and hardness coupled with the environment. The amount and type of lubrication present also affects wear resistance.

Corrosion properties of various hard-facing materials affect wear resistance. Some materials wear faster in oxygenated water than others, and this may have to be considered if slight out-of-tolerance chemical conditions are anticipated in the primary system. An additional side effect of corrosion is the release of abrasive corrosion products. If entrapped, such corrosion products can multiply the wear rate out of proportion to what is commonly expected. Temperature has a large effect on the wear resistance of valve hard-facing materials by reducing their hardness and by increasing the corrosion rate. Wear rates at a temperature of 200°F may bear no relationship to wear rates at a temperature of 500°F. Other mechanical considerations and lubrication effects are important to the in-service wear rate but are difficult to predict. The corrosion film on some materials may be effective in reducing the wear rate until the application of sufficiently concentrated forces breaks the film and causes an increase in the wear rate (Ref. 14, pp. 241-244).

From the standpoint of wear, good materials have emerged from valve in-service experience in most cases. Where better hard-facing wear properties are required, use can be made of the experience gained in the design, testing, and use of such materials for other components, such as pumps, mechanisms, and steam generators, in primary systems. The successfulness or unsuccessfulness of hard-facing materials is reflected by the reliability of all components, and cross comparisons can be made among the various types of components for support of material wear properties. Stellites found to be good canned-rotor water-lubricated pump bearings should also show up well in valve stem and disc wear tests. It appears that assurance of good wear properties can be obtained only by investigating the results of actual tests that closely duplicate the service conditions, including the use of actual hardness values which are being considered. Hardness often increases wear resistance but sometimes does not. Hardness cannot be relied upon completely as a measure of wear resistance. This is an important point and can be the source of serious mistakes in the selection of materials.

The hardness of hard-facing materials, as distinct from wear resistance, is important in its own right. In seat-disc contact theory, it is important that the mating surfaces be as hard as is practical. The capability of the valve to resist flow erosion seems to be directly coupled with the hardness of the hard-faced seat-disc mating surfaces. Investigation has shown that whatever other properties the hard-faced seating surfaces have in the way of corrosion and wear resistance, the surfaces will not be satisfactory for prolonged use if they are not hard.[16] The composition and hardness of some typical hard-facing materials are given in Table 3.7.

For high-pressure, high-temperature steam or water application, disc and seat hard-faced contact surfaces should have a Rockwell "C" hardness value of 35 or better. Such hard contact surfaces will improve valve seat leakage characteristics and will reduce the stem force required for seating. In addition, high hardness will reduce deformation caused by unusual stem forces and dirt or abrasive particles trapped during seating. In high-pressure relief valves where seat leakage problems can be particularly severe, it is generally considered that the mating surfaces of the seat and disc should be as hard as is metallurgically possible. It should also be remembered that it is high-temperature hardness that is

Table 3.7. Composition and Hardness of Typical Hard-Facing Materials (from Refs. 17 and 18)

Alloy	Percentage of Alloying Elements											Rockwell C Hardness
	Co	Ni	Si	Fe	Mn	Cr	Mo	W	C	B	Other	
Cobalt Base												
Haynes Stellite												
Alloy No. 1	Bal.					30.00		12.00	2.50			52-54
Alloy No. 6	Bal.					28.50		4.00	1.10			44
Alloy No. 6H	Bal.	1.50[a]		1.25[a]		29.50		5.50	1.30			37-46
Alloy No. 12	Bal.					29.00		8.00	1.35			47
Alloy No. 21	Bal.	2.75				27.00	5.00		0.25			35
Wallex No. 1	43.00[b]		1.25			30.00		12.50	2.25		6.00[a]	50-55
Wallex No. 6	55.00[b]		1.25			29.00		4.50	1.00		6.50[a]	39-44
Nickel Base												
Colmonoy No. 4		Bal.	2.25	2.50		10.00			0.45	2.00		35-40
Colmonoy No. 5		Bal.	3.75	4.25		11.50			0.65	2.50		45-50
Colmonoy No. 6		Bal.	4.25	4.75		13.50			0.75	3.00		56-61
Colmonoy No. 70		Bal.	3.25	3.75		11.50		16.00	0.55	2.50		50-55
Colmonoy No. 72		Bal.	3.60	3.50		13.00		12.00	0.75	2.70		58-63
Haynes Alloy												
No. 40		Bal.	4.00	4.00		15.00			0.65	3.50		57
No. 41		Bal.	3.50	3.00		12.00			0.35	2.50		51
No. 41N		Bal.	3.50	0.50[a]	0.25[a]	12.00			0.50	2.50		51

[a] Maxinum.

[b] Minimum.

required in valves for high-temperature service. Low-temperature hardness may not be important.

The hardness of various hard-facing materials at different temperatures is given in Table 3.8. The nickel and cobalt alloys have a higher resistance to temperature than ferrous alloys. The possibility of thermal fatigue cracking of the hard-facing as applied to the disc and seat is remote, especially if cobalt base alloys are used. Similar cobalt base alloys are used in hard-facing for combustion engine valves in heavy-duty use. In fact, cobalt base alloy is considered the best possible hard-facing material for aircraft engine exhaust valves (Ref. 10, pages 626-634).

Table 3.8. Hardness of Hard-Facing Materials at Different Temperatures (from Refs. 17 and 18)

Hard-Facing Material	Approximate Rockwell C Hardness at Temperatures of				
	70°F	400°F	600°F	800°F	1000°F
Cobalt Base					
Haynes Stellite alloy no. 1	53	52	51	49	44
Haynes Stellite alloy no. 6	39	36	33	31	30
Nickel Base					
Colmonoy no. 4	35	32	30q	29	24
Colmonoy no. 5	46	43	41	39	33
Colmonoy no. 6	54	51	49	46	38

The metallurgical process of "super-hardening" may assist in increasing the hardness of contact surfaces (Ref. 19, page 95). This is a process of localized compression of the surfaces beyond yield stress. Super-hardening occurs when compressive deformation takes place in hard materials, and it is dependent upon the material and the degree of deformation. It is a form of cold working applied only to the surface of the material. In this instance, it is expected at the contact area between the seat and the disc after compressive deformation results from seating the disc tightly. The amount of super-hardening possible appears to be limited, although this area does not appear to have been well investigated with respect to valve seat hard-facing materials.

Differential hardness between the seat and disc is not often made a design requirement. In most valves of current construction, the mating surface of the disc is just as hard as that of the seat because seat maintenance by lapping is usually possible. In spite of this, there is some possibility that reduced maintenance requirements may actually require reduced hardness of disc hard-facing to provide preferential wear at this location. Disc wear may be easier to fix than seat wear in valves used for radioactive service.

For hard-facing to work properly, it must be applied to a base metal. This is normally done by a weld process. In some cases, the hard-facing material and the base material may not be compatible. When the base metal is carbon steel, there are no restrictions on the selection of an alloy; but there are some restrictions with other base metals, including the stainless steels (Ref. 10, page 823). The compatibility of some base metals and facing materials is given in Table 3.9.

Table 3.9. Suitability of Hard-Facing Coatings For Various Base Materials (from Ref. 10, p. 823)

Base material to be hard faced	Facing alloys: Iron-base: To 20% alloy	Facing alloys: Iron-base: Above 20%	Facing alloys: Cobalt-base	Facing alloys: Nickel-base	Facing alloys: Tungsten carbide: *Inserts*	Facing alloys: Tungsten carbide: *Deposits*
Carbon steels:						
0.10 to 0.35% C	Yes	Yes	Yes	Yes	Yes	Yes
0.35 to 1.0% C	Yes (a) (b)	Yes (a) (b)	Yes (a) (b)	Yes (a) (b)	Yes (a) (b)	Yes (a) (b)
Low-alloy structural and constructional steels; 0.30% max C	Yes	Yes	Yes	Yes	Yes	Yes
Gray, malleable and nodular irons	Yes (b) (c)	Yes (b) (c)	Yes (b) (c)	Yes (b) (c)	Yes (b) (c)	Yes (b) (c)
Low-hardenability martensitic stainless (410, 403)	No	Yes (a) (f)	Yes (a) (f)	Yes (a) (f)	Yes (a) (f)	Yes (a) (f)
High-hardenability martensitic stainless (420, 440)	No	Yes (a) (c) (f)	Yes (a) (f)	Yes (a) (f)	Yes (a) (f)	Yes (a) (f)
Type 321 austenitic	No	Yes (c)	Yes (d)	Yes	Yes	Yes
Type 347 austenitic	No	Yes (c)	Yes	Yes	Yes	Yes
All other type 300 austenitic	No	Yes (c)	Yes	Yes	Yes	Yes
Monel	No	Yes (c)	Yes	Yes	Yes	Yes
Nickel	No	Yes (c)	Yes	Yes	Yes	Yes
13% Mn steels	Yes (e)	Yes (e)	Yes (e)	Yes (e)	No	Yes (e)

(a) Preheat. (b) Gas welding preferred. (c) For limited applications only. (d) Use type 347 interlayer. (e) Use nickel-base interlayer. (f) Post-isothermal anneal.

The hard-facing process requires the same precautions as any other welding process to prevent cracking of the base metal during heating and cooling.[10] Electric welding can be used to hard-face carbon steels with less than 0.28% carbon without preheating or postheating. Preheating and postheating are not required for atomic hydrogen welding methods when the carbon content is about 0.30%, nor are they required for oxy-acetylene welding when the carbon content is 0.35%. However, all base metals with a carbon content higher than 0.35% should usually be preheated and postheated to decrease brittleness in the base metal near the bond.[10]

Since the leak tightness of valves is directly affected by seat and disc corrosion, the corrosion resistance of the hard-facing material must be good. Even small amounts of corrosion can lead to uneven seat and disc contact and open up leakage paths. Even though the hard-facing material may have superior corrosion resistance, it should be remembered that fluid chemical conditions also affect the corrosion resistance of the materials. The general susceptibility of various hard-facing materials to general surface corrosion is given in Ref. 10, pages 466 through 488; Ref. 14, pages 95 through 120 and 171 through 185; and Ref. 15, pages 515 through 552 and 626 through 631. Crevice corrosion and contact corrosion of trim and hard-facing materials are discussed in Ref. 14, pages 171 through 172.

3.3.7 Valve Fastener Materials

This discussion of valve fastener materials is related primarily to the bolts, studs, and nuts used to fasten valve bonnets to valve bodies. Much of the information is also applicable to fasteners used in other locations on valves such as nozzles, flanges, and connections between operators and valve bonnets. Fasteners for valve body-to-bonnet flanges are particularly important because these connections are frequently opened for valve maintenance.

The selection of fastener material for pipe joints is in general a subject that is well covered in the technical literature. Pipe flanges have stress considerations that may require the use of special material for fasteners. Creep resistance is a special requirement for fasteners in high-temperature applications, and special tests, such as wedge tests, are usually specified for acceptance of fasteners. Detailed information on material selection, fabrication, and testing of fasteners for use in piping systems is given in Ref. 10, pages 174 through 178 and 604 through 611.

For the special application of fasteners in valve body-bonnet joints, there are several requirements over and above those prescribed for fasteners in piping systems. These requirements arise from the fact that body-to-bonnet joints may be subject to contact with a certain amount of weepage or leakage of system fluid, flange leakage (rare), or packing-gland leakage; and they include consideration of corrosion, galling, and relaxation resistance.

(a) Corrosion Resistance. Hot, oxygenated water can be present and if leachable chemicals can come out of the surrounding insulation, the exact nature of the corrosive environment may be difficult to determine. The fastener material selected for valve bonnet-to-body joints should have no susceptibility to any type of penetrating corrosion such as stress corrosion cracking. The inaccessibility of the fastener surfaces for chloride checking is pertinent here. In addition to reducing fastener strength, surface corrosion can coat the threads with abrasive corrosion products that can cause difficulty in removing nuts and false torque readings when tightening them. Many valve bonnet fasteners are in the form of studs which have been screwed into the body with an interference fit and are therefore not easily replaceable if thread corrosion has occurred. Material selection for such studs should provide an intended lifetime at least equal to that of the valve body. Information needed to insure the absence of valve bonnet stud corrosion problems can usually be obtained from material testing data or from experience with fasteners.

(b) Galling Resistance. A susceptibility to galling arises if non-hardenable fastener materials such as austenitic stainless steels are used. There are several lubrication and oxide layer film techniques for inhibiting galling, but actual cold-welding, which forms the basis for galling, of the materials is best reduced by providing a material difference between the pieces. Differences in material hardnesses and grain sizes can also be important. For instance, the bonnet ring of a pressure seal bonnet valve would be made of a harder material than stainless steel. Otherwise, cold-welding and galling may result. As a final point on fastener galling, the effect of soft metal plating applied to fastener contact surfaces is to cause galling. Chromium and nickel plating may reduce surface corrosion of fasteners, but they are subject to flaking under the influence of high contact stress. Flaked particles then become abrasive and rapidly freeze the pieces together.

(c) Relaxation Resistance. Relaxation and creep of fastener materials is covered adequately in Ref. 10 but need some amplification for use in inaccessible valve locations. Even at the high temperatures of nuclear reactor systems, it may be possible to design valve bonnet closures to preclude relaxation of studs over a period of time. This is especially important in the design of valves for reactor plants since these valves may be located in relatively inaccessible containment areas where maintenance cannot be performed without plant shutdown. Since gasketed flanged joints can leak after stud relaxation, it is generally true that such joints are undesirable for use within containment areas of high-temperature, high-pressure radioactive systems. Since relaxation is a long-term process which requires high temperatures, production testing of the fastener material is not possible and reliance should be placed on material testing information that is readily available.

3.3.8 Valve Seal Materials

The various materials used for valve pressure boundary (including stem) seals are discussed here. As is the case with valve structural materials, consideration of valve seal materials must go beyond a review of their ductility and resistance to corrosion. Additional factors to be considered include self-lubrication qualities, good resilience, low heat expansion, chemical inertness, and freedom from leachable chlorides or other halides (these act similarly to chlorides in causing stress corrosion). These characteristics must exist over the entire range of operating temperatures. Many elastomers used as seal materials undergo decomposition in high-temperature water. For example, Viton releases fluoride ions when exposed to water at temperatures above 400°F. Any halides released as a part of the decomposition can cause stress corrosion cracking of austenitic stainless steels. In brief, evaluation of a material is not complete unless the material has been checked under conditions equaling or exceeding those experienced in normal use.

The pressure boundary seals covered in this discussion of valve seal materials include gaskets, pressure ring seals, O-rings, and seal-weld rings. Stem seals are packing and bellows, but the design of bellows is covered in Section 4.7 and will not be discussed here. Further, the coverage here is limited insofar as practical to materials and corrosion aspects to avoid duplication of the information presented in Section 4.4.

(a) Gaskets. Gasket-type seals may be used in locations where absolute leak tightness is not a requirement and relative ease of closure opening for maintenance is desirable. Gaskets are usually of a soft material which is designed to increase microscopic contact of the seal area by plastic deformation. The softness also allows the material to flow to repair its own leaks, but this characteristic must be designed to balance out the natural tendency of soft material to break away and erode under micro-leakage conditions. Thus, gasket material must be soft enough to plug its own leaks and hard enough to resist being eroded away. This best explains the features of the Flexitallic gasket, which embodies both characteristics through the use of two materials: metallic and nonmetallic. Flexitallic gaskets are made of circular layers of teflon or asbestos trapped between layers of stainless steel. This arrangement is an attempt to achieve both the required softness and hardness characteristics.

A large selection of gasketing materials and geometries is available from the gasket market. In making a selection, the following points should be considered.

1. The gasket material should be sufficiently ductile or soft to flow, and upon compression, it should achieve near 100% surface-to-surface contact.

2. At the same time, the material should have enough strength to resist damage by internal pressure forces even in the event of small leakage and highly erosive weepage.

3. Since the gasket is in a crevice, good crevice corrosion resistance should be obtained.

4. Penetrating corrosion, such as chloride stress and intergranular and deep pitting, should be controlled by proper material selection and chemical control measures.

5. The material or materials used should retain these properties at operating temperatures.

(b) Pressure-Seal Rings. Pressure-seal rings are used in the pressure-seal bonnet-to-body joints. These rings consist of a soft metal core plated with a corrosion resistant plating, and they are subject to pressure forces (vice torqued bolt compression). The mechanical operation of such sealing devices is discussed in Section 4.4. From a material standpoint, the following requirements must be met.

1. The soft metal core, which is usually soft iron, must have high ductility and deformability, even to the point of sealing small leakage paths as they form.

2. The plating, usually silver, should be highly corrosion resistant, of sufficient depth to not be penetrated by scratches, and be ductile and deformable.

(c) O-Rings. O-rings embody an attempt to combine the simplicity of gaskets with an additional compressive force contributed by the pressure of the system fluid to aid in achieving leak tightness. O-rings may be of either metallic or elastomeric material. The metallic O-rings are generally stainless steel and are suitable for use in elevated pressures and temperatures. Seals of this type are used in a system fluid environment, and they usually have the pressure of the system fluid acting on them to increase the sealing force. They must have good ductility as well as good corrosion and abrasion resistance. The elastomer O-ring seals are synthetic rubbers used at lower temperatures and pressures but with the same sealing principles as the metallic seals. The elastomers must be checked at all operating conditions to insure their compatibility with the intended environment.

(d) Seal-Weld Rings. Seal-weld ring seals consist of a weld canopy usually of the same material as the valve body or bonnet to permit ease of seal-welding. The basic considerations in the use of such seals are that the material be weldable to the valve body or other closure and that it be sufficiently flexible to resist expansion or other deformation stresses.

(e) Stem Packing. There is a wide variety of stem packing material and geometries, and some important considerations in their use are as follows.

1. The packing material should be adequately service tested to insure that periodic re-tightening requirements are not excessive. The packing must have good resilience.

2. The packing should be self-lubricating so as not to require weepage of system fluid for lubrication and to avoid drying-out problems during disuse of the system.

3. The packing should be inert and not deteriorate under the temperature, chemical, and radiation conditions present.

4. The packing should have a low heat expansion.

3.3.9 Materials Commonly Used in Nuclear Valves

The material requirements for valves intended for service in nuclear reactor systems are described in preceding portions of this section, and those materials commonly used for the major components of nuclear valves are given in Table 3.10. From a materials standpoint, these valves are grouped in two categories: the stainless steel or corrosion-resistant valves and the carbon steel and low-alloy steel valves. The stainless steel valves are generally used for the fluid systems circulating reactor primary cooling water on a continuous basis and for the portions of other systems which would otherwise require increased maintenance but have restricted accessibility because of their location within a high radiation area or within the containment area. The carbon steel and low-alloy steel valves are used extensively in all other systems where the water chemistry can be controlled to restrict the corrosion rate.

Since the respective system conditions for which the valves and materials were intended cannot be described in this table, Table 3.10

Table 3.10. Materials Commonly Used in Nuclear Valves*

Component	Stainless Steel or Corrosion-Resistant Valves	Carbon Steel or Low Alloy Steel Valves
Body and Bonnet		
Castings	ASTM A 351 Gr. CF8 or CF8M	ASTM A 216 or Gr. WCB
Forgings	ASTM A 182 Gr. F304 or F316 ASTM A 336 Gr. F8 or F8M	ASTM A 105 Gr. II ASTM A 181 Gr. II ASTM A 182 Gr. F 1, F11, or F22
Stem	ASTM A 276 Gr. 316 or 410 17-4 PH Stainless Steel	ASTM A 182 Gr. F6 17-4 PH Stainless Steel
Disc or Wedge	ASTM A 351 Gr. CF8 or CF8M Hardfaced with Stellite 6**	ASTM A 216 Gr. WCB ASTM A 108 Gr. 1020 ASTM 217 Gr. WC9 All hardfaced with Stellite 6**
Seat Ring	ASTM A 312 Gr. TP 304 ASTM A 182 Gr. F 316 Both hardfaced with Stellite 6**	ASTM A 216 Gr. WCB ASTM A 106 Gr. B ASTM A 217 Gr. WC9 All hardfaced with Stellite 6**
Internal Fittings (Wetted Hardware)	ASTM A 351 Gr. CF 8 ASTM A 276 Gr. 410 or 416 ASTM A 193 Gr. B8 Inconel X750	ASTM A 108 Gr. 1020 ASTM A 276 Gr. 410, 416 or 304 17-4 PH Stainless Steel
Stem Packing	John Crane 187, 187-1, 187-ICR, 177-A1, C955	John Crane 187, 187-1, 187-ICR, 177-A1, C955
Gaskets	Spiral Wound SS 316 & Asbestos	Spiral Wound SS 316 & Asbestos
Bolts, Studs, and Nuts	ASTM A 193 Gr. B7 ASTM A 194 Gr. 2H	ASTM A 193 Gr. B7 ASTM A 194 Gr. 2H
Drain and Bypass Piping	ASTM A 312 Gr. TP 304 or TP 316	ASTM A 106 Gr. B

* The listing of a material in this table does not constitute a recommendation for any component-material combination.

** Stellite 6 appears to be the most common hardfacing. However, Stellite 1 or 3, and Colmonoy 4, 5 or 6 are often given as acceptable alternates.

does not constitute a recommendation for any of the listed component-material combinations. The final selection of a suitable material for a specific application is dependent upon the requirements of that application.

3.4 Structural Design and Analysis

A guide for the structural design and analysis of a complete valve assembly, including the pressure boundary, closure, flanges, and internals, is presented in this section to provide a comprehensive method for evaluating the structural adequacy of valves. The design philosophy and rules contained in Section III of the ASME Boiler and Pressure Vessel Code and in the ASME Code for Pumps and Valves for Nuclear Power have been followed wherever possible, and sample calculations are provided to illustrate the use of the design methods.

3.4.1 Valve Body Design

The design of the valve body should meet the requirements of Article 4 of the ASME Code for Pumps and Valves for Nuclear Power. Care should be taken to employ the latest revision of this code in the evaluation of a valve design. The aspects of valve body design covered in the design method given in Article 4 include

1. minimum wall thickness;
2. limitations on body shapes and contours;
3. primary membrane stresses caused by internal pressure;
4. secondary stresses caused by internal pressure, pipe reactions, and thermal effects;
5. fatigue requirements; and
6. cycling caused by temperature and pressure transients.

Familiarization with the rationale and techniques of Article 4 is necessary for a proper review of the sample stress analysis given in the following subsection.

3.4.2 Sample Stress Analysis of Valve Body Design

The sample calculation presented herein covers the design of a 4-in. valve body, including the body portion of the body-to-bonnet flanged joint. A layout of the valve body is shown in Fig. 3.43, and the flange

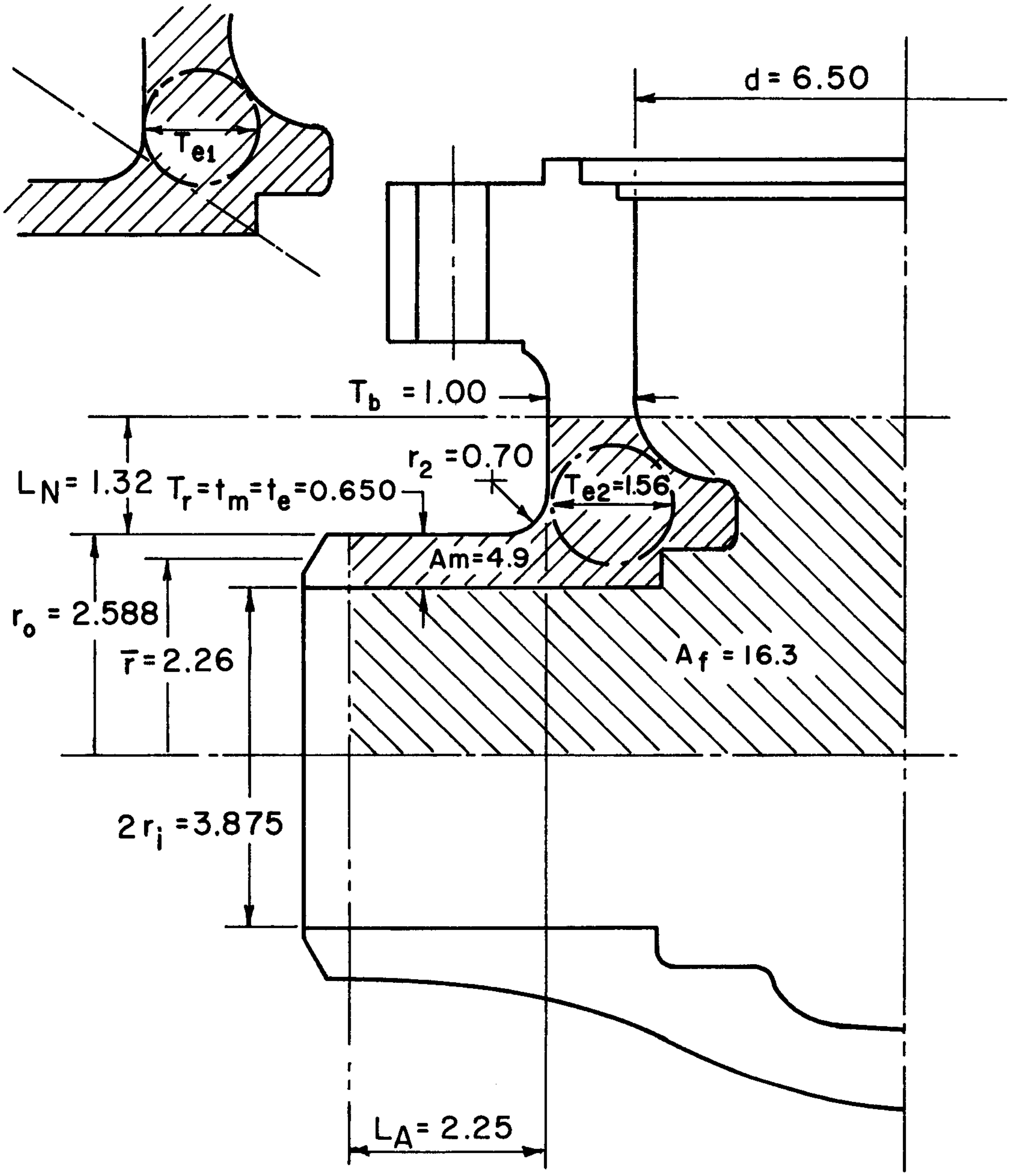


Fig. 3.43. Layout of Valve Body Analyzed in Sample Stress Analysis.

dimensions are illustrated in Fig. 3.44. The functional requirements of the valve analyzed are as follows.

Valve type:	Gate
Classification:	Class I

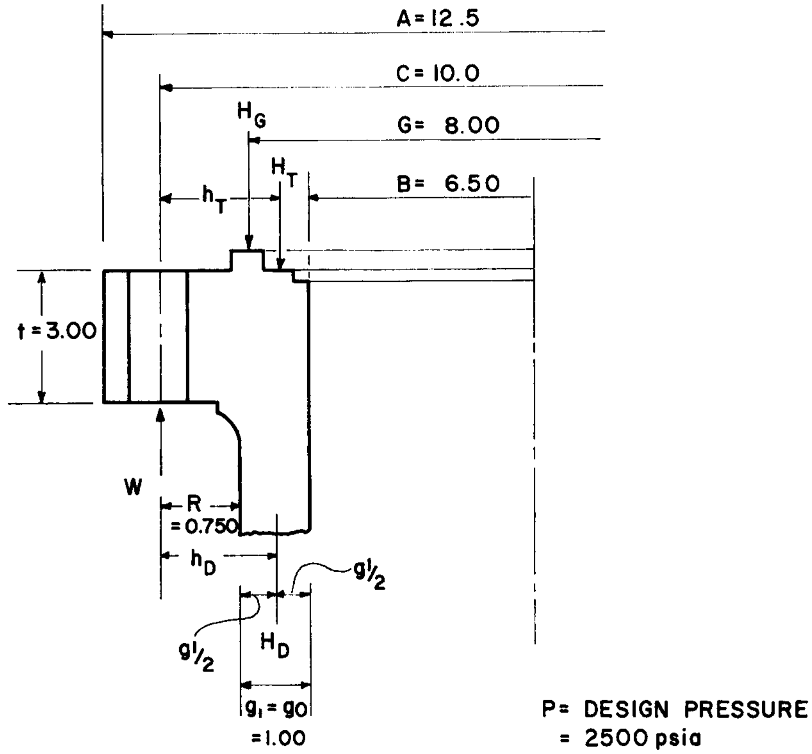


Fig. 3.44. Flange Dimensions for Valve Analyzed in Sample Stress Analysis.

Valve inlet ID:	3.875 in.
Body material:	Low-alloy steel, A-217 Grade WC1
Design pressure:	2500 psi
Design temperature:	400°F
Attached pipe:	4 in. Schedule 80
Cyclic duty:	(a) 150 start-ups and 150 shutdowns at 100°F/hr
	(b) 16 fluid temperature increases of 200°F from design conditions at 50°F/min
	(c) 40 fluid temperature reductions of 150°F from design conditions at 50°F/min
	(d) 400 fluid temperature reductions of 50°F, considered instantaneous

(a) Minimum Wall Thickness. The valve in question does not have a standard pressure rating. Therefore, the rules of Section 452.1b of the ASME Code for Pumps and Valves for Nuclear Power (Summer 1970 Addenda) should be followed. Entering Tables 451.5 and 451.6 of the Code with the appropriate material (carbon moly), the design pressure (p_d = 2500 psi), and the design temperature (400°F); the following information is obtained.

p_{r2} = 1500 lb,
p_{r1} = 900 lb,
p_2 = 3330 psi, and
p_1 = 2000 psi.

Entering Table 452.1 of the Code with the valve ID (taken as 4 in.), p_{r2} = 1500 lb, and p_{r1} = 900 lb; the thicknesses t_2 = 0.83 in. and t_1 = 0.51 in. are obtained. The minimum body wall thickness, t_m, is given by the following equation.

$$t_m = t_1 + \frac{p_d - p_1}{p_2 - p_1}(t_2 - t_1)$$

Thus, for this analysis,

$$t_m = 0.51 + \frac{500}{1330}(0.32)$$
$$= 0.63 \text{ in.}$$

The actual wall thickness of the body is 0.650 in., which is acceptable. The primary pressure rating, p_r, corresponding to design conditions is given by the following equation.

$$p_r = p_{r1} + \frac{p_d - p_1}{p_2 - p_1}(p_{r2} - p_{r1})$$

For the valve analyzed,

$$p_r = 900 + \frac{500}{1330}(600)$$
$$= 1128 \text{ psi} .$$

(b) Body Shape. Certain restrictions are placed on the shape and transitions of valve bodies in Section 452.2 of the ASME Code for Pumps and Valves for Nuclear Power to assure that they fall within the scope of the stress analysis model used. The effects of these restrictions on the valve in question are as follows.

1. The fillet at the neck-to-body junction has a radius greater than $0.3t_m$ as required by the Code.

2. Sharp fillets have been used to form the ring grooves. However, these grooves are isolated from the major primary and secondary stresses of the body.

3. There are no penetrations of the pressure boundary other than the neck intersection.

4. There are no lugs or protuberances on the pressure retaining boundary.

5. Out-of-roundness of the valve body is within the 5% allowed by the Code.

6. The valve has no flat or doubly curved sections.

(c) Primary Membrane Stress Caused By Internal Pressure. The method of determining the primary membrane stress in valve bodies is given in Section 452.3 of the ASME Code for Pumps and Valves for Nuclear Power. For the valve being analyzed, enter Figure 452.3a of the Code with the primary pressure rating (p_r = 1128 psi) to obtain p_s = 2400 psi. From Fig. 3.43 (page 117), the distances limiting the fluid area, A_f, and the metal area, A_m, are as follows.

Bonnet ID (d):	6.50 in.
Bonnet thickness (T_b):	1.00 in.
Body thickness (T_r):	0.650 in.
External fillet radius (r_2):	0.70 in.

The distance (from Fig. 3.43)

$$L_A = \frac{6.50}{2} - 1.00$$
$$= 2.25 \text{ in.}$$

The distance (from Fig. 3.43)

$$L_N = 0.5(0.70) + 0.354[1.00(6.50 + 1.00)]^{1/2}$$
$$= 1.32 \text{ in.}$$

The metal area A_m and the fluid area A_f are determined by some appropriate method such as by a planimeter or by "counting squares".

$$A_m = 4.9 \text{ in.}^2$$
$$A_f = 16.3 \text{ in.}^2$$

The general primary membrane stress, P_m, in the crotch of the valve body is determined as follows.

$$P_m = \left(\frac{16.3}{4.9} + 0.5\right)(2400) = 9120 \text{ psi} \; .$$

This is acceptable because the criterion is that $P_m \leq S_m$ for the valve body material at a temperature of 500°F. This is given as $S_m = 20{,}200$ psi in Table A.1 of the Code. It should be noted that this Code section specifies the comparison of P_m with S_m at a temperature of 500°F rather than at the design temperature of 400°F.

(d) Secondary Stresses in Valve Body. The methods for determining secondary stresses in valve bodies are given in Section 452.4 of the ASME Code for Pumps and Valves for Nuclear Power. From Section 452.4a for determination of pressure, the following dimensions are noted from Fig. 3.43 (page 117).

Inside radius of body at crotch (r_i) = 1.938 in.

Body thickness at crotch (t_e) = 0.650 in.

Therefore,

$$Q_p = 3\left(\frac{1.938}{0.650} + 0.5\right)(2400) = 25{,}000 \text{ psi} \; .$$

From Section 452.4b of the Code for determination of the pipe reaction, the axial load effect is determined from the following equation.

$$P_{ed} = \frac{F_d S}{G_d} \; .$$

From Figure 452.4b(2) of the Code, $F_d = 3.9$ in.^{2} Since the material of the connected pipe was not specified, a value of $S = 30{,}000$ psi was selected.

$$G_d = \frac{\pi}{4}\left[(5.175)^2 - (3.875)^2\right] = 9.24 \text{ in.}^2$$

Therefore,

$$P_{ed} = \frac{3.9(30{,}000)}{9.24} = 12{,}660 \text{ psi} \; .$$

The bending effect is determined from the following equation.

$$P_{eb} = C_b\left(\frac{F_b S}{G_b}\right) \quad .$$

From Fig. 452.4b(4) of the Code, $F_b = 7.6$ in.^{3}; and from Fig. 452.4b(6) of the Code, $C_b = 1.0$. The valve body is circular in cross section at the crotch. As such, its bending modulus is

$$G_b = \left(\frac{\pi}{4}\right)\frac{(2.588)^4 - (1.938)^4}{2.588} = 9.35 \text{ in.}^3$$

Therefore,

$$P_{eb} = \frac{1.0(7.6)(30,000)}{9.35} = 24,400 \text{ psi} \quad .$$

In determining the torsional effect, $G_t = 2G_b$ and $F_t = 2F_b$ for a circular cross section and $P_{et} = P_{eb} = 24,400$ psi. Since P_{ed}, P_{eb}, and P_{et} are each less than $1.5S_m$ for the valve body material at a temperature of 500°F ($1.5S_m = 30,300$ psi), the requirement of Section 452.4b of the Code is satisfied. The selected value of $P_e = P_{eb} = 24,400$ psi.

From Section 452.4c of the Code for determination of thermal secondary stress, the dimensions $T_{e2} = 1.56$ in. and $T_{e1} = 1.35$ in. are noted from Fig. 3.43 (page 117). Thus,

$$\frac{T_{e1}}{t_e} = \frac{1.35}{0.650} = 2.08 \quad ,$$

$$\frac{T_{e2}}{t_e} = \frac{1.56}{0.650} = 2.40 \quad , \text{ and}$$

$$\frac{\bar{r}}{t_e} = \frac{2.26}{0.650} = 3.48 \quad .$$

From Figure 452.4c(3) of the Code, $Q_{T1} = 100$ psi; from Figure 452.4c(4) of the Code, $C_2 = 0.40$; and from Figure 452.4c(5) of the Code, $\Delta T_2 = 1.0$ °F. Therefore,

$$Q_{T_2} = 220(0.40)(1.0) = 88 \text{ psi}$$

and

$$Q_{T_1} = 100 \text{ psi} \quad .$$

Thus,

$$Q_T = 100 + 88 = 188 \text{ psi} \quad .$$

Based on start-up—shutdown, the total primary plus secondary stress is

$$S_n = 25{,}000 + 24{,}400 + 376 = 49{,}776 \text{ psi} .$$

Since S_n is less than $3S_n$ for the valve body material at a temperature of 500°F ($3S_n$ = 60,600 psi), the criterion of Section 452.4 of the Code is satisfied.

(e) Fatigue. The method for determining fatigue in the valve body is given in Section 452.5 of the ASME Code for Pumps and Valves for Nuclear Power.

$$S_{p1} = \frac{2}{3}(25{,}000) + \frac{24{,}400}{2} + 88 + 1.3(100)$$

$$= 29{,}068 \text{ psi} .$$

$$S_{p2} = 0.4(25{,}000) + \frac{2[24{,}400 + 2(88)]}{2}$$

$$= 34{,}566 \text{ psi} .$$

Entering Figure 452.5(a) of the Code with $S_p = S_{p2}$ = 34,566 psi, Na is approximately 1.6 x 10^4 cycles. This is greater than the 2000 cycles required by Section 452.5 of the Code.

(f) Cyclic Rating. There are several thermal cycles which could produce significant stresses. These cycles must be superposed to provide the most adverse temperature range, and all temperature changes are assumed to occur instantaneously.

N_{ri}	ΔT_{fi} (°F)	N_i	N_{ri}/N_i
16	350	60	0.27
24	150	700	0.03
360	50	1350	0.30
		I_t =	0.60

Since $I_t = 0.60 < 1.0$, the valve is adequate for the specified thermal cyclic loading. Thus, the valve body meets all the requirements and is considered acceptable for the specified application.

(g) Bonnet-to-Body Flange Stresses Considering Seismic Forces. Utilizing Paragraphs UA-47, 48, 49, 50, and 51 of Section VIII of the ASME Boiler and Pressure Vessel Code as required by the ANSI Standard Code for Pressure Piping ANSI B31.7, Nuclear Power Piping; the stresses for the flange illustrated in Fig. 3.44 (page 118) are determined as follows.

$$P = P_{design} + P_{eq}$$

$$= P_{design} + \frac{16M}{\pi G^3} + \frac{4F_s}{\pi G^2} + \frac{4F_R}{\pi G^2} ,$$

where

P_d = 2500 psig,

M = longitudinal bending moment caused by seismic force acting on the operator in a direction parallel to the flow axis

= 1 g(400 lb)(25 in.) = 10,000 in.-lb,

G = 8 in. from Fig. 3.44 (page 118),

F_s = longitudinal force acting on the bonnet caused by the seismic acceleration in the vertical direction

= 0.1 g(400 lb) = 40 lb,

F_R = longitudinal force acting on the bonnet from the reaction to the maximum stem-closing force

= 50,000 lb .

$$P = 2500 + \frac{16(10,000)}{\pi(8.0)^3} + \frac{4(40)}{\pi(8.0)^2} + \frac{4(50,000)}{\pi(8.0)^2}$$

$$= 2500 + 99 + 0.8 + 995$$

$$= 3595 \text{ psi} .$$

Using the dimensions shown in Fig. 3.44 (page 118) and the graphs and nomenclature of Section VIII of the ASME Boiler and Pressure Vessel Code, the longitudinal hub stress, the radial flange stress, and the tangential flange stress are obtained. The longitudinal hub stress, S_H, is determined from the following equation.

$$S_H = \frac{fM_o}{Lg_1^2(B + g_1)} + \frac{PB}{2g_1} ,$$

where

f = 1 for this flange design,

$M_o = M_D + M_T + M_G$,

$L = \frac{te + 1}{T} + \frac{t^3}{d}$,

P = 3595 psi,

$B = 6.50$ in. from Fig. 3.44 (page 118), and

$g_1 = 1.00$ in. from Fig. 3.44 .

$$M_o = M_D + M_T + M_G$$
$$= H_D h_D + H_T h_T + H_G h_G ,$$

where

$H_D = 0.785B^2P$

$= 0.785(6.5)^2(3595) = 119{,}233$ lb;

$h_D = R + 0.5g_1$

where $R = 0.750$ in. from Fig. 3.44,

$= 0.750 + 0.5(1.00) = 1.250$ in.;

$H_T = 0.785P(G^2 - B^2)$

$= 0.785(3595)[(8.0)^2 - (6.5)^2] = 61{,}380$ lb;

$$h_T = \frac{R + g_1 + \frac{C - G}{2}}{2}$$

where $C = 10.0$ in. from Fig. 3.44,

$$= \frac{0.750 + 1.00 + \left(\frac{10.0 - 8.0}{2}\right)}{2} = 1.375 \text{ in.};$$

$H_G = 6.28bGmP$

$= 6.28(0.25)(8.0)(3.0)(3595) = 135{,}460$ lb; and

$$h_G = \frac{C - G}{2} = \frac{10.0 - 8.0}{2} = 1.0 \text{ in.}$$

$$M_o = 119{,}233(1.250) + 61{,}380(1.375) + 135{,}460(1.0)$$
$$= 368{,}899 \text{ in.-lb} .$$

$$L = \frac{te + 1}{T} + \frac{t^3}{d} ,$$

where

$t = 3.00$ in. from Fig. 3.44 (page 118);

$$e = \frac{F}{(Bg_1)^{1/2}}$$

where F = form factor from graph UA-51.2 = 0.909;

T = form factor from graph UA-51.5 = 1.5; and

$$d = \frac{U}{V}(Bg_1)^{1/2}(g_1)^2$$

where U = form factor from graph UA-51.1 = 3.6

V = form factor from graph UA-51.3 = 0.550 .

$$L = \frac{3.0\left[\dfrac{0.909}{[6.5(1.0)]^{1/2}}\right] + 1}{1.5} + \frac{(3.0)^3}{\dfrac{3.6}{0.550}[6.5(1.0)]^{1/2}(1.0)^2}$$

$$= 1.380 + 1.618 = 2.998 \ .$$

Therefore,

$$S_H = \frac{fM_o}{Lg_1^2(B + g_1)} + \frac{PB}{2g_1}$$

$$= \frac{1.0(368{,}899)}{2.998(1.0)^2(7.5)} + \frac{3595(6.5)}{2(1.0)}$$

$$= 16{,}406 + 11{,}684 = 28{,}090 \text{ psi} \ .$$

This is acceptable because the criterion is that $S_H \leq 1.5S_m$ for the valve body material at a temperature of 500°F ($1.5S_m$ = 30,300 psi).

The radial stress, S_R, in the flange is determined from the following equation.

$$S_R = \frac{(1.33te + 1)}{Lt^2B}(M_o) \ .$$

Using the values derived to determine the longitudinal hub stress,

$$S_R = \frac{[1.33(3.0)(0.3565) + 1]}{2.998(3.0)^2(6.5)}(368{,}899)$$

$$= 5095 \text{ psi} \ .$$

This is acceptable because the criterion is that $S_R \leq 1.5S_m$ for the valve body material at a temperature of 500°F ($1.5S_m$ = 30,300 psi).

The tangential stress, S_T, in the flange is determined from the following equation.

$$S_T = \frac{YM_o}{t^2B} - ZS_R \ ,$$

where

Y = form factor from graph UA-51.1 = 3.15

Z = form factor from graph UA-51.1 = 1.8 .

$$S_T = \frac{3.15(368,899)}{(3.0)^2(6.5)} - 1.8(5095)$$

$$= 19,864 - 9171 = 10,693 \text{ psi} .$$

This is acceptable because the criterion is that $S_T \leq 1.5S_m$ for the valve body material at a temperature of 500°F ($1.5S_m$ = 30,300 psi).

3.4.3 Experimental Stress Analysis

A valve body design that is found to be marginally inadequate by using the simplified stress analysis method described in Subsection 3.4.2 may be shown to meet the design requirements of Section III of the ASME Boiler and Pressure Vessel Code by means of more detailed experimental stress analysis techniques. The rules governing experimental stress analyses given in Article I-10 of Section III of the ASME Boiler and Pressure Vessel Code should be followed. Some of the more important rules are mentioned in the following paragraphs.

(a) Sufficient locations on the valve shall be investigated to insure that measurements are taken at the most critical areas. The location of the critical areas and the optimum orientation of test gages may be determined by a brittle-coating test.

(b) Permissible types of tests for the determination of governing stresses are strain-measurement tests and photoelastic tests. Brittle-coating tests may be used only for the purpose described in (a) above. The results of displacement-measurement tests and tests to destruction are not acceptable.

(c) Strain gage data may be obtained from the actual valve or from a model valve of any scale that meets the strain gage length requirements of Article I-10. The model material need not be the same as the vessel material but shall have an elastic modulus which is either known or has been measured at the test conditions. The requirements of dimensional similitude shall be met as nearly as possible.

(d) Internal pressure or mechanical loads shall be applied in such increments that the variation of strain with load can be plotted to establish the ratio of stress to load in the elastic range.

(e) The experimental results obtained shall be interpreted on an elastic basis to determine the stresses corresponding to the design loads. That is, in the evaluation of stresses from strain gage data, the calculations shall be performed under the assumption that the material is elastic.

(f) The extent of experimental stress analysis performed shall be sufficient to determine the governing stresses for which design values are unavailable. When possible, combined analytical and experimental methods shall be used to distinguish between primary, secondary, and local stresses so that each combination of categories can be controlled by the applicable stress limit.

Stresses obtained from the experimental analysis should be interpreted in accordance with Article 4 (Design) of Section III of the ASME Boiler and Pressure Vessel Code. The rules of this article should also be followed in determining allowable stresses and performing fatigue analyses.

3.4.4 Other Design Requirements

Rules for the design of those portions of a valve not covered in Subsection 3.4.1 are provided here. However, these rules are not intended to replace in any way or limit the applicability of the stress analysis method described in Subsection 3.4.2. The rules cover reinforcement of openings, valve internals, and shock and vibration.

(a) Reinforcement of Openings. The use of penetrations in a valve body should be avoided because these penetrations increase the magnitude of peak stresses. When penetrations are used, they should not be located in regions of high stress such as the crotch area or sidewalls of the body. Openings in the valve pressure boundary should be reinforced in accordance with the rules given in Section N-450 of Section III of the ASME Boiler and Pressure Vessel Code. These rules require that reinforcement be provided in an amount equivalent to the cross-sectional area of metal removed from the valve to make the penetration.

(b) Valve Internals. In general, valve internals (seat, seat ring, stem, etc.) should be designed to the same basis as the valve body. For example, this means that the seat and disc should provide an effective pressure boundary if the valve should be pressurized on one side while in the closed position. Because of the variety of seat designs and the inaccuracy inherent in analyzing stresses in valve internals, no attempt has been made to set down specific formulae. The valve vendor should perform suitably conservative analyses of his design where possible, and in addition, the following general rules should be followed.

1. Seating structures that are an integral extension of the pressure retaining boundary, such as weld inlays or welded seat rings, should be designed with adequate flexibility to assure resistance to failure by thermal cycling compatible with the cycle life requirements of Subsection 3.4.2(f).

2. The thickness of the valve bridge wall should not be less than t_m, as specified in Subsection 3.4.2(a).

3. The primary stresses in the valve bridge wall should be less than S_m at design temperature and pressure. This should be demonstrated by means of hydrostatic testing at 1.5 times design pressure. The hydrostatic test pressure should be applied alternately, both above and below the seat. The valve should be seated with a torque no greater than the design value of seating torque recommended by the valve manufacturer. Seat leakage should be within the design value. Note that the seat leakage test as used to determine bridge wall adequacy does not include the effect of pipe reactions. If the requirements of Subsection 3.4.2(d) are met, distortion of the valve body will be negligible in actual service. Hence, loading of the bridge wall caused by bending of the valve body may be neglected.

4. Failure of valve internals caused by overtorquing of the valve operator or handwheel is not normally of concern because of the conservative design of these structures relative to the forces which might be developed by the stem screw mechanism. In any case, a recommended value of seating torque should be obtained from the valve manufacturer, and to prevent seat damage, this torque value should not be exceeded in normal operation.

(c) Shock and Vibration. Shock and vibration loads imposed by valve actuation will vary with each vendor's design and with the particular function that a valve fulfills in a system. For example, shock loading generated by slamming of a 6-in. check valve in a particular application will vary as a function of the design of valve internals. The same valve will also react differently when used in another system application. As a result, it is not possible to provide a meaningful uniform analytical technique for predicting or establishing the capability of a given valve design to resist damage caused by valve actuation. Accordingly, the required duty of a valve with respect to actuation and internal shock should be stated in the purchase specification. The purchase specification should require the vendor to demonstrate valve adequacy for a given number of actuations by either life cycle testing of the valve or by a suitable short-term strain gage test accompanied with an analytical extrapolation for the life of the valve.

3.5 Fabrication

Complex body shapes are generally required for valves because of their dual requirements for serving as a pressure boundary and housing internal operable parts. However, valves are unique with respect to the large influence that fabrication requirements have on the general configuration and shape of the valve body. Thus, the final design for a valve must represent to some degree a compromise among mechanical design, flow design, and fabrication requirements.

Because of their complex shape requirements, valve bodies are made by either casting or hot-forging methods. The choice between these two fabrication methods has historically been determined by shop capability and economics. Casting imposes fewer restrictions on valve cavity shapes, and this permits more freedom in the disc and seat arrangement and flow design. Casting patterns are also significantly cheaper than forging dies. Casting therefore favors a limited production run. On the other hand, forgings are generally more uniform and structurally superior. In a large production run, forgings are competitive with cast valve bodies.

Both casting and forging are rough fabrication methods, and the valve bodies require finish machining when made by either method. Finishing costs are about equal for valve bodies made by casting or forging.

The larger valves required by the nuclear industry are presently castings. The lower range of valve sizes, from 4 to 12 in. nominal pipe size, may be either castings or forgings, with the percentage of forgings increasing in the smaller sizes. The valve fittings and trim are amenable to standard fabrication methods and have little influence on the ultimate shape or configuration of the valve. Fabrication of these valve components is therefore not discussed here.

3.5.1 Casting

Casting is defined as the introduction of molten metal into a mold to produce an object of the desired shape. Casting is particularly suited to the fabrication of valve bodies because it is generally the most efficient and economic method of forming metal into the complex shapes required by the functional objectives of the valve. The valve designer must necessarily concern himself with casting technology and methodology to some degree to properly select and specify the casting materials and procedures that will assure a valve body structure of suitable strength and integrity. The physical design of the valve body, the design of the mold pattern, the design and construction of the mold, the foundry methodology, and the melt material all have considerable influence over the properties of the resultant casting. Even if the designer invokes strict quality assessment requirements and limitations on the "as-cast" valve body, it may be impossible or impractical for the supplier to meet these requirements if the designer has not observed certain basic casting requirements in the development of the valve body design.

Since it is beyond the scope of this document to provide the designer with the complete knowledge prerequisite to the appropriate design of a cast valve body, the information that is presented is intended to assist the designer in the recognition of potential problem areas involved in the design of cast valve bodies and to render him sufficiently conversant to communicate his requirements to the casting suppliers. This information should also serve to alert a designer to those problems he might anticipate in the selection of an off-the-shelf cast valve body. The information presented here is limited primarily to those materials permitted for use in nuclear valves in accordance with ASME standards.

Unless the valve designer has a good metallurgical background plus considerable knowledge of foundry practices, the best assurance for obtaining a satisfactory casting is close cooperation with the patternmaker and the foundryman during the early stages of the valve body design. This assurance can then be supplemented by an appropriate quality assurance program.

(a) The Casting Process. The more important casting problems of concern to the valve designer are (1) shrinkage, (2) segregation, (3) gas porosity, and (4) low hot strength. These are all generally avoided by controlled solidification of the molten metal, but each problem is worthy of individual discussion.

Shrinkage in molten metal occurs in three stages: liquid shrinkage, solidification shrinkage, and solid-state contraction. Liquid shrinkage occurs because the molten metal has a certain amount of superheat when it is introduced into the mold. For steel, this superheat usually amounts to about 100°F and will result in about 0.9% shrinkage per 100°F for 0.35% carbon steel. This shrinkage is compensated by additional melt supply, such as that from the risers provided in a conventional sand mold. Solidification shrinkage in steel is in the order of 3 to 4% and must be compensated by additional melt or cavities will result. Here, compensation is more of a problem and necessitates control of solidification to provide flow channels for the compensating melt. Solid-state contraction is unavoidable, amounts to about 7% in carbon steel, and must be compensated by the patternmaker's oversized dimensioning.

Before discussing the other problems of casting, some basic knowledge of solidification is necessary. When pure metals solidify in a mold cavity, they form a layer which progresses inward from the wall of the mold cavity until the entire casting is solidified. With alloys, however, this process may be more complex. Some alloys, such as cast iron, freeze or exhibit low fluidity over a wide temperature range, and this results in the formation of small solid crystals throughout the melt. These crystallites grow in a tree-like form and are called dendrites, the Greek word for tree. Other alloys, such as low-alloy steel, freeze over a relatively narrow temperature range, and the dendrites form initially at the outer portion and develop progressively ahead of the advancing solidification front. The chemical and structural heterogeneities that occur in dendritic solidification influence the properties of the casting significantly and must be controlled to achieve the desired properties.

Segregation in a casting refers to the chemical separation of the melt brought about by the freezing process. In gross form, this is often referred to as coring, wherein the initial crystals that freeze adjacent to the mold walls are lowest in alloy content and the core with the resultant high alloy concentrations bears little resemblence to the intended composition. Micro-segregation occurs when this same process takes place within each of the dendrite arms, and it is the more typical problem. In large grain castings, this can result in significant reduction in the desired properties, including those of strength and corrosion. More rapid solidification of the melt by heat removal or use of thinner sections can often promote small grain structure and reduce the overall effects, or postheat treatment can sometimes be used to promote granular diffusion.

Another important form of micro-segregation is the precipitation of a second phase, often as an intermetallic or nonmetallic compound. These precipitates appear in the form of inclusions and are of obvious disadvantage. For example, small quantities of silicon are generally added to cast steel to prevent gas pore formation, and as solidification proceeds, oxygen is rejected by the growing iron dendrites and combines with the silicon instead of forming gas pores. But the oxygen and silicon combine to form SiO_2 or small glass inclusions in the dendrite arms. Although these may be less detrimental than gas pores, they are nonetheless undesirable. The rate of solidification can be used to reduce the size of or sometimes eliminate such inclusions.

Gas porosity results from gases that are dissolved in the metal in the molten state. As the metal cools it becomes supersaturated, causing the gases to come out of solution. They are trapped between the dendrite arms and form tiny voids known as micropores or pinholes. This condition can be largely eliminated by proper degasification of the melt prior to pouring.

Another form of porosity, and one more difficult to eliminate, occurs in alloys with a wide freezing range and is caused by solidification shrinkage. As the dendrites solidify and shrink in volume, replacement of melt must flow from a riser along a tortuous path of intervening dendrites (dendrites form simultaneously throughout an alloy with a wide freezing range), and the resistance to flow may be too great in the latter stages of solidification. The result is microvoids or microporosity very similar to that caused by trapped gases. The heterogeneities that occur in the dendritic structure of cast alloys are illustrated in Fig. 3.45.

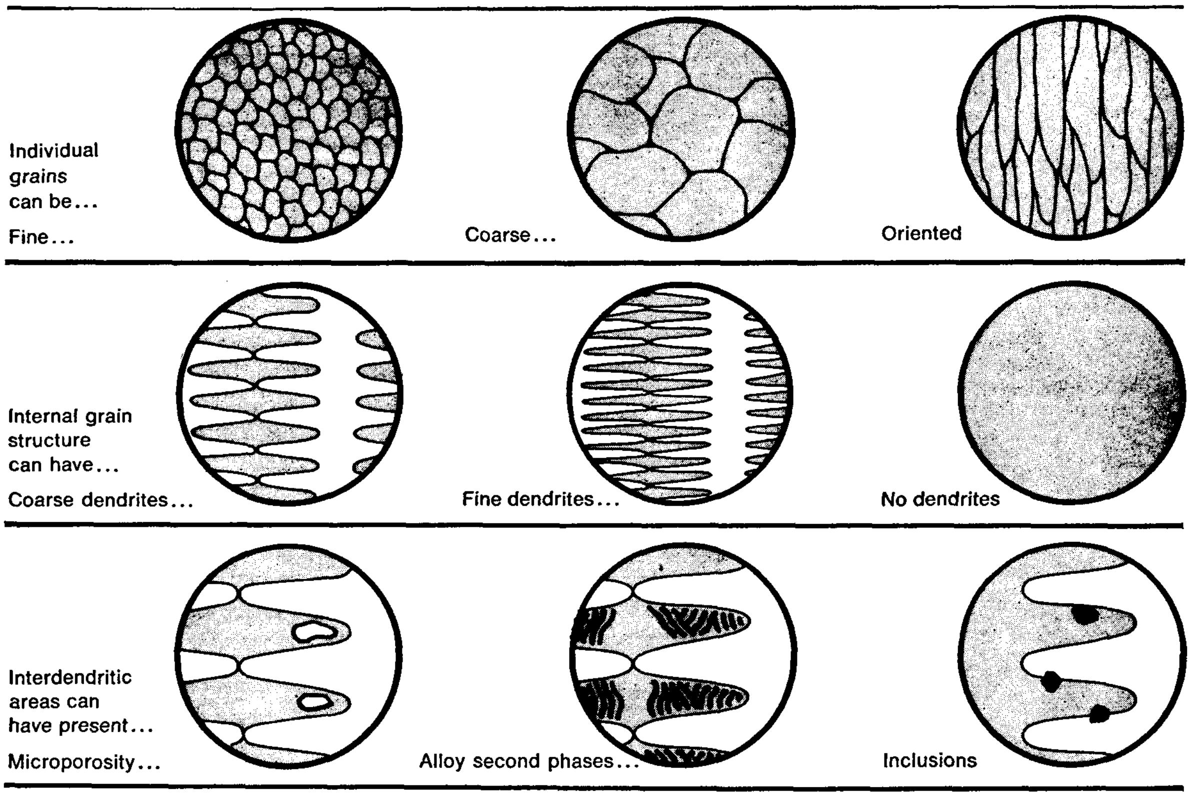


Fig. 3.45. Heterogeneities That Influence Cast-Alloy Properties (from Ref. 20).

Porosity of larger size and less even dispersion can also be caused by moisture, volatile matter, or poor venting in sand molds or by gases or vapors emitted by other types of molds. These are sometimes referred to as blowholes.

Low hot strength is a problem resulting in well-defined cracks or hot tears that develop during solidification and subsequent cooling of castings. The strength of metal at temperatures near its freezing or melting point is very low, and flaws will develop if the design of the casting is such that stress concentrations occur because of uneven contractions associated with cooling or interference with the mold during cooling. In addition to the application of design considerations to avoid high stress concentrations, it is also often possible to reduce the magnitude of stress concentrations by controlling the directions and rates of cooling.

Other defects that occur in casting that are more directly the result of poor molding or foundry practice include cold shuts, dirt spots, penetrations, swells, scabs, and washes. Cold shuts are irregular laps on the surface of a casting resulting from low casting temperature or an oxide film on previously solidified metal. Dirt spots are caused by particles of sand, refractories, or slag that are picked up by the molten metal. Penetration results when metal penetrates the course sand grain openings and gives a rough, sandy surface. Swells are bulges on the surface of the casting that result from insufficient compaction of the mold sand. Scabs and washes are the result of erosion of the mold cavity by the flowing of hot metal.

(b) Designing for Casting. Certain problems in casting are the mutual responsibility of the designer, the patternmaker, and the foundryman. Being the first man in the chain of operations or events, the designer can avoid much grief for himself and ultimately obtain a better casting if he considers the problems of subsequent operations as he develops the design for the valve body.

The initial consideration of the designer is usually selection of the casting material. In addition to the usual considerations involved in material selection, the casting designer must also consider the addition of chemical deoxidizers and other alloys to improve the quality of the casting. He should carefully evaluate the use of various forms of heat treatment of castings to reduce some of the problems inherent in cast materials and to enhance other properties as desired. For example, full annealing often can be used to relieve casting stresses and promote homogenization by diffusion of micro-segregation. Normalizing or even double normalizing can sometimes be used to reduce grain size and significantly increase strength. For some casting designs that will permit rapid cooling without cracking, quenching and tempering can be used for even better mechanical properties. The details of material selection and heat treatment are outside the scope of this document but are readily available in most metallurgical texts and handbooks.

The next consideration of the designer is of moldability, and it simply involves keeping a design configuration which can be molded. This includes the provision of a draft of about 1/8 in. per foot or 0.5° per side to permit withdrawal of the pattern if sand or plaster molds are to be used and more if permanent molds are anticipated. These draft allowances can be removed by subsequent machining in some cases, but the configuration must permit their addition by the patternmaker. The designer should also keep the number of molding cores required to a minimum.

Avoidance of hot spots is perhaps the most important consideration of the designer since the trapping of molten metal and slower cooling in these areas can lead to most of the previously discussed problems. Fundamentally, hot spots are usually the result of heavy sections which occur at intersections of structural members. They are classically categorized as being the L, T, V, X, and Y types since their configuration typically approximates these letters, as is illustrated in Fig. 3.46. The general methods for correcting these hot spots are also illustrated in Fig. 3.46.

Where the use of rapid changes in mass is unavoidable, they must be located so that they can be fed directly from equally heavy risers, thereby not requiring feed flow through lighter sections, or the mold must be designed with heaters or chills to assure that shrinkage in the heavier sections has been compensated before the lighter sections solidify sufficiently to restrict feed flow. Cross sections of a typical cast gate valve and a cast globe valve are illustrated in Fig. 3.47. The flanges on these valves are significantly heavier than the other portions

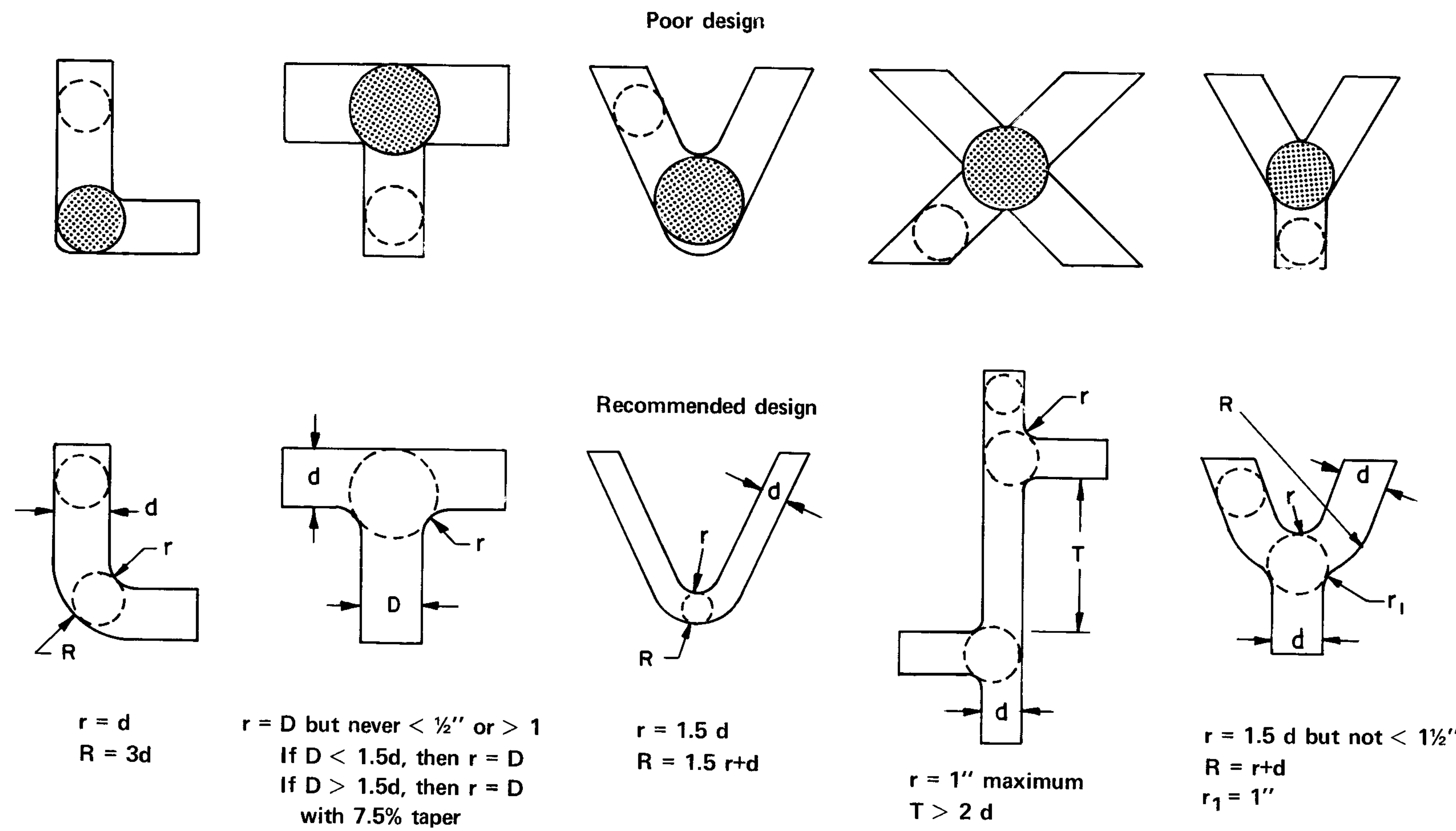
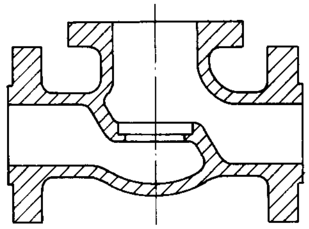
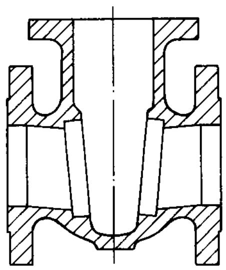


Fig. 3.46. Typical Configurations of Hot Spots and Methods for Their Prevention (from Ref. 21).



A. GLOBE VALVE



B. GATE VALVE

Fig. 3.47. Cross Sections of Typical Cast Globe and Gate Valve Bodies.

of the body, but they were probably cast with separate risers for each flange. The gate valve also has a slightly heavier section in the area around the seat inserts, but this could also have been provided with a separate riser at the base if it were needed. Or a chill block could have been used to assure solidification of this heavier bridge wall before the passages feeding molten metal could solidify and block the infusion of make-up metal.

Stress concentrations are generally more of a problem in cast valve bodies than in those which are forged or machined. Metal at the points of stress concentration in castings will be considerably weaker, doubling the potential for failure at such points. Any sharp angle or drastic change in section will result in planes of weakness caused by the intersection of columnar dendrites from the opposing surfaces, as illustrated in Fig. 3.48. The solution to these problems is the use of generous radii for sections and fillets plus gradual tapers between sections. This will improve the dendritic pattern and therefore the structural

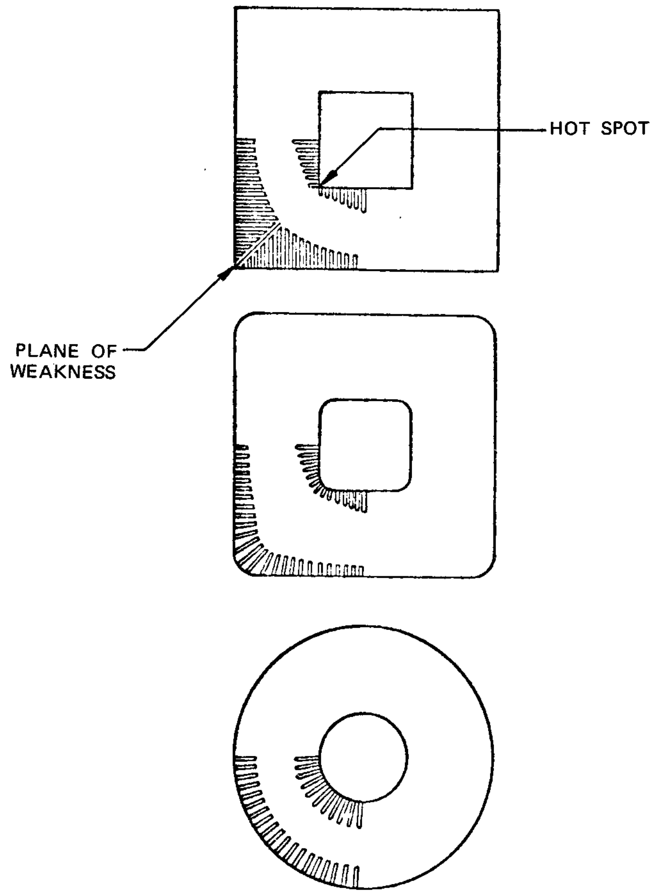


Fig. 3.48. Crystallization Pattern Governed by Mold Shape (from Ref. 21).

design. Where sharper angles are required, it may be desirable to form them by machining the casting. It is sometimes possible to reduce the stress concentrations by altering the dendritic structure via heat treatment if the stress levels are not high enough to cause cracking when the casting cools initially.

Another form of stress concentration occurs when the design is such that contraction of the casting upon cooling is restricted by the mold. Projections, such as lugs and bosses, may produce sufficient restriction to cause hot tears in the casting. This difficulty can often be reduced by the addition of ribs to reinforce thin sections and prevent undue warpage. Sand castings and collapsible cores can also provide some flexibility to further reduce these problems.

Dimensional considerations are also required of the casting designer. For example, cast holes cause difficulty if the mass of material around the hole is too great because the sand in the core used to form the hole becomes fused and cannot be removed. Some generally acceptable ratios of dimensions for casting holes are illustrated in Fig. 3.49. Variations in mold accuracy, shrinkage, and warpage will limit dimensional tolerances that can be met for surfaces not to be machined. Some generally accepted

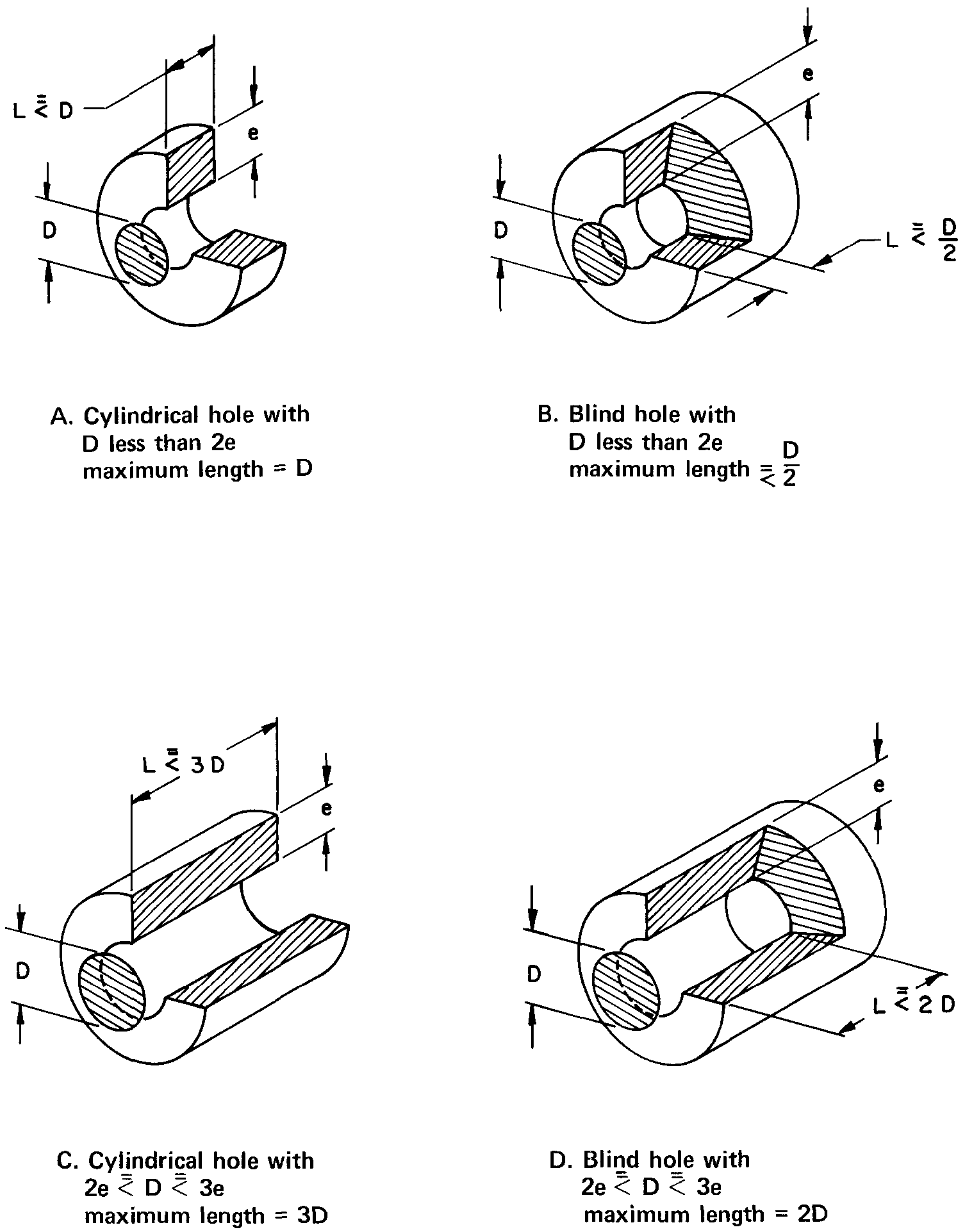


Fig. 3.49. Dimensional Relationships for Cylindrical Through-Holes and Blind Holes In Castings (from Ref. 22).

tolerances for steel castings made in sand molds are given in Table 3.11. Other types of molds can often be used to obtain closer tolerances, and one or two dimensions on a casting can generally be held somewhat closer without undue difficulty. Other dimensional limitations, such as minimum section thickness, usually will not affect the design of valve bodies.

Table 3.11. Suggested Minimum Tolerances for Steel Sand Castings Not To Be Machined[a] (from Ref. 22)

Length[b] (in.)	Tolerance[c] (in.)		Thickness (in.)	Tolerance (in.)		Length[d] (in.)	Tolerance[c] (in.)	
Up to 4	+0.078	-0.063	1/4 to 1	+1/8	-1/16	Up to 8	+3/8	-3/8
4 to 8	+0.125	-0.063	1 to 1 1/2	+3/16	-1/8	8 to 15	+1/2	-3/8
8 to 16	+0.188	-0.063	1 1/2 to 3	+1/4	-3/16	15 to 30	+3/4	-1/2
16 to 32	+0.250	-0.125				30 to 50	+1 1/4	-3/4
						Over 50	+1 1/2	-1

[a]From a design and economic standpoint, tolerance should be as liberal as application warrants with due regard for assembly, service, and replacement.

[b]Transverse dimensions from established longitudinal center line $\pm$ 1/8 in.

[c]For longitudinal dimensions between unmachined surfaces.

[d]Transverse dimensions from established longitudinal center line $\pm$ 3/8 in.

These are only some of the major considerations that must be given the design of castings, and considerably more detailed information is presented in Refs. 22, 23, and 24. The best general rule for the designer is to think through each basic casting step in relation to his valve design: the pattern design, the mold design, the direction and rate of solidification and shrinkage, and the resultant dendritic structure. However, the best course of action for the inexperienced casting designer is close cooperation with the patternmaker and the foundryman.

(c) Effects of Pattern Making and Foundry Practice. The excellent design of a valve body for casting does not in itself assure a satisfactory final product; the patternmaker and foundryman must also make significant contributions. Their contributions basically consist of introducing the melt under proper conditions and adequately controlling the rates and directions of solidification, plus adherence to good practice in the design and construction of the mold and in the preparation of the melt.

Selection of the proper type of molding method and mold material is their first important decision. Most few-of-a-kind valve bodies will be cast in sand molds or in shell molds made from a silica sand mixed with a thermosetting resin. The shell mold offers closer dimensional tolerance and superior surface finish but requires a metal pattern. For large numbers of castings, permanent molds (for more than 500 castings) or die casting (for more than 10,000 castings) may be used. These also generally offer a valve of superior quality, including smaller grain size from more rapid cooling which enhances strength properties.

The patternmaker must then add allowances for shrinkage and select the positioning of the casting, the pouring direction, the location of risers and vents, and the mold parting lines and directions. He must also select the appropriate directions and rates of solidification and allow for the use of chills (and sometimes heaters or endothermic mold liners) to control solidification. It is often desirable that he recognize the critical operational stress points in the valve body and the stress vectors in order to control solidification to produce premium quality in these areas. It is his control of the dendritic structure that makes his contribution so important.

The foundryman must also work with the patternmaker to transfer the pattern planning into construction of the mold. In all molds, and particularly in sand molds, the composition of the mold affects the quality of the casting. Sand selection, moisture content, and compaction are most important, but there are many additional facets that are a product of experience. The preparation of the melt, its deaeration, its degree of superheat, and its handling from the cupola all affect the quality of the casting. Care in pouring is also important to assure that impurities are not dislodged from the mold.

Thus, the product of the foundry is dependent to a major extent upon the capability and experience of its people. Modern methods of mold making, pyrometry, melt handling, and good practice also contribute significantly. These variables result in a wide variety of casting quality obtainable from different foundries.

(d) Assurance of Casting Quality. In view of the many variables which affect the quality of a casting, careful attention must be given to the quality assurance requirements for the valve. Most specifications for nuclear valves require the casting of integral or separate test bars from each metal heat with subsequent testing of the bar to ascertain whether the melt variables were properly controlled. Unfortunately, the properties of this test bar may differ significantly from those of the melt which solidified in the other portions of the same mold. Most specifications also invoke nondestructive test requirements that will usually detect the presence of physical defects such as cracks or porosity.

The degree to which additional control is necessary and can be justified economically depends upon the functional criticality of the valve body in its intended application, the number of valve bodies to be cast, the cost of each additional casting, and the tightness and criticality of the delivery schedule. Other factors, such as the previous performance of castings of a similar design, the extent of the changes from the preceding design, the previous performance of the foundry on similar castings, and the cost of the test work, may also enter the picture. Usually it will be the responsibility of the designer to make or to influence the decision relative to the appropriate degree of additional quality control and assurance measures to be invoked in detailed specifications or purchase orders. Since this decision is based upon many factors, some of the more general types of rather strict quality assurance measures are discussed here to assist the designer in his decision.

Design qualification tests may be specified to assure that if the valve body is fabricated to meet the expectations of the designer, it will indeed function satisfactorily under all combinations of variables to which it may be exposed. These tests normally expose the valve to various combinations of fluid pressure, temperature, thermal shock, hydraulic shock, flow, vibration, environmental variables, operator forces, wear cycling, nozzle reactions, etc., associated with the intended use of the valve. Although this type of qualification is not unique to cast valve bodies, the casting will generally perform differently than bodies of the same design prepared by other fabrication methods.

Fabrication qualification tests may be performed to assure that if certain fabrication procedures are observed, the valve body material will have the physical and chemical properties anticipated by the designer. These tests usually involve sectioning and destructive testing of a limited number of castings or of periodic samples. The testing may include etching; microscopic, chemical, and spectographic analyses; as well as routine physical testing. These tests will ascertain whether the various casting problems previously discussed have been adequately overcome by the design, patternmaking, and foundry procedures used.

Nondestructive testing and procedural control may then be instituted to assure to the maximum degree possible or necessary that production valve bodies are sufficiently similar to those samples found to be satisfactory in design and fabrication qualification testing. Typical nondestructive tests of castings include x-ray, gamma, and beta (for massive sections) radiography; ultrasonic inspection; reverberant sonic testing; and visual inspection of the body casting plus destructive chemical and physical testing of removed coupons or test bars. However, these nondestructive tests are limited almost entirely to detection of cracks and voids.

Procedural control may include written procedures for each individual job in the shop with appropriate audit and record systems. This may be further extended to include materials, personnel, and equipment qualification and auditing. Chemical and thermal control of the melts may also be instituted since it is possible to obtain a satisfactory coupon or test bar but poor casting material if, for example, the degree of superheat is insufficient to assure adequate feeding of remote portions of the casting.

If the designer is faced with the selection or evaluation of previously cast valve bodies, he must resort to statistical sampling and testing methods. As can be seen, complete quality assurance programs for castings can be very expensive and time consuming. They must generally be tailored to be commensurate with the cost of failure. In other words, the cost of failure times the probability of failure (as derived through quality assurance) generally should not be significantly more or less than the cost of the quality assurance.

3.5.2 Forging

The process of plastically deforming a cast or wrought bar or billet to produce a desired form by either hammering or squeezing is called forging. Although some materials can be forged at room temperature or slightly higher, valve body materials plus the large amount of plastic deformation required necessitate hot forging.

When compared with castings, forged valve bodies generally offer the advantages of more uniform structure, greater density, higher strength properties, enhanced directional characteristics, closer dimensional tolerances, and greater economy in mass production. On the other hand, their shape must be relatively simple, the quantity must be sufficient to justify the set-up cost, their size is often limited by available equipment, the material required must be forgeable, and the bodies will be somewhat heavier. The directional structure (flow lines) of a forged piece is compared with the structure of similar pieces fabricated by casting and by machining in Fig. 3.50.

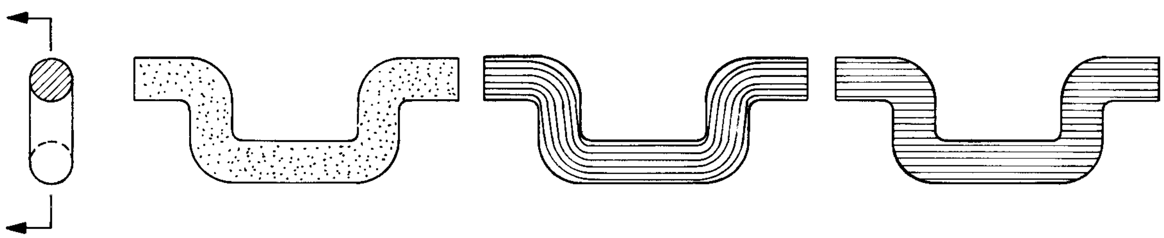


Fig. 3.50. Comparison of Directional Structure of Cast, Forged, and Machined Parts (from Ref. 25).

Most valve bodies are forged by using the closed-die method. The advantages of this method are close dimensional control, close control of physical properties, and control of grain flow in the forged part so that the orientation is in the direction requiring the greatest strength. Other forging methods, such as open-die forging and precision forging, are infrequently used in the fabrication of valve bodies.

(a) Forgeability. Forgeability of a material is defined as the relative amount of work necessary to transform a bar or billet into a shape with defined physical and mechanical properties. Valve bodies may be forged from carbon steel, low-alloy steel, and stainless (high-alloy) steel. The relative forgeability of these steels is given in Table 3.12, where the value of the relative forgeability is based on 0.35% carbon

Table 3.12. Relative Forgeability of Steels[a] (from Ref. 26)

Material	Forgeability
Low carbon steel (0.35% C)	1.0
Lower alloys of Cr-Ni-Mo	1.2
High carbon steel (0.95% C)	1.2
Higher alloys of Cr-Ni-Mo	1.3
400-series stainless steel	1.5
300-series stainless steel	1.65
17-4 PH, 17-7 PH	2.0

[a]Based on 0.35% carbon steel as unity.

steel as unity. Thus, increasing values indicate less forgeability. However, the numbers in Table 3.12 have only a comparative value since there are other variables that have an influence on forgeability.

Variations from nominal chemistry and differences in grain size and surface conditions of the slug can appear in the same alloy poured from different heats, resulting in variations in forgeability. The forgeability is also affected by slight variations of forging temperature, handling speed, and time spent in the furnace. The speed of deformation and the amount of force necessary to move material is extremely important in certain alloys. Some crack when forged at high speed and must therefore be squeezed. The surface finish of dies, die lubricant (type, amount, and placement in the die), and die temperature can affect forgeability, as can the design of dies, skill of operators, and change of furnace atmosphere.

The forgeability of the different steels varies significantly because of their differences in plasticity and hot strength. Steels which are relatively plastic at lower temperatures may generally be forged over a wide temperature range with good die lifetime. The high-alloy steels must be forged at higher temperatures and within narrow temperature limits. The higher temperatures and higher hot strengths result in a shortened die lifetime. For any given shape, die life will be shorter when forging stainless steel than in forging carbon and low-alloy steel.

(b) Designing for Forging. Because of the many materials used in forging different shapes and the variations in the final requirements for valve bodies, this discussion must be general in nature. The designer must consider the shape of the dies to be used, direction of the impact, flow of the forged metal, removal of the dies and cores used, and the relationship of the metal flow lines with the structural aspects of the valve body. Since one objective of the forging operation is to reduce the requirement for subsequent machining to a minimum, the designer must weigh the disadvantages of complicated forging against the cost of subsequent machining.

The major rule in the design of forged bodies is that the design which allows the metal to flow in the easiest and most direct path (up, down, or sideways) is the best from a forging standpoint. If the material cannot move from its original position to its ultimate destination without folding in upon itself or cannot get there because the most direct path is in another direction, it must be forced into its proper path through the use of preliminary dies and operations. Sharp fillets which restrict the flow and allow the material to chill, sharp corners which are difficult to fill, and awkward distribution of mass should be avoided.

In general, fillets can be sharper when the material flows toward them, and they should be relatively large when the material flows away from them, as is illustrated in Fig. 3.51. If a sharper fillet is required, a series of dies (two or more) may have to be used to lead the material around the fillet. Fillet radii for different forgeability values are given in Table 3.13. Drift or taper on the surfaces of a forging permits separation from a die, and a drift value of 5° for steel and 7° for tougher alloys is standard in most forge shops.

Valve bodies may be forged as a solid metal mass and the flow passages subsequently bored, or the major flow passages may be formed during hot pressing. The choice is determined by the shape of the forging. For example, forgings cannot have a large inside void with small openings, nor can the bridge wall structure be too complicated or the flow passages too angular. Consequently, the flow passages of gate valves and swing check valves are usually formed by hot pressing, while the flow passages

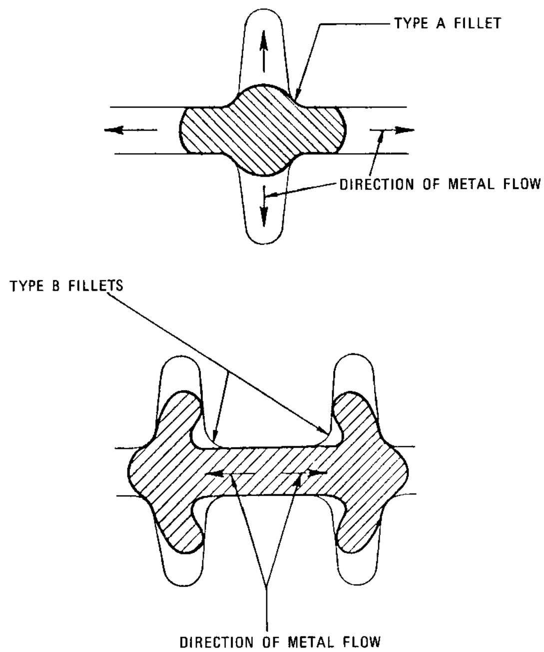


Fig. 3.51. Design of Fillets for Forging (from Ref. 26).

Table 3.13. Minimum Radius of Fillets for Different Forgeability Values (from Ref. 26)

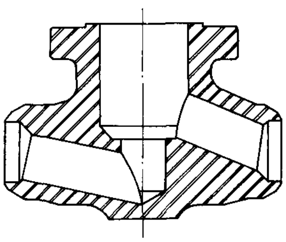
Relative Forgeability Value	Type B Fillet[a] (in.)	Minimum Radius for: Type A Fillet[a] (in.)	Minimum Radius for: Corner Radius[b] (in.)
1.00	0.25		
1.10	0.31	0.12	0.06 to 0.10
1.20	0.38		
1.30	0.50		
1.50	0.62	0.25	0.12 to 0.18
1.75	0.75		
2.00	0.88	0.38	
2.50	1.00	0.50	0.25 to 0.38
3.00	1.50	0.75	
3.50	2.00	1.00	0.5
5.00		2.00	0.75

[a]See Fig. 3.51 for type designation of fillet. These values are based on a fillet leading into a boss or rib 1 in. high. For other heights, increase or decrease fillets by same percentage of change from the 1-in. height.

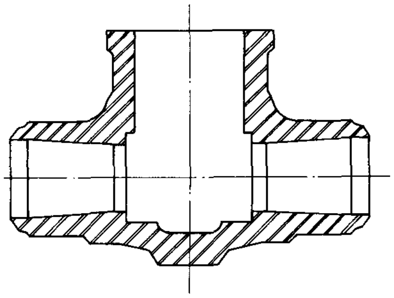
[b]The larger figure in each case represents the preferred corner radius. The corner radius also defines the minimum suggested rib width.

of globe valves with in-line nozzles are usually machined. Forged bodies of gate and globe valves are shown in Fig. 3.52. The flow passages of the gate valve show the draft typical of a forging, but the flow passages of the globe valve are parallel. The gate valve has had subsequent machining of the gate chamber and seat areas.

Close temperature control during preheating and forging is essential to the production of a high-quality forging. The optimum forging temperature for the various grades of carbon or low-alloy steel is influenced somewhat by the carbon content. The recommended maximum forging temperature for carbon steel used as a nuclear valve material is about 2300°F. The finishing temperature should be high enough above the transformation range to prevent rupture of the forging and excessive die wear but low enough to prevent grain growth. A finishing temperature of about 2000°F is satisfactory for all carbon steels with a carbon content of 0.80% or less. A typical temperature cycle with two-stage preheating used for Cr-Ni stainless steel is shown in Fig. 3.53, and temperature ranges for preheating and forging of commonly used stainless steels are given in Table 3.14.



A. GLOBE VALVE



B. GATE VALVE

Fig. 3.52. Forged Globe and Gate Valve Bodies.

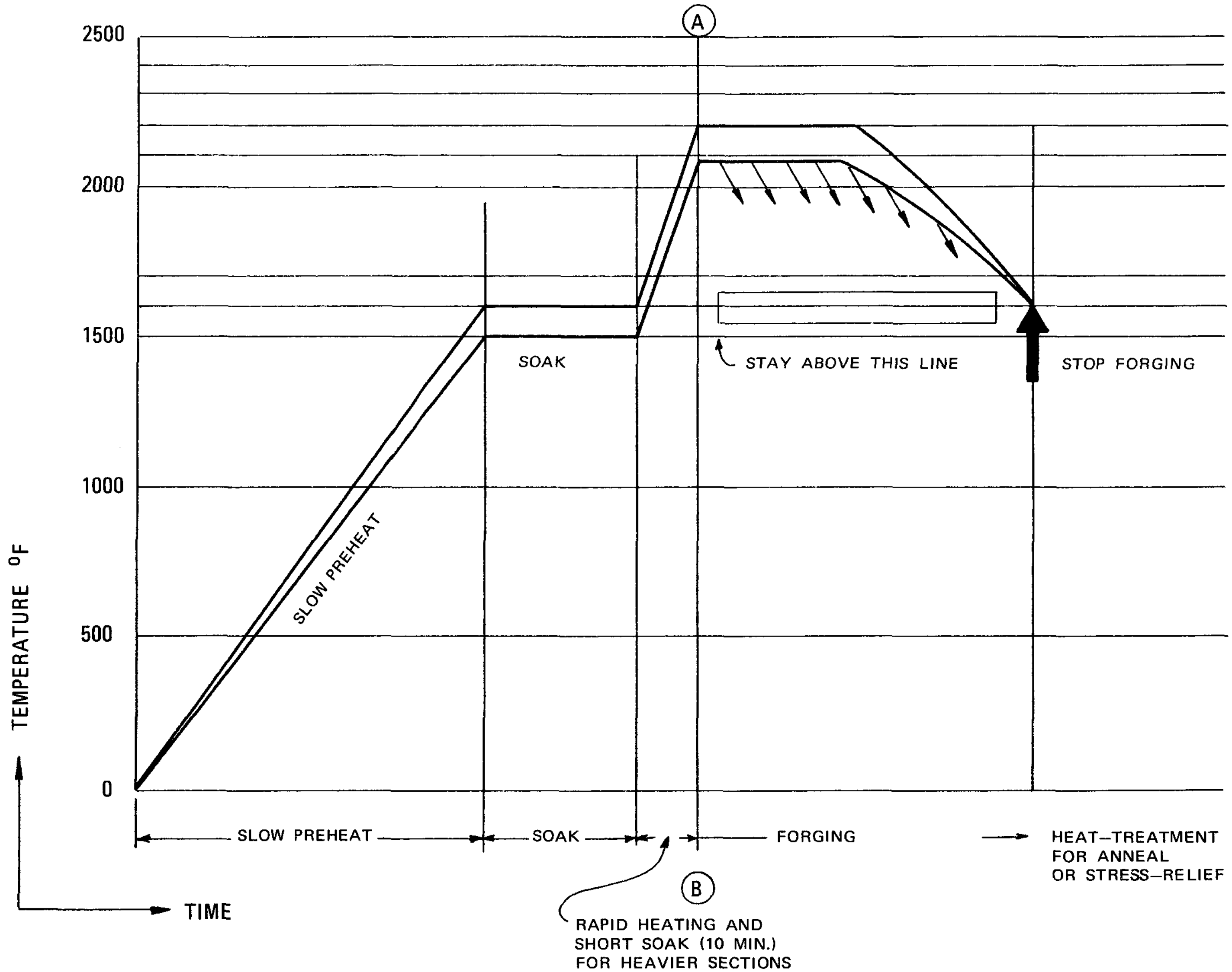


Fig. 3.53. Forging Temperature Ranges With Two-Stage Heating for Chromium-Nickel Grades of Stainless Steel (from Ref. 25).

Table 3.14. Temperature Ranges for Preheating and Forging Stainless Steel (from Ref. 25)

	Preheating Temperatures		Forging Temperatures	
Stainless Steel Type	First Stage Soaking (°F)	Second Stage Quick Heating (°F)	Starting Range (°F)	Finishing Range (°F)
301	1500 to 1600	a	2050 to 2200	1600 to 1700
302	1500 to 1600	a	2050 to 2200	1600 to 1700
303[b]	1500 to 1600	a	2050 to 2250	1700 to 1800
304	1500 to 1600	a	2050 to 2200	1600 to 1700
305	1500 to 1600	a	2100 to 2200	1600 to 1700
308	1500 to 1600	a	2100 to 2200	1600 to 1700
316	1500 to 1600	a	2150 to 2250	1600 to 1700
321	1500 to 1600	a	2100 to 2200	1600 to 1700
347	1500 to 1600	a	2100 to 2200	1650 to 1750
309[c]	1500 to 1600	a	2100 to 2200	1750 to 1850
310[c]	1500 to 1600	a	2100 to 2200	1750 to 1850

[a]Heat to starting range for forging.

[b]Less adapted to severe forging because of its nonmetallic inclusions. It should preferably be worked while at the higher limits of the forging temperature range, particularly in the production of bushings and thin-wall rings.

[c]The forging temperature range is quite limited.

Die wear is brought about by erosion of the die surface caused by abrasion during forging. The amount of abrasion varies with pressure and rate of flow, with the greatest amount of wear occurring in areas of maximum pressure and flow such as around fillets. Accordingly, the general practice is not to assign fillet tolerances, but where they are necessary, die wear should be taken into account. Length-width tolerances allow for variations in the forging caused by different cooling rates of successive forgings, different dwell-time, different die cooling or heat buildup, etc. Length-width tolerances should be applied between points with minimum die wear. Mismatch tolerance is defined as the maximum allowable shift of one die relative to the other, as depicted in Fig. 3.54.

Machining allowance is dependent upon the size of the part, its forgeability, and its setup on the machine. Some commonly used allowances are as follows. Small steel forgings with a relative forgeability up to 1.75 have a minimum allowance of 0.06 in., medium-size steel forgings

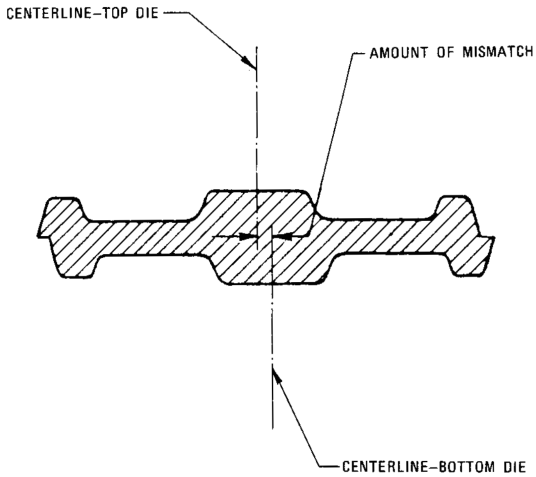


Fig. 3.54. Mismatch or Forging Die Shift (from Ref. 26).

with a relative forgeability of 2.5 have a minimum allowance of 0.12 in., and large steel parts with the same relative forgeability have a minimum allowance of from 0.18 to 0.25 in. Steels with a relative forgeability greater than 2.5 should have a minimum machining allowance of 0.25 in. The relative locations of the various allowances and tolerances are illustrated in Fig. 3.55, and some typical recommended commercial tolerances for steel forgings are given in Table 3.15. Most commercial tolerances are based on the area and weight of the forging.

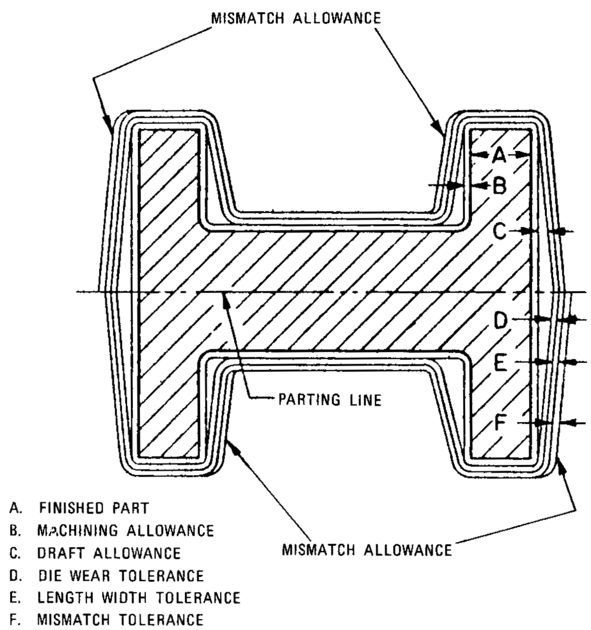


Fig. 3.55. Relative Location of Forging Tolerances (from Ref. 22).

Table 3.15. Recommended Commercial Tolerances for Steel Forgings (from Ref. 22)

Size of Forging: Area ($in.^2$)	Size of Forging: Weight (lb)	Thickness Tolerance: Plus (in.)	Thickness Tolerance: Minus (in.)	Mismatch Tolerance Plus (in.)	Die Wear Tolerance Plus (in.)
5.0	1.0	0.031	0.016	0.016 to 0.031	0.031
7.0	7.0	0.062	0.031	0.016 to 0.031	0.062
10.0	1.5	0.031	0.031	0.016 to 0.031	0.031
12.0	12.0	0.062	0.031	0.016 to 0.031	0.062
20.0	2.0	0.062	0.031	0.016 to 0.031	0.062
20.0	30.0	0.062	0.031	0.020 to 0.040	0.062
38.0	4.5	0.062	0.031	0.016 to 0.031	0.062
38.0	80.0	0.062	0.031	0.025 to 0.050	0.062
50.0	8.0	0.062	0.031	0.020 to 0.040	0.062
50.0	60.0	0.062	0.031	0.020 to 0.040	0.062
50.0	100.0	0.062	0.031	0.025 to 0.050	0.062
95.0	11.0	0.062	0.031	0.020 to 0.040	0.062
132.0	17.0	0.062	0.031	0.025 to 0.050	0.062
166.0	73.0	0.094	0.031	0.030 to 0.060	0.094
175.0	150.0	0.094	0.031	0.030 to 0.060	0.094
201.0	40.0	0.062	0.031	0.025 to 0.050	0.062
240.0	51.5	0.094	0.031	0.030 to 0.060	0.094
250.0	250.0	0.094	0.031	0.030 to 0.060	0.094
265.0	60.0	0.094	0.031	0.030 to 0.060	0.094
275.0	65.0	0.125	0.031	0.047 to 0.094	0.125
300.0	75.0	0.125	0.062	0.047 to 0.094	0.125
300.0	350.0	0.094	0.031	0.030 to 0.060	0.094
375.0	450.0	0.125	0.031	0.047 to 0.094	0.125
415.0	306.0	0.125	0.062	0.047 to 0.094	0.125
525.0	750.0	0.125	0.062	0.047 to 0.094	0.125
900.0	1000.0	0.125	0.062	0.047 to 0.094	0.125

(c) Forging Defects. Most of the defects in forged products originate in the casting of the ingot and therefore are not unlike the casting defects previously discussed in Subsection 3.5.1. However, there are some defects that are a product of the forging operation, and some of the ingot defects may be reduced by forging.

The major shrinkage cavity or "pipe" that occurs at the top of the ingot generally contains a significant amount of segregation and should be removed. Secondary piping may occur within the ingot as center porosity and microsegregation unless solidification of the ingot was properly controlled. Hot working may or may not reduce the microsegregation and weld shut the porosity, depending upon the location, extent, and the amount of working received.

Blowholes are large voids caused by dissolved or occluded gases in the ingot and are also sometimes welded shut during working. Nonmetallic inclusions cannot be removed and may be elongated to form stringers. They are typical in small quantities in most steels and generally cause trouble only when they result in surface irregularities or are sufficiently concentrated to affect strength.

Corner cracks may occur if sharp angles lead to stress concentrations in the ingot, and they are usually difficult to work sufficiently to reweld because of their locations. Internal cracks, called shatter cracks, flakes, or bursts, are often found in larger sections and are believed to be the result of internal stresses and hydrogen evolution. Their formation can be reduced by annealing or slow cooling of the forging.

Overheating prior to hot working or heat treating causes excessive grain growth, oxidation of grain boundaries, and serious surface oxidation or scaling. Whenever steel is heated under oxidizing conditions, decarburization can occur. If this is excessive, it can result in a soft, weak skin. It can be prevented by inert blanketing or reduced time of oxidizing exposure.

Cold shuts occur when surfaces folded together do not weld; shifting of dies causes mismatch; and insufficient metal in the die causes underfill. Surface defects include scabs from metal splashes on the wall of the ingot mold, worked-in scale from improperly cleaned ingots, scores from die imperfections, tears from insufficient surface temperature, and blisters from dissolved gases working to the surface.

(d) Assurance of Forging Quality. All tests and inspections performed on forged valve parts should conform with the Draft ASME Code for Pumps and Valves for Nuclear Power and with additional requirements if the application of the valve calls for more stringent codes. The ASME Code requires ultrasonic examination in the as-furnished or finished condition and magnetic particle or liquid penetrant testing in the finished condition. This code does not require radiographic examination for forgings as it does for castings. Additional quality assurance measures can be specified or influenced by the valve designer when appropriate. The requirements for design and fabrication qualification tests are the same as those discussed for cast valve bodies in Subsection 3.5.1(d). Additional nondestructive testing can also be specified, as was discussed for castings, but some destructive sectioning for micrographic, chemical, and physical testing of samples from each portion of the forging is prerequisite to complete assurance of uniform or minimum structural qualities. Procedural controls must also be extended to the casting of the ingot if they are to be totally inclusive.

3.6 Installation, Maintenance, and In-service Inspection

Only the more general aspects of installation and maintenance are covered here, and detailed instructions for these operations should be obtained from the valve manufacturer. The design features of a valve along with proper maintenance techniques are most effectively and usefully conveyed by a valve "manual". The manual should include all pertinent valve design information, drawings, part identification, and spare part recommendations. The need for and usefulness of this information to the valve user cannot be overemphasized, and the user should insure that he obtains this information for every valve design in his plant. Valve procurement specifications should include requirements for supplying a valve manual and any special tools required for maintenance.

3.6.1 Installation

The installation phase marks the departure of the valve or valves from the design and procurement people into the jurisdiction of the plant operating and maintenance personnel. It is very important that the operating and maintenance personnel understand the design of the valve so that its utilization (operation) will be optimum and its maintenance efficient and effective. Optimum utilization means that a valve should not be expected to perform a function, such as throttling, for which is was not designed; and conversely, it should be used to the full extent of its capabilities.

Installation should be preceded by a careful check of the component to assure that it is in accordance with the specifications and has not suffered shipping damage or become dirty. While thorough receipt inspection procedures are highly desirable, it is recognized that the justification for them is tempered by the degree of previous inspection, such as during manufacture, and the costs of establishing receipt inspection procedures. Receipt inspection should encompass a

1. check to insure appropriate certification of materials and manufacturing inspections,
2. check of external dimensions for compatibility with installation plans,
3. visual check of exterior for damage,
4. cleanliness check,
5. check of valve operation (manual or otherwise),
6. check to see that all shipping supports and/or desiccants are removed, and
7. a check to assure that end connections for mating to piping system are correct.

It follows that the receipt inspection procedures must be carried out in a clean area to prevent the introduction of foreign matter into the valve. Following receipt inspection, the component should be dried out if it has been wetted during inspection, sealed, and stored until installation. Evidence of satisfactory receipt inspection should be affixed to the valve. If extended storage of the valve is anticipated before installation and system testing, special procedures may be necessary to guard against drying out of the stem packing. The manufacturer should be consulted for recommendations in this case.

Valve installation should also be accomplished under conditions that give maximum assurance that no foreign matter, such as stray nuts and bolts, pieces of welding rod, etc., are introduced into the valve. These precautions are particularly significant where the valve is in the reactor coolant flow stream. In essence, a valve serving as a component of a system important to the operation of the plant merits close attention to installation procedures to prevent introduction of a potential cause of failure into the system.

Should the installation be a replacement in an operating plant, it is essential that all applicable radiation control procedures be followed and that plant lineup be such as to insure safety of installing personnel. It is suggested that as a minimum, all valves isolating the work area from the rest of the plant be "locked" shut and tagged to preclude inadvertent operation. Where isolation valves are remotely controlled, their operating circuits should be deactivated and the controls tagged with instructions not to operate.

The installation should be in accordance with manufacturer's instructions, insuring that the physical orientation of the valve is suitable for satisfactory operation and that the flow orientation is proper. The space envelope (unless compromised by overall space limitations) should be such that the valve and/or operator can be removed and/or disassembled and routine maintenance, such as packing replacement, etc., can be performed effectively. Fit-up to the adjacent piping and subsequent welding should be in accordance with appropriate specifications. Adequate support of valve and operator must be assured. During welding, it is preferable to have the disc in a mid-position when possible in the event there is any seat or backseat distortion from excessive heat. Special welding techniques may be required for some valves (those with limited physical separation between the weld and valve seat area) to limit welding-induced distortion to acceptable levels.

Following installation, the connections should be tested with appropriate nondestructive tests and the valve should be cycled to insure that it can operate freely. A thorough final check should be made to insure that all instructions have been complied with. The results of the final inspection of the installation along with the results of the nondestructive tests should be included in the inspection report.

3.6.2 Maintenance

Effective and efficient maintenance is best assured by continued monitoring of valve performance and early correction of any malfunction. Thus, maintenance can properly be divided into preventive and corrective maintenance. Corrective maintenance is the more widely understood and practiced in that it covers the correction of established deficiencies. However, in a nuclear plant where access to equipment may be limited during operation because of radiation, it is desirable to conduct maintenance operations during scheduled outages, such as a refueling period, when the plant is shut down for other purposes. The accent should therefore be on preventive maintenance.

Preventive maintenance requires a continuing assessment of component performance. This assessment should involve an attempt to identify incipient deficiencies so that they may be corrected during a scheduled outage. As an example, assume that a refueling period during which maintenance can be performed without entailing unscheduled down time is scheduled for one month hence. Valves which are suspected "leakers", that is, over the leakage specification requirements but within tolerable limits, should be scheduled for seat and disc lapping during this refueling period. This will permit the correction of a minor deficiency before it becomes one which might require a plant shutdown. If the normal useful life of a gasket or set of packing is to be reached shortly after a scheduled down time, it is also desirable to perform this replacement work during the scheduled down period.

The valves in a system should be checked on a routine basis as part of an effective preventive maintenance program. These checks should be run before scheduled outages to identify areas of potential difficulty, and their timing should be worked into the schedule of plant operations on a "not-to-delay" basis. Examples of checks that might be made are

1. leakage checks of valve seats, backseats, and packing;
2. operability checks for freedom of movement, unusual noises, or vibrations;
3. check of opening and/or closing times within prescribed limits; and

4. periodic nondestructive tests of components, welds, brazing, etc., to insure that no defects, such as a fatigue crack in a weld, have arisen as a result of valve operation.

To augment an effective preventive maintenance program, it is also desirable to maintain a "valve history" file containing records of corrective and preventive maintenance work performed on all valves so that the performance of each valve can be evaluated. These records also help to develop and identify proper intervals for certain preventive maintenance operations.

Since maintenance and plant operations may be carried out simultaneously, it is important that adequate safeguards be established for protection of personnel. As previously stated, all work on radioactive systems must be in compliance with radiation control procedures. Administrative procedures should be developed to specify the degree of isolation from operating systems that is required when maintenance is performed. The operations department should prepare specific instructions for each maintenance operation, and these instructions should reflect the pressure and temperature conditions in the operating systems from which isolation is desired and identify the valves to be shut, tagged, etc. The maintenance department should require a copy of this instruction certified as completed before maintenance is started. When the maintenance work has been completed, it should be carefully inspected and the maintenance department should certify to the operating personnel that the work has been completed, isolation of the sections can be secured, and the section repressurized. Adherence to these or similar procedures will help instill in personnel a sense of responsibility for equipment and personnel safety.

3.6.3 In-Service Inspection

Nuclear valves are important pressure-retaining components of water-cooled nuclear reactor power plants. When used as components in a Class I nuclear piping system, the valves will be subjected to periodic in-service inspection in accordance with the requirements of Section XI, Inservice Inspection of Nuclear Reactor Coolant Systems, of the ASME Boiler and Pressure Vessel Code. The design of the valve and its location within the fluid system must provide access for this in-service inspection. The areas to be accessible over the service lifetime of the valve for inspection by visual and volumetric examination (radiographic, ultrasonic, etc.) methods are

1. pressure-retaining seam welds in valve bodies,
2. valve-to-safe-end welds or valve-to-pipe welds, and
3. pressure-retaining bolting.

Additional areas requiring examination are weld repairs in the base material that are not associated with the weld seams. Records of weld repair locations should be maintained to provide definite knowledge of their presence, location, and extent so that if signs of distress are encountered during plant operation, they can be related to the original fabricated condition and standards.

4. DESIGN OF VALVE PARTS

Design features and requirements for valve bodies and bonnets; seats and discs; stems; closures and seals; handwheels, operators, and position indicators; locking devices; and bellows are presented in this chapter.

4.1 Bodies and Bonnets

The body and bonnet together comprise the largest component of a valve and functionally represent its principal part. The body and bonnet must

1. provide the housing for the flow control device with a penetration for its operating mechanism,
2. provide a containing structure of sufficient strength to retain the process fluid over its full range of design pressures and temperatures without bursting,
3. provide a container with the leak-tightness required by the process system criteria and the criteria for control of process fluid within the valve environs,
4. provide fluid flow passages suitable to direct the fluid through the flow control device and return it to the process system piping, and
5. provide a structural member representative of the displaced section of process system pipe.

4.1.1 Shape

The design of the body and bonnet is most heavily influenced by the functional requirements that dictate the valve type, such as gate, globe, and check. These functional requirements dictate the general arrangement of the seat, disc, and stem attachment; and this arrangement in turn controls the general shape of the body and bonnet. Within the configuration established by the valve application, fluid flow considerations also have a major role in shaping the body and bonnet. The effect of these fluid flow considerations is illustrated by the adoption of the angle globe valve and the Y-body globe valve.

The design of the body and bonnet is further influenced by the properties of the materials used and stress limitations, as discussed in Sections 3.3 and 3.4, respectively. Allowances must also be made for the capabilities of the fabrication processes used, as discussed in Section 3.5. Valve body shapes are also influenced by reinforcement requirements, and the basic stress requirements for valve body reinforcements are outlined in Section 3.4. For example, the joint between the bonnet nozzle and the main body of the valve requires the addition of metal in the form of reinforcement to reduce the stresses in this area. In smaller valves, the generous overall thickness of the valve body may form its own reinforcement. The type of reinforcement geometry selected is normally a function of economics and fabrication techniques, as is discussed in Section 3.5.

In a broad sense, the body contour is considered to mean that metal which is superfluous to the wall thickness and reinforcement requirements for the body. Body contour is often necessary for casting or forging design, and other body contour requirements arise from the flow design of the valve.

4.1.2 Wall Thickness

The design thickness of the wall of the valve body and bonnet is determined primarily by the stress analysis and the requirements of the ASME Code for Pumps and Valves for Nuclear Power, as is pointed out in Section 3.4. Sections 3.3 and 3.5 also contain information about body wall thickness since this thickness must ordinarily be increased to include a corrosion allowance and to provide a margin for fabrication tolerances. The tolerance requirements for casting are generally greater than those for forging. Thicknesses around weld joints may be increased to provide for possible offset alignment during welding joint fit-up.

Emphasis should also be placed on the effects of erosion when determining the thickness of valve body walls. Valve body materials will not withstand erosion and cavitation, as is pointed out in Section 3.2, and care must be taken to minimize cavitation by flow design and by matching the valve to the system. The addition of thickness to the walls of valve bodies cannot by itself compensate for major erosion or cavitation damage. It is pertinent to note that the stress design of the valve remains valid only as long as the wall thickness of the body is not reduced below its design value by corrosion or erosion.

The general design sections referenced in this discussion deal with the strength requirements that legislate the thickness of valve body walls. There are also the economic and practical objectives of minimizing the wall thicknesses of valve bodies to reduce costs and improve handling of valves once the objectives of safety and flow design have been met.

4.1.3 External Dimensions

The external dimensions of valve bodies and bonnets are prescribed largely by the ANSI Standard B16.5, Steel Pipe Flanges and Flanged Fittings, and ANSI Standard B16.10, Face-to-Face and End-to-End Dimensions of Ferrous Valves. The relationship of these standards to the valve body is given in Section 3.1. These standards have been developed and adopted by the valve industry to facilitate the conveyance of valve spatial requirements and mating dimensions to the designers and constructors of process systems. Adherence to these standardized dimensions reduces the confusion between the valve manufacturer and the user.

The external dimensions of valves are influenced the most by valve types. A gate valve with a relatively narrow disc will have a smaller end-to-end dimension than a globe valve of the same size. There are, of course, special applications that may justify modification of standard dimensions. For example, it may be desirable to taper an enlarged section to provide a more gradual change in cross-sectional area, and the addition of safe-ends to facilitate field installation may make it desirable to increase the face-to-face dimensions of the valve body. The height of the bonnet of a globe valve may be increased to allow full nesting of the disc within the bonnet in the fully open position.

Fabrication requirements sometimes affect the outside dimensions of valve bodies and bonnets. This may be the case for a forged valve body in which the flow path must be machined out of a blank forged body. The end-to-end dimensions must not be too large to interfere with proper setup of the boring tool. Gate valves with wedge discs must be dimensioned to permit the proper setup of tools to machine the seat faces. A very important factor that may require a long nozzle dimension is the requirement to minimize welding heat on the seating area of the valve. If the valve is to be welded into the system and a multi-pass weld must be used, the heat associated with welding may damage the valve seating area by causing distortions in the valve body or seat.

4.1.4 Finish

The finish requirements for valve bodies and bonnets are influenced by various considerations based on whether the interior or exterior of the valve is being evaluated. Some of these considerations, such as nondestructive testing, are applicable to both inside and outside finish. While no attempt is made here to discuss nondestructive testing in detail, its significance with respect to surface finish is that the successful performance of this type of testing requires a certain quality of surface finish. Poor surface finish can lead to improper interpretation of radiographs. A surface irregularity can be interpreted as a crack. On the other hand, poor surface finish can hide cracks with liquid penetrant testing. Similarly, any surface roughness can prevent meaningful results from magnetic particle and ultrasonic tests. Thus, poor surface finish prevents reliable interpretation of the results of nondestructive tests.

Another impetus for smooth internal surfaces is to lower surface friction to decrease flow resistance. Since extensive internal machining is impractical to perform and would be prohibitively expensive, surface finish conditions are determined by fabrication. Normally, the surface finish for forged valve bodies is suitable for nondestructive testing and fluid flow requirements, but special care may be required to achieve a suitable finish when the valve body is cast. Both internal and external surface finish requirements must account for corrosion, as discussed in Section 3.3.

A final incentive to achieve a smooth internal surface is to reduce the collection of radioactive corrosion products in the open pores of valves used in process systems circulating reactor coolant. This can be particularly important in reducing the difficulty of performing maintenance on valve internals.

4.2 Seats and Discs

The leak-tightness of valve seats and discs is a particularly important aspect of valve design for nuclear service. Although the proper initial leak-tightness of a valve may be proved by initial testing, extended long-term leak-tightness of the valve cannot be predicted by initial testing. Yet, it is this extended leak-tightness of a valve that determines its ultimate capabilities for satisfying the system requirements. Extended leak-tightness requiring little or no maintenance takes on added importance in nuclear reactor system applications. The design considerations important to achieving this extended leak-tightness of valve seats and discs are discussed in this section. These include the mechanics of seat-disc contact; seat-disc contact theory; valve seat and disc design for globe, gate, and check valves; and valve backseat design.

4.2.1 Seat-Disc Contact Mechanics

Metallurgical contact theory deals with microscopic contact between materials. When magnified, even the smoothest of surfaces appears irregular and rough. Contact between such surfaces therefore consists of contact between only the peaks of the rough irregular surfaces that are called "asperities". Thus, the actual contact area between the surfaces is smaller than the apparent contact area by a large factor. In fact, actual contact is limited to only the spots where the asperities contact.

The importance of this small contact area is apparent for fluid leak-tightness at the disc-seat contact area in a valve. The fact that the disc and seat are in contact and under high contact pressure does not in itself guarantee that any more than a small percentage of the areas are in contact. Since only a small percentage of the mating surface areas are actually in contact, there is room for the fluid to pass between the contact surfaces. If the mating surfaces are quite rough, that is, if the asperities are relatively large, the flow passages between the mating surfaces will be large and a high leakage rate can be expected. If the surfaces are polished, the flow passages will be small and the leakage rate will be small. Thus, the rate of leakage through the junction is a function of the smoothness and finish of the surfaces.

However, the actual microscopic contact area may be quite small even for two polished surfaces. Furthermore, extending the line width of the apparent mating contact area between the pieces does not necessarily stop leakage; it merely reduces leakage by providing higher flow resistance in the same way that lengthening a pipe increases flow resistance but does not shut off flow. If it is required that the mating surfaces form an absolute leak-tight joint, the surfaces must be forced together with the asperities deforming elastically and/or plastically until a large percentage of the mating surfaces are in actual microscopic contact.

A parameter indicative of good contact is the ratio between the actual microscopic contact area and the apparent contact area of the mating surfaces. Good contact is then defined as the degree to which the area ratio approaches one. The following simplified relationship has been used to describe the area ratio between two mating surfaces.[27]

$$\text{Area Ratio} = \frac{\text{Microscopic Contact Area}}{\text{Apparent Contact Area}}$$

where the microscopic contact area is approximately equal to the applied force divided by the material flow pressure. Therefore,

$$\text{Area Ratio} = \frac{\dfrac{\text{Applied Force}}{\text{Material Flow Pressure}}}{\text{Apparent Contact Area}} = \frac{\text{Applied Pressure}}{\text{Material Flow Pressure}} \; .$$

The material flow pressure is approximately three times the yield strength of the softer material of the two in contact.[14]

An example will illustrate the smallness of the area ratio obtainable in actual practice. Consider a seal material with a yield strength of approximately 70,000 psi. The flow pressure would then be very nearly 200,000 psi. If an applied pressure of 1000 psi is used to force the surfaces together, the

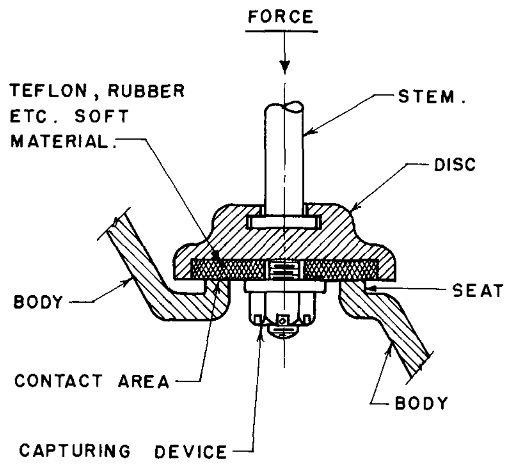
$$\text{Area Ratio} = \frac{1000}{200{,}000} = \frac{1}{200} = 0.005 \ .$$

This shows that only a small percentage of the two surfaces ever come together in contact. The forces pushing the two surfaces together are seldom sufficient to deform or flatten more than a small percentage of the high points in the two mating surfaces. Thus, an absolute leak-tight seal is an idealized goal.

Practical characteristics of fluids and seal materials make it possible to achieve a leak-tight seal within the sense of a measurable leakage rate without achieving an area ratio of 1.0. The area ratio relationship is useful in determining how best to minimize leakage by maximizing the area ratio. First, the applied contact force and the applied pressure can be increased to increase the area ratio. This would increase the size of the valve components to provide the additional strength needed.

Second, the applied pressure can be increased by decreasing the contact area. This is done in high-performance valves by designing the valve seating surfaces to mate with "line" contact. Line contact is so named because contact takes place around the seat on only a circular line. Since this line is designed to be as thin as possible, the integrated surface of the contact area is very small. The applied pressure is therefore high, increasing the area ratio. It should be noted that surface deformations, which may occur because of high applied pressure, tend to increase the width of the line as this deformation proceeds. This increase in line width decreases the area ratio. Where this can occur, the valve is generally designed to minimize such deformation through the provision of maximum hardness in the mating surfaces.

Last, the contact area ratio equation indicates that changing the material flow pressure of the contact surfaces also changes the area ratio. Taken on face value, it would appear that a low material flow pressure is desirable. In fact, this is the case for many valves where soft seats are acceptable. Soft-seated valves provide a high area ratio because of the readiness with which microscopic deformations take place so that the contact areas may blend together. The geometry of the mating surfaces in soft-seated valves is generally like that depicted in Fig. 4.1. It is clear that the applied pressure on the contact area varies only with stem force. Deformation of the mating surfaces does not affect the applied pressure.



$$\text{CONTACT PRESSURE} = \frac{\text{STEM FORCE}}{\text{CONTACT AREA}}$$

NOTE: EFFECT OF PROCESS FLUID PRESSURE NEGLECTED

Fig. 4.1. Geometry of Mating Surfaces in Soft-Seated Valves.

Where soft surfaces will not stand up under severe operating conditions, hard materials and a totally different configuration must be used. A valve designed to use hard seat materials is illustrated in Fig. 4.2. In this valve, a differential angle is provided in the geometry between the disc and the seat to establish a line contact. This limits actual contact to a very narrow line around the valve seat. Maximum hardness of the seating materials is desirable in this type of valve. If the hardness is not sufficient, the geometry of the mating surfaces is such that deformation will allow the apparent contact area to increase through indentations in both the seat and the disc. An increase in the apparent contact area would mean that for a constant stem force, the applied pressure of contact would decrease. Therefore, increasing the hardness of

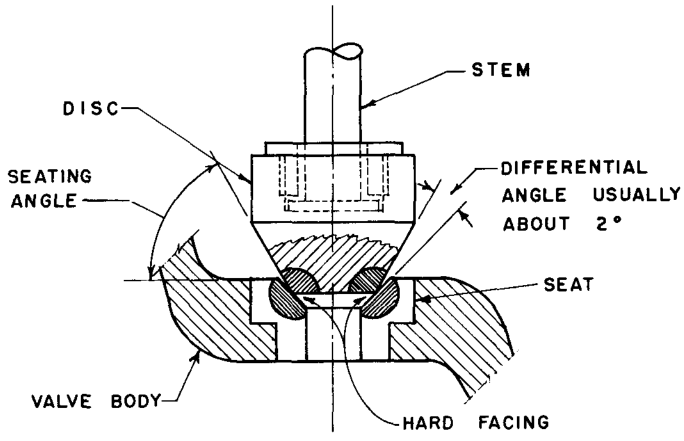


Fig. 4.2. Included-Angle Globe Valve Designed for Hard Seat Materials.

the mating surfaces in such a seating arrangement improves the ability of the valve to maintain a narrow line seal with a small apparent contact area over its lifetime. Plastic deformation of a soft material used in this type of seal design can result in an increase in the apparent contact area of a factor of two or three with a corresponding decrease in the applied pressure. Thus, hardness of seating surfaces greatly increases the numerator of the area ratio equation.

On the other hand, hardness and surface strength property changes in the mating surfaces can only affect the material flow pressure of the material by fractions. Thus, an increase in the hardness has only a weak effect on the increase in the denominator of the area ratio equation. The net result is that the higher the hardness, the greater is the area ratio and the more leak-tight the seal is apt to be. This applies to differential-angle seats only. It will be seen that this is a common configuration and an ideal to be attained. Where maximum hardness is not required for contact considerations, it usually is for other reasons such as maximum wear resistance, which is discussed later.

Some important conclusions can be drawn from the preceding analysis. First, this analysis indicates that well-aligned, line-contact globe valves will provide good seat leakage characteristics. It indicates that the leak-tightness of gate valves may not be as good because true line contact is not attainable with the geometry of gate valves. Further, since wedge gate valves apply high pressures through the additional mechanical advantage of the wedge disc, they may have higher area ratios and less leakage than non-wedge types. This is not true for very large parallel disc gate valves. Not only is line contact not possible for them, but the additional applied pressure from stem force and wedging action cannot be brought to bear. In very large parallel disc gate valves, any force provided by the stem is usually small and incidental when compared with the very large force from the differential pressure across the large disc. When this condition occurs, the parallel disc gate valves are equally as leak-tight as the wedge gate valves. In addition, parallel disc valves have a seat wiping feature that may provide a measure of seat cleanliness not obtained with wedge gate or globe valves. However, from the standpoint of control theory alone, it appears that such valves cannot as easily bring as many factors to bear in increasing the area ratio as can wedge gate valves or globe valves.

4.2.2 Seat-Disc Contact Theory

Using the elementary discussion on the mechanics of surfaces in contact presented in the preceding subsection and supporting it with trial and error, ways of improving leak-tightness of valves can be found. A comparative analysis is presented here in which the problem is reduced to a mathematical model and solved to determine the quantitative importance of the various parameters which contribute to seat-disc leak tightness. The new emphasis on analytical theory in valve seat-disc design has resulted from the more stringent requirements for leak-tightness in nuclear reactor plant application.

Surface contact analyses in the literature are presented from several standpoints and with different mathematical models. Yield and plastic deformation are assumed to occur in some cases, and purely elastic deformation is assumed to occur in other cases. The surface is assumed to consist of small wedges in some cases and of small caps of spheres in others. Since surface wave lengths of several magnitudes can be present simultaneously, a "wave" concept is built into some theories. It is likely that none of these mathematical models give a true picture of the phenomenon of surfaces pressed together to form a seal. As our knowledge of the mechanics of surfaces in contact increases, the mathematical model used to describe this phenomenon will become more accurate.

Such analyses are not without practical benefit. Since the problem of valve leak-tightness is reduced to a mathematical model, several important parameters affecting seat material and geometry can be determined. For instance, the relative merits or rating for each type of seal (line contact or area contact) can be indicated in the analysis, and other variables in the design (material hardness, contact pressure, and surface roughness) may be evaluated. All of these factors can be determined within the inherent accuracy of the analysis.

For the applications considered in this document, it will usually be assumed that the seating contact will consist of surfaces with exceptional hardness that are polished to a high degree of smoothness. The seal must also remain tight over a large number of opening-closing cycles. Therefore, the analysis using a mathematical model of sphere caps and assuming only elastic deformation can generally be considered applicable. Such a quantitative analysis was developed by the Berkeley Nuclear Laboratories in England probably because of the high leak-tightness requirements for valves in HTGR service,[28] and the theory of that analysis is presented here. The treatment is generally applicable to any fluid (liquid, vapor, or gas) with the inclusion of a suitable viscosity constant.

The analysis can be briefly described as follows. First, the geometrical model is established, as illustrated in Fig. 4.3. The surface is assumed to consist of small caps of spheres placed adjacent to each other in rows and columns. These small spherical caps are identical in size. The height, h, of each spherical cap is taken as the distance from a line through the point at which the cap connects with the adjacent spherical caps to the upper or outermost point of the spherical cap. It may be seen in Fig. 4.3 that if R is the radius of each spherical cap, a measure of the surface smoothness is h/R. At this point, it is assumed in the analysis that a perfect plane is laid in contact with the tops of the

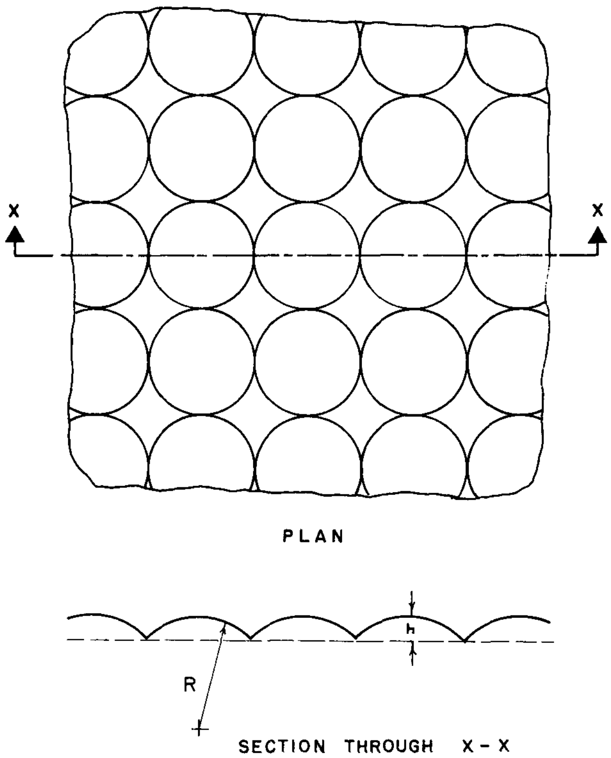


Fig. 4.3. Model of Surface Roughness.

spherical caps. These surfaces are then loaded together so that the tops of the spherical caps are flattened. Consider a point on each body such that these two points approach each other by a distance α as the two bodies are loaded together. In this case, α is the deformation of the spherical caps or the distance by which the two surfaces approach each other as the force pressing these surfaces together is increased. The distance α can be calculated by elastic theory as a function of the radius R, the apparent contact pressure P_A, and the material characteristics ν and E.

For small apparent contact pressures, a flow path will exist between the rough surface and the perfect plane. The volume of fluid between these two surfaces will depend upon the approach distance α until the spherical caps are flattened. It is assumed in this analysis that α cannot exceed h. The simplification is also made that this fluid volume can be expressed as a uniform gap of height δ. That is, a fictional gap between smooth surfaces of height δ will contain the same volume as the irregular voids remaining between the rough surface and the perfect plane. For this purpose, it is desirable to express the gap in the non-dimensional form δ/h.

It can be shown by fluid flow theory that the quantity of fluid passing the gap δ between two smooth surfaces is proportional to δ^3 when a given fluid pressure differential from gap inlet to gap outlet is assumed. The quantity of fluid leaking through the gap is also a function of the length of the leakage path or the line width of the apparent contact area. That is,

$$Q \text{ is proportional to } \delta^3/w$$

where

Q = volumetric leakage rate per unit length of seal and

w = seal line width or the straight-line distance leakage must pass going through the seal.

Simultaneous solution of the related equations provides a plot of the cube of the equivalent gap $(\delta/h)^3$ as a function of the apparent contact pressure expressed in non-dimensional form. A family of curves for various ratios of surface roughness given in the h/R relationship is illustrated in Fig. 4.4. Since the leakage rate is proportional to δ^3/w,

$$Qw \text{ varies as } \frac{\delta^3}{h^3} \text{ or } \left(\frac{\delta}{h}\right)^3 .$$

Therefore, the ordinate is also a scale for the product of leakage rate multiplied by seal line width.

This contact theory is useful in evaluating the various factors affecting valve seat leakage and determining how they may be changed to improve an existing valve design. However, the results would not be very accurate when used to determine actual leakage rates from certain given parameters. The contact theory is not sufficiently accurate to permit experimental duplication of the curves illustrated in Fig. 4.4, but experimental work is expected to produce a set of curves of similar geometry to confirm the relationships of the various parameters.

The relationships illustrated by the curves in Fig. 4.4 can be explained as follows. The ordinate, $(\delta/h)^3$, is a dimensional quantity which is directly proportional to the volumetric leak rate multiplied by the seal line width. The seal line width is the straight-line distance

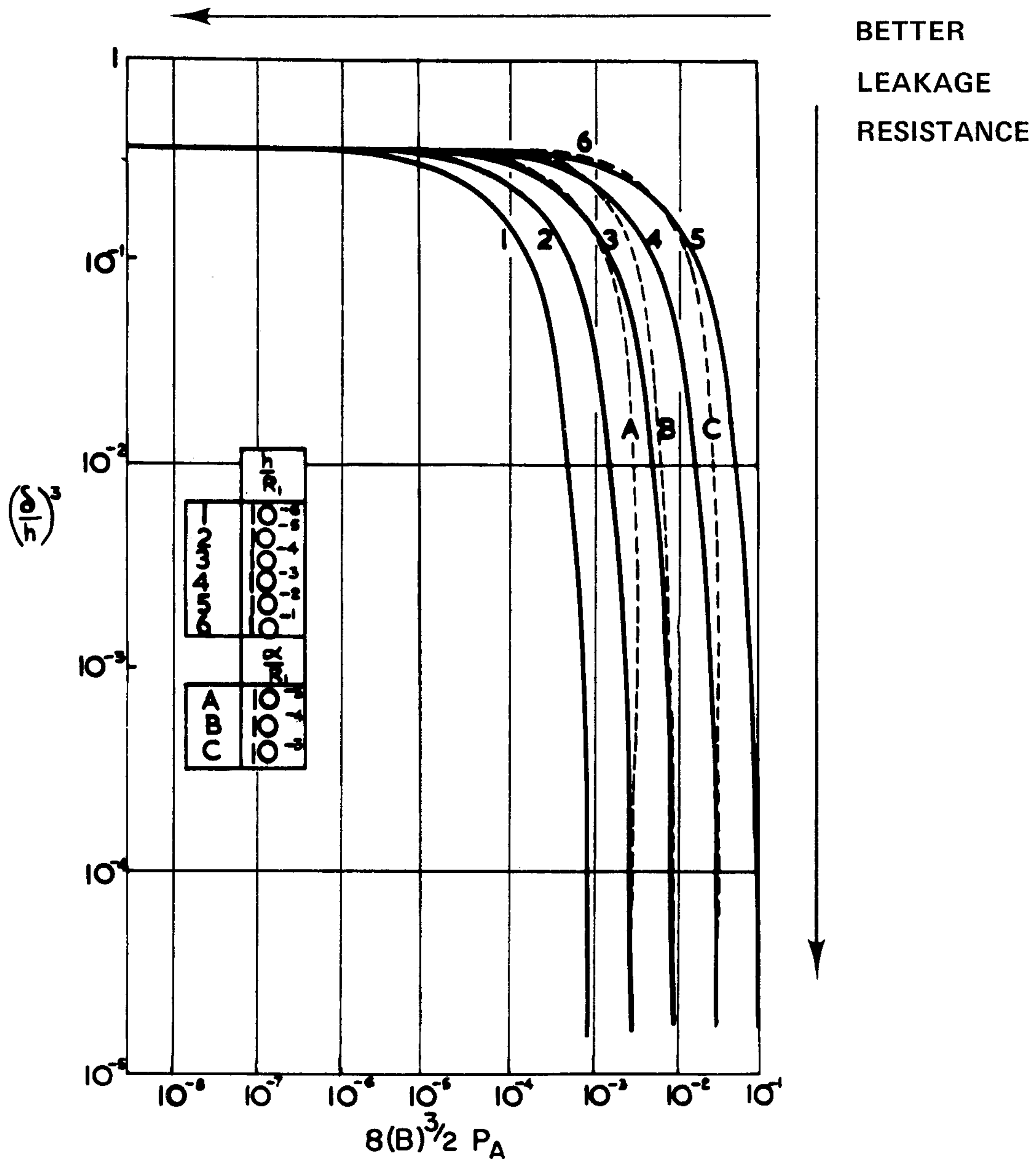


Fig. 4.4. Variation of the Cube of the Equivalent Gap With Apparent Contact Pressure (from Ref. 28).

the leakage must travel in going through the seal. There are other factors, such as fluid viscosity, pressure drop, etc., that must be assumed constant for the sake of simplicity. So, $(\delta/h)^3$ should be reduced, if possible, in good seal design to reduce both leakage and seal line-width requirements. Obviously, if Qw is small, the leakage rate must be small. However, a high $(\delta/h)^3$ can be tolerated if the seal length is allowed to be large since leakage is also inversely proportional to seal length. One must keep in mind that seal length is dependent upon other factors.

The abscissa is $8(B)^{3/2}P_A$, which is a dimensionless quantity representing the contact pressure where B is a parameter defined as a mathematical combination of the contact material properties of Poisson's ratios and Young's moduli.

$$B = \left[\frac{9}{16}\left(\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2}\right)^2\right]^{1/3}$$

where

ν_1, ν_2 = Poisson's ratios of the two bodies in contact and

E_1, E_2 = Young's moduli of the two bodies in contact.

The parameter B does not differ greatly for metals. The abscissa is therefore almost a direct function of the apparent contact pressure P_A.

The need to use sufficient apparent contact pressure on the surfaces to deform the spherical caps elastically and reduce the leakage gap is illustrated in the sectional view of Fig. 4.3. The resulting curves of $(\delta/h)^3$ as a function of contact pressure illustrated in Fig. 4.4 indicate that $(\delta/h)^3$ remains constant up to a critical contact pressure, determined by the degree of smoothness of the surface, at which point $(\delta/h)^3$ decreases rapidly. At this point, the surfaces have been brought into a high degree of contact and leakage is cut off. Thus, the first objective in obtaining leak-tightness is to apply enough apparent contact pressure to operate the seal in the region to the right of the knee in the curves. In this region, small increases in apparent contact pressure produce a rapid decrease in leakage or, specifically, in Qw.

The index h/R is a dimensionless term representing smoothness. The index α/R represents an elastic deformation limit. A surface finish h/R cannot have a deformation greater than $\alpha/R = h/R$ and still be considered within the scope of this analysis. Since h/R is not directly convertible to rms surface finish measurements, assumptions must be made about the measurable surface finish to be specified for the seat surfaces.

The group of curves for h/R illustrated in Fig. 4.4 illustrates that improving the surface finish moves the knee in the curve to the left.

Thus, the surface finish can change the required apparent critical pressure by factors of 10 to 100 for a given leakage rate. This can be a high premium to pay for not polishing the contact surfaces. Rapid multiple-factor increases in leakage are also a high premium to pay for surfaces roughened by erosion and wear or for a design where there is a high degree of sensitivity to slight roughening caused by normal operation. Further, it is interesting to note that surface finish has a large effect on leak-tightness, at least in the elastic theory. Undoubtedly, where small plastic local deformations occur, the initial surface finish has less of an effect. What actually happens in seat overstress is not clear and has not been well investigated. In any case, the harder and stronger the hard-facing seating material, the more applicable is the elastic analysis, which includes this emphasis on surface polish.

One other factor about Fig. 4.4 should be emphasized. The ordinate $(\delta/h)^3$ is directly proportional to the product of the leakage rate and seal line width. Should structural limitations prevent the use of an apparent contact pressure to the right of the knee in the h/R curves, the leakage rate may be reduced by making the line width of the seal very wide. In this case, the value of Qw remains constant but Q becomes smaller as w increases.

From the preceding discussion, it is apparent that there is some seal line width that will be more apt to leak than others. A way of designing to the right of the knee in the h/R curves is to make the line width of the seal very narrow, increasing the value of P_A. But, as previously pointed out, making the line width of the seal very wide also reduces the leakage rate. Thus, there is a poor seal line width for any given applied force on the seat, as is illustrated in Fig. 4.5. The maximum leakage will occur when the various parameters place the seal design in the knee of the h/R curves of Fig. 4.4.

The conclusion generally reached from the results of the theoretical analysis is that the valve seat should be either very narrow or very wide. Clearly, this must be tempered with other factors such as compressive stress resistance and thermal stress design. This can be immediately recognized in valve practice as the difference between the approaches taken in globe valve design where line contact is used and in swing check

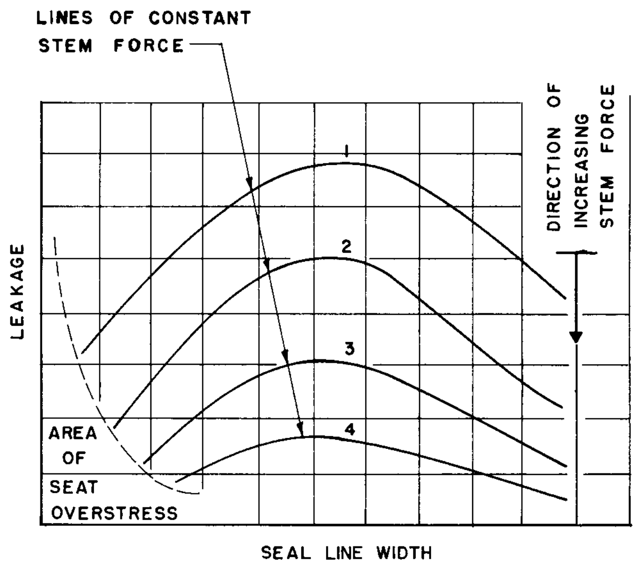


Fig. 4.5. Conceptual Graph of Leakage as a Function of the Seal Line Width for Constant Stem Force.

valve design where a large flat-faced seat is used. Each can be designed to have acceptable leak-tightness from the standpoint of the theory, given no other complicating factors. But within the scope of practical limitations, the basic fundamental difference between "contact pressure" seals, high pressure and low area, and "area" seals, low pressure and high area, should be remembered.

4.2.3 Seat and Disc Design for Globe Valves

Globe valves designed for high-temperature and high-pressure nuclear service normally have either an integral hard-faced seat or a welded-in seat ring. This practice gives almost complete assurance of no leakage around the back of the seat ring. Since integral seats obviously cannot be replaced and welded-in seat rings are difficult to replace at best, the design should provide for in-place lapping of the seat a number of times over the service life of the valve.

If the integral hard-faced seat is used, the requirements for applying a thick, uniform, and hard material to the valve seat should be recognized. The better hard-facing materials are applied by a shielded welding process, and the hard-facing process must be performed to a qualified welding procedure. The base material on which the hard-facing material is to be laid must be sound. Careful control of the temperatures during each step of the process is necessary to assure that the hard-facing material is uncontaminated and free of cracks. Sensitization of the valve body material must also be guarded against. When the hard-facing is properly applied, the integral hard-faced seat provides a reliable and trouble-free design.

The use of a welded-in seat ring offers a number of advantages. First, the material properties desired at the seal surfaces are different than those desired for the valve body. By using a harder material for the seat, the cyclic life of the valve may be greatly extended. On the other hand, hardness is not a requirement of the valve body, especially if it is achieved at the sacrifice of ductility. The longer cyclic life afforded by the harder seat material will obviously reduce the frequency of lapping required to keep the seat-disc closure leak-tight and might obviate the need to replace the valve during the life of the process system.

From a fabrication standpoint, the use of a seat ring offers further advantages. Fabrication of the seat ring is easier to control because of the uniformity of material thickness and the smaller size of the item, and the material properties are closer to the desired conditions. If hard-facing is to be applied to the seating surface, the hard-facing procedure is better controlled when working with the smaller item. The separate seat ring is also easier to inspect by radiography, sonic, and dye penetrant methods than seats machined directly in the valve body. Finally, if the valve body is a casting of a globe valve with a line type of seal, even minor porosity at the lip of the seal would require extensive welding repairs or make the whole casting worthless. This problem is avoided by inserting a forged seat ring into the more porous casting.

Valves intended for low-temperature and low-pressure service often have replaceable seat rings to improve the speed and ease with which a leaking valve seat may be made leak-tight again. In theory, replacement of the seat ring is a quick and easy way of correcting a leaking seat condition. In practice, replaceable seats are frequently difficult to remove. At worst, such removable valve seats allow leakage around the back of the seat, thereby eroding the valve body and causing general deterioration that may require replacement of the entire valve. If replaceable valve seats are used, care should be taken in the design and fabrication of the seat, particularly the seal between the seat and valve body, to prevent the creation of another leakage path through the valve. Even in low-pressure low-temperature systems, replaceable seats can be the cause of leakage and short valve life.

In spite of their problems, there are a number of valves with replaceable seat rings on the market. This type of design has become more reliable in recent years because of the advances made in non-weld seal design. Metallic O-rings and similar types of gaskets now available have generally given good service. From a maintenance standpoint, replacing a valve seat appears to be easier and quicker than either lapping a valve or replacing it, even at the expense of more frequent maintenance. A case in point is the design of high-pressure steam traps. Since steam-trap service is particularly severe, the internals of steam traps do not normally last more than a few years. In such cases, the replaceable seats are a design requirement. At the same time, it is worth noting that maintenance of traps involving replacement of the seat requires care to insure that the repaired trap does not leak around the back of the new seat and exhibit worse leakage than before repair when the valve leaked by the eroded seat-disc contact surface. Most valves in most systems can be designed (allowing for maintenance) to be properly leak-tight for the full design lifetime of the valve. In many valves, the assurance of leak-tightness provided by a seat weld is an important factor in valve design even where good non-weld seals could be used.

There are a number of ways of fastening a seat in a valve. Replaceable type seats may be fastened with a bonnet hold-down or screwed into the body, as illustrated in Fig. 4.6. Non-replaceable seats are shrink-fitted, screwed, or fastened by welding. In high-temperature high-pressure valves, the seats are typically shrink-fitted and fastened by a weld, as shown in Fig. 4.7. When the valve does not have a compressed gasket between the seat and body, the shrink-fit type of fastening provides an easier method than the screwed type of fastening. It also

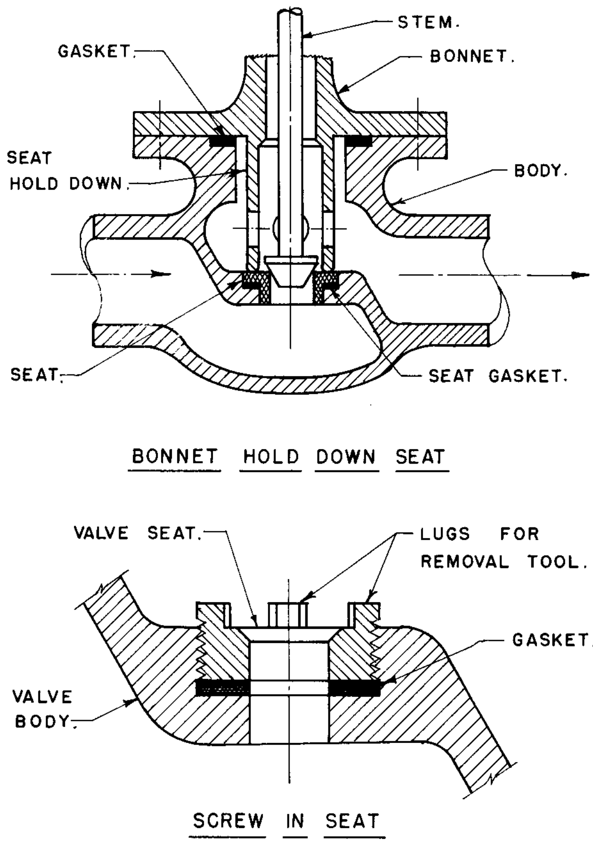


Fig. 4.6. Methods of Securing Replaceable Seats.

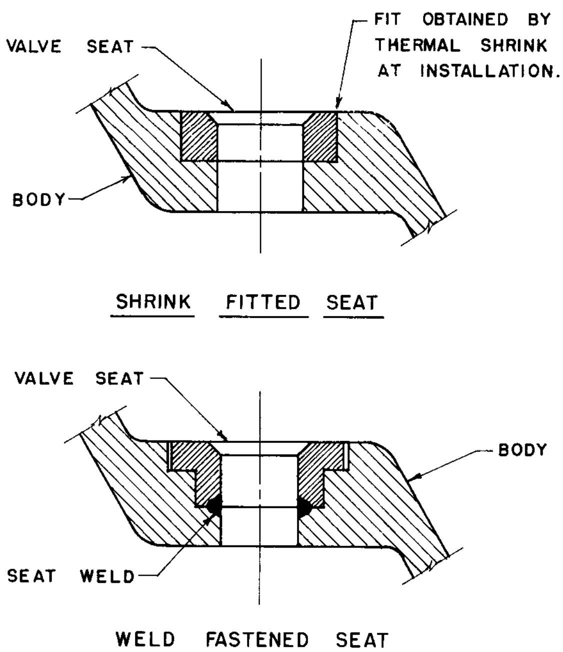


Fig. 4.7. Methods of Securing Non-Replaceable Seats.

resists loosening under thermal contraction conditions. Once fitted, a shrink-fit type of seat cannot be replaced. However, screwed-in seats are not easily removed either. Corrosion, galling, and freezing of the threads can effectively preclude seat removal. Screwed-in seats can also suffer thermal notch fatigue at the thread roots.

The seats of high-temperature valves must be designed to resist thermal stresses. Such thermal stresses can be quite severe and can lead to cracking of the seat and overstress of the valve body. Thermal stresses may be caused by the differences in temperatures and thermal expansion coefficients of the valve seat and body, or the disc seating force may be increased because of the differences in temperatures and thermal expansion coefficients of the valve body, bonnet, and stem. The design for the valve must include provisions for these thermal stresses, particularly if the fluid temperatures are subject to rapid fluctuations. In valves with screwed-in seats, the thread clearance should not be so small that an interference fit or high constraint is provided. In valves with shrink-fit seats, the degree to which the seat is compressively stressed should be limited so that thermally induced stresses will not crack the seat.

In valves with welded seats, the seat-to-body weld should not be exposed to the thermal stresses just discussed. Normally the seat is designed with a shoulder, as illustrated in Fig. 4.8, so that axial expansion is constrained by the shoulder and not the weld. When fitting up the seat in the body, the seat should be in full contact with the shoulder before welding. Since it is important that thermal stresses be kept off the weld, the weld is normally placed on the bottom of the seat adjacent to the seat shoulder. As indicated in Fig. 4.8, were the weld placed at the top of the seat, it would receive thermal stresses. Since it becomes difficult to place the weld below the seat in smaller valves and it must be placed above the seat for better accessibility, the seat design should allow for thermal expansion. In this case, the shoulder should be moved up to the top of the seat and a gap provided between the bottom of the seat and the body, as illustrated in Fig. 4.8.

Radial thermal expansion of the seat must be carefully controlled to preclude seat warpage and the consequent loss of leak-tightness.

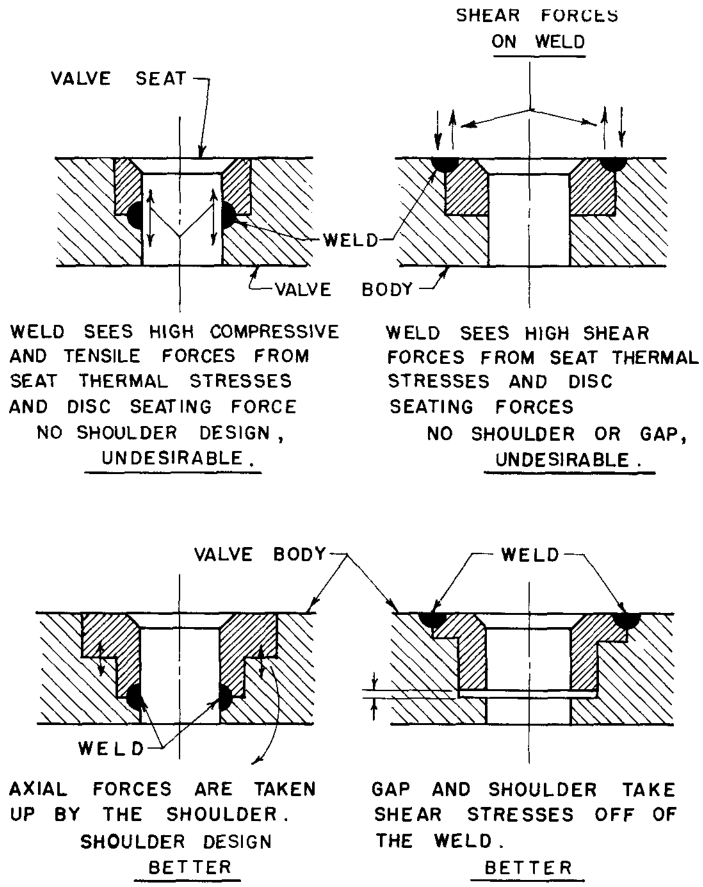


Fig. 4.8. Design for Shoulder in Valves With Welded Seats.

Seats are usually well constrained by the body, and thermal expansion in the radial direction is generally taken up by internal stresses with perhaps some strain ratcheting. Some attempts have been made to reduce the possibility of seat warpage by thermal effects through the use of a "floating" seat design, as illustrated in Fig. 4.9. Such a design provides an annular gap between the seat and the body to free the seat from radial body constraint. Where thermal expansion of the valve body can be uneven because of uneven body thickness etc., causing compression of the seat ring and warpage of it during thermal cycling, such a "floating" design has merit. Although successful from a standpoint of thermal stress, this design suffers from other drawbacks, including those normally associated with any deep crevice within a valve. In addition, there is the possibility that the weld will experience greater stress in this design.

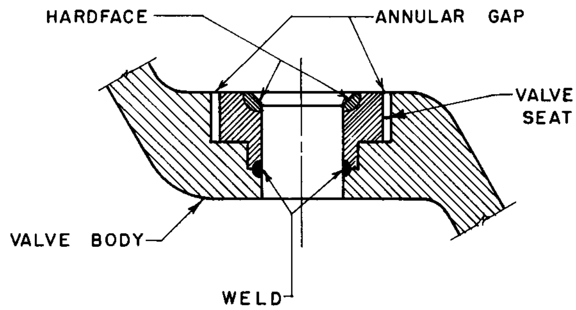


Fig. 4.9. "Floating" Seat Design.

(a) Flat-Faced Seat and Disc. Along with the material requirements for valve seats and discs, mating geometry is an important part of valve design for leak-tightness. Globe valve seats may be of either the flat-faced type illustrated in Fig. 4.1 (page 168) or the included seat angle type illustrated in Fig. 4.2 (page 169). The included seat angle types may have shallow angles or deep angles and differential angle features. The widths of the seat and disc mating surfaces may be small or large, and the width of the mating surface of the seat may be different from that of the disc. It may be important in some applications to allow for lapping the hard-facing as the lifetime of the valve progresses. The type of mating contact used should be considered in providing disc guidance since certain types of mating contact may have very low tolerance to a slight out-of-alignment condition of the disc.

The case for the flat-faced seat and disc geometry is straightforward. This geometry has the advantage of being simple and not susceptible to alignment problems. Contact is usually of the "area" type, and machining of the flat and right-angled dimensions to good tolerances is relatively easy. The seat can be machined and inspected to a given flatness tolerance. Almost no disc alignment is necessary since the disc is guided by the mating surface of the seat. Since area contact is used, lateral guidance is controlled only by the amount of overhang provided by the disc.

The disadvantages of the flat-faced type of seat-disc mating surfaces are also quite clear. First, the design is more amenable to the area type of seal. For a flat-faced type of seat and disc to have a line-contact pressure type of seal implies either a thin seat surface or a small differential angle on the face of either the disc or seat. A thin seat involves the risk of deformation and failure, and a small disc or seat differential angle takes away the advantage of flatness. Further, line-contact pressure seals require a certain amount of seat elasticity that is not easily obtainable in a flat-faced seating design. As a result, a slight out-of-flatness condition in a line-contact pressure seal on the part of either the seat or the disc would require excessive stem forces to achieve the necessary complete contact. For these reasons, the flat-faced type of seat-disc geometry generally employs area contact and a wide seat.

(b) Included-Angle Seat and Disc. Since mating conditions cannot be controlled precisely, valves with included-angle seats and discs are more amenable to line-contact pressure mating because of the additional mechanical advantage built into the seat and disc contact through the wedge design. Some typical included-angle valve seat configurations are illustrated in Fig. 4.10. However, slight out-of-tolerance conditions in the angles or the circularity of either the seat of disc can leave large portions of the mating surfaces out of contact and result in leakage through the seal.

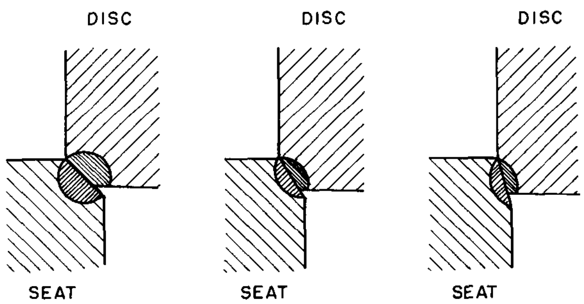


Fig. 4.10. Typical Included-Angle Valve Seat Configurations.

Included-angle seats and discs increase the contact pressure by factors of about 1.5 to 5 without the use of heavier and larger stems and yokes. Included-angle seats have greater elasticity than flat-faced seats because of the elastic properties of the inner diameter of thick-walled cylinders. Included-angle seats also have some well-known deficiencies. If the angle is narrow, the disc can seriously impair seating. Care must be taken to limit the applied seating force or the valve body may be overstressed as a consequence of the mechanical advantage inherent in the wedge design.

The mechanical advantage obtainable with included-angle globe valves as a function of the included angle is illustrated in Fig. 4.11. Surface friction, which reduces contact pressure, has been considered. This is an important fact for designs with relatively shallow angles since high friction can reduce the contact force. As may be seen in Fig. 4.11, the mechanical advantage increases rapidly as the included angle becomes larger than 40 or 45°, especially for the low friction factor curves. However, above about 60°, there is an increasing probability that friction can be high enough to cause sticking of the disc within the seat. The boundary at which sticking is apt to occur is superimposed on the graph. It is clear that above 70 or 80°, even surfaces with low friction can stick together.

The conclusions that can be drawn from the curves shown in Fig. 4.11 are quite clear. There is not much of a gain in using a shallow angle. Included-angle seats are practicable for included angles of from about 40° to about 70 to 80°. If all possibility of disc sticking must be precluded for accident design where thermal effects are present and the mating surfaces are roughened by erosion, the maximum practicable angle is about 55 to 60°. It should be noted that the friction forces occurring in valve seats result from very high contact pressures. In addition, such friction factors may vary over the wear lifetime of the surfaces if disc-seat galling occurs. The results of friction tests made on some materials under high contact force conditions indicate that while friction factors can be below 0.2 and even as low as 0.02, they will increase rapidly in the event of seizing or galling.[29]

Actual valve practice reflects these conclusions drawn from Fig. 4.11. Most included-angle globe valves have seat angles of approximately 45°. This makes substantial use of the mechanical advantage while precluding the possibility of disc sticking. Nor does it put a restrictive limitation on the design of the seat for resistance to overstress under the combined influence of stem and pressure forces.

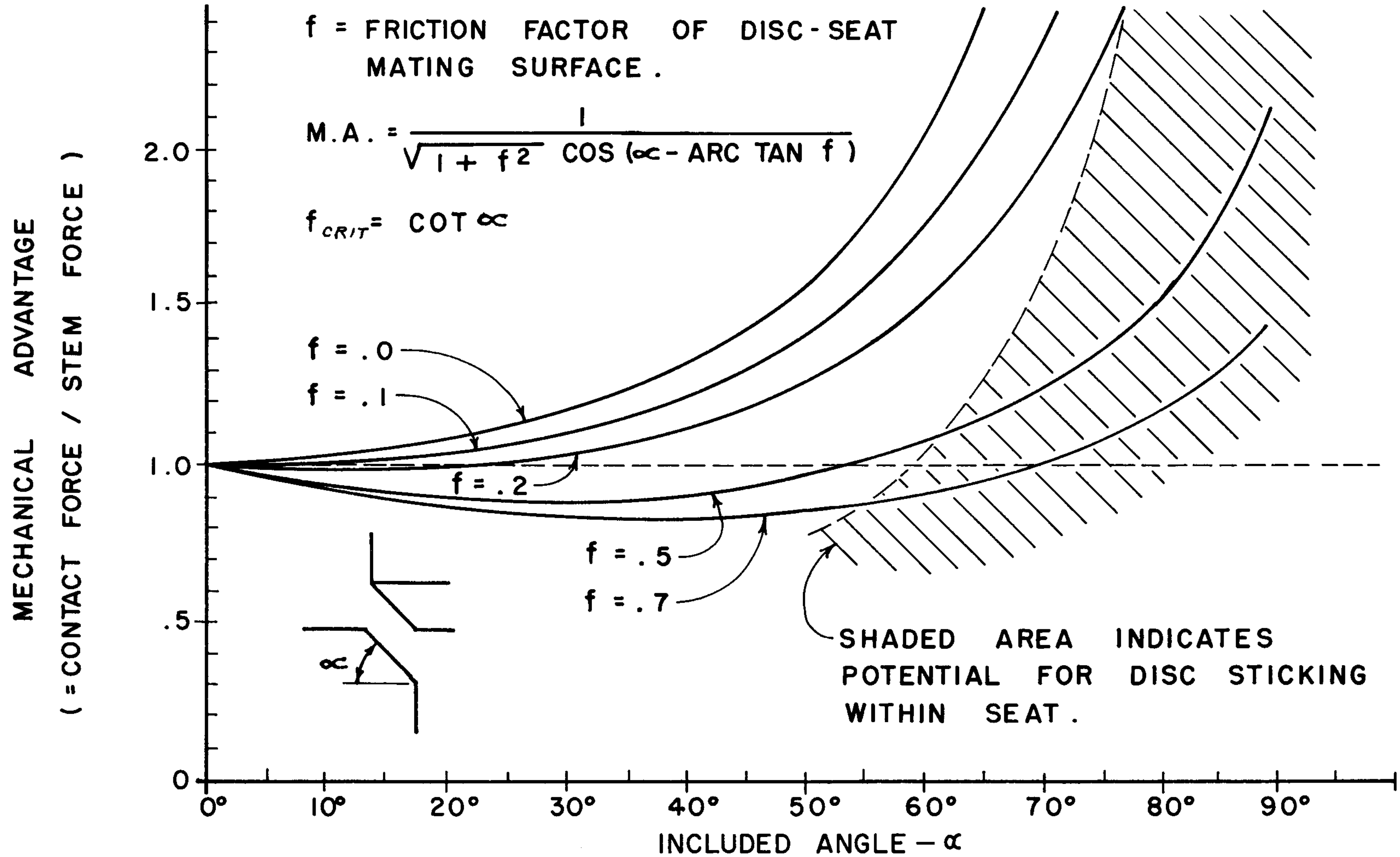


Fig. 4.11. Mechanical Advantage Obtainable With Included-Angle Globe Valve.

The maintenance to be performed on seats and discs should be given consideration in the design of included-angle globe valves. The cross-sectional dimensions of the seat and disc will change progressively after successive refinishing operations, as illustrated in Fig. 4.12. As may be seen, the hard-facing is applied to the seat in a manner to account for the included angle. Allowance should be made for several refinishing operations throughout the service lifetime of the valve with 10 to 20 mils removed at each operation. Since line contact, if provided, should be maintained throughout the lifetime of the valve, the small differential angle must be reestablished in the refinishing operation. This angle usually allows the line contact to occur on the lower or inside portion of the seat. Normal valve practice dictates that the differential angle be a positive one that allows contact on the inner edge.

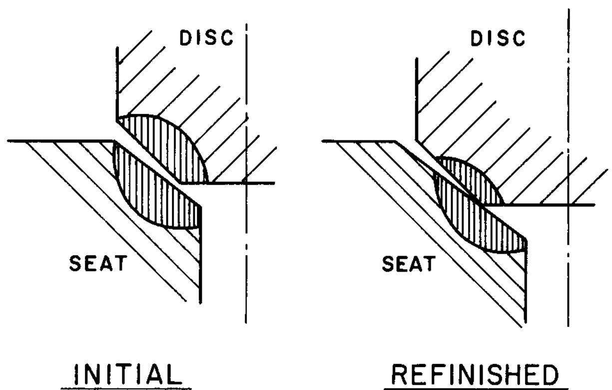


Fig. 4.12. Geometry Changes Caused By Refinishing Operations.

(c) Seat and Disc Alignment. With respect to alignment of the disc within the included-angle seat, the area contact type of disc is inherently more self-aligning than the line-contact pressure type. The two types of line-contact included-angle seat closures are illustrated in Fig. 4.13. One type employs the knife edge of the disc to achieve line contact, and the other has a wide disc sealing face but makes use of a small differential angle between the seat and disc to establish line contact. In either case, when the valve is being closed, the center lines of the disc and seat must be aligned when the seal mating surfaces come into contact. If the disc guide is not sufficient to do this, the disc will not contact the seat uniformly around the periphery of the mating

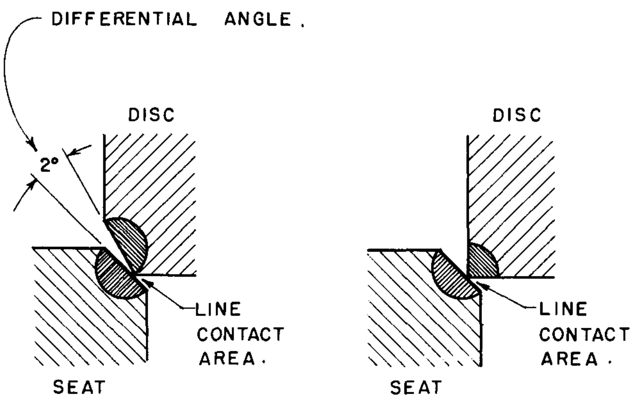


Fig. 4.13. Line-Contact Pressure Type of Included-Angle Seat Closures.

area and good pressure contact will not be obtained. The self-alignment properties of discs with line-contact closure are not as good as those of discs with area contact closure. The line-contact pressure type of included-angle globe valve seat requires disc guidance of both the stem and the seat for best results.

It should be noted that one disadvantage of the line-contact included angle seat without the differential angle is that lapping operations will tend to extend the area of contact. This tends to reduce contact effectiveness as well as the allowable time between lapping operations. Perhaps some margin could be gained by designing the seat and disc contact initially as a non-differential type of contact and proceed through lapping operations to the line-contact differential-angle type of seat and disc contact. However, this is somewhat exotic and more complicated than should be necessary.

Discs are sometimes designed to be self-aligning on the seat. This means that as the disc approaches the seat, a lower tailpiece or disc guide contacts the inner edge of the seat and aligns the disc seating area on the seat seating area, as illustrated in Fig. 4.14. Such a disc guide is considered useful for large valves where misalignment between the body and the bonnet could cause the disc to seat poorly. The disc guide normally has a slight taper and should be designed so that it cannot contribute to disc sticking within the seat. Disc guides can be used on globe and check valves but not on gate valves. Since these guides can provide an extra measure of leak tightness, especially for larger valves,

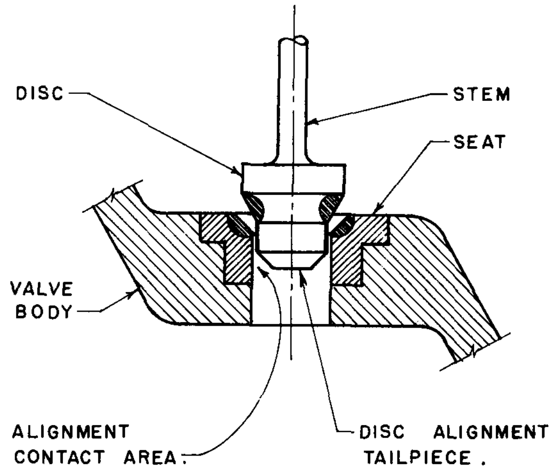


Fig. 4.14. Disc Alignment on the Seat.

greater seat tightness is usually achieved in globe valves than in gate valves.

If the disc is guided by the stem, more allowance in the guide design should be given for transverse disc freedom than angular freedom. Stem transverse misalignment and slight misalignment caused by cocking of the disc require take-up in transverse freedom, as is illustrated in Fig. 4.15. It should be noted that angular misalignment between the seat and disc will always occur, although it is usually small. The geometrical analysis of the reduction in effective contact is difficult to obtain

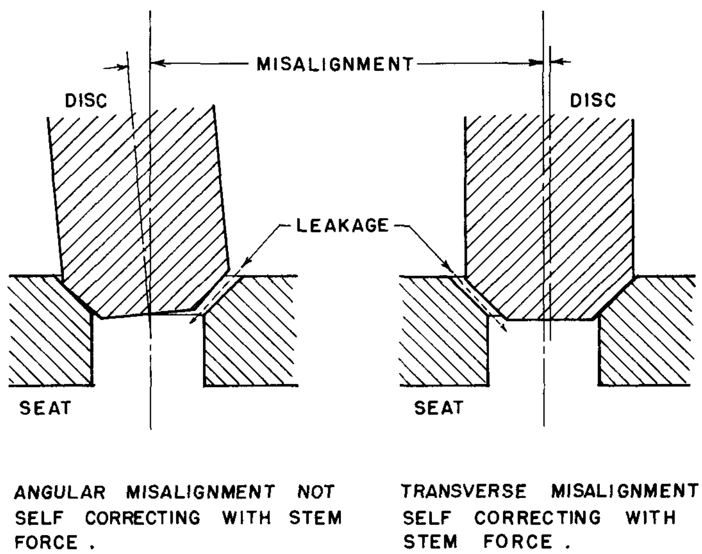


Fig. 4.15. Leakage Caused By Disc Misalignment.

accurately, but this type of effect can be easily measured in a function type of test performed during the initial valve leak-tightness qualification. If necessary, this type of test could be performed by conducting valve opening and closing cycles with the stem in a horizontal position and subsequently testing leak tightness.

The conclusions that may be drawn from the preceding analysis of seat and disc geometry are as follows. In high-pressure and high-temperature applications, the line-contact pressure type of seating with hard-facing is mandatory. An included-angle seat with a wide seating surface could reduce flow erosion, particularly at the seal contact line. Thus, a differential-angle line-contact type of geometry might be selected. The seat angle would be 45° to preclude the possibility of disc seizing and to reduce hoop stresses in the valve seat and body. This type of arrangement requires very accurate disc alignment. Therefore, the disc would be aligned on both the stem, with a slight amount of lateral allowance, and on the seat with an extended tailpiece below the seat.

(d) Disc Guidance. In mechanical operation of the valve, the disc is normally moved from a position out of the flow stream to a position in the flow stream and then to a position against the valve seat. Movement of the disc must be guided to preclude vibrations, disc cocking and hang-up, and to promote good and accurate seating. Within these guidlines there is some margin for slack. Some valve designs require a close-tolerance disc guide and others permit relative freedom of the disc. Whatever the valve design, there are usually very specific requirements for disc guidance.

Generally for all valves, the disc is guided either on the stem, the bonnet, the body, the seat, or on a combination of these parts. Some valves do not require disc guidance for seating purposes but merely to prevent excessive vibration of the disc while the valve is open. However, disc guidance is an important factor in the leak tightness of globe valves. The disc guides necessary to provide good seating of the disc and proper seat motion in globe valves are discussed here. The disc guidance for proper seating in included-angle seat valves is also discussed.

Since globe valve discs, especially the line-contact variety, require close guidance during valve operation, guide design is an important part of the valve. In smaller globe valves, the stack-up of tolerances is not large enough to seriously misalign the bonnet from the body of the valve, and the disc can be accurately guided from the stem. Virtually all globe valve designs have the swivel disc feature; that is, the disc is allowed to rotate with respect to the stem. This is important in the design of globe valves since it has been proved that sliding contact of seating surfaces does not contribute as much to leak tightness as it does to wear of the seating surfaces. This applies regardless of the size of the valve or the material of the seating surfaces. Swivel type of seating contact is therefore recommended for globe valve seats except for the backseats where heavy wear action does not occur.

Since swivel disc design is an integral part of globe valves, the dimensional tolerances of the swivel disc connection require careful analysis. Several considerations are uppermost. Since stem forces can be large, some wear can occur on the bearing surface between the stem and disc. Additionally, the dimensions of the bearing surfaces should be generous to prevent an excessively high contact pressure between the stem and disc. Otherwise, this area is subject to galling and seizing that result in loss of swivel action and rapid wear of the seat contact surfaces during valve operation. At the same time, the tolerances must be tight enough to provide good disc alignment and seating, especially if the disc is aligned off the stem. A careful balance of disc tolerances must therefore be obtained.

As previously noted, it is most important to preclude disc cocking and angular misalignment with the seat. A certain amount of lateral misalignment is allowable and necessary to account for bonnet misalignment with the body and to make up for whatever angular misalignment occurs. Given this fact, it becomes clear that the proper alignment can best be achieved by using a deep-disc design with the stem inserted deeply within the disc, as illustrated in Fig. 4.16. As may be seen in Fig. 4.16, not much angular misalignment can occur in this design and there is still some freedom for transverse or lateral misalignment.

In larger globe valves, guidance of the disc by the stem is less reliable because the stem itself may be misaligned with respect to the seat. This is brought about by the naturally larger stack-up of

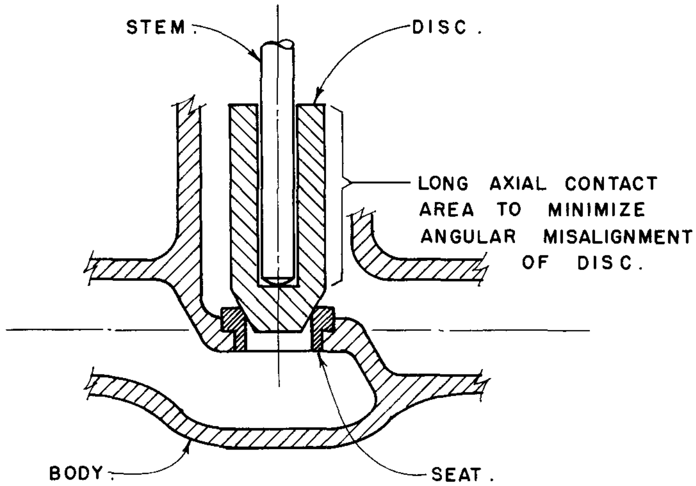


Fig. 4.16. Design for Globe Valve Stem-Guided Disc (Stop Check Valve Disc Illustrated).

tolerances between the seat, bonnet, and the stem. In these valves, other methods of guiding the disc are important. The disc may be guided by the body, bonnet, or by the seat itself. The disc may be guided off the body or bonnet of the valve by using the pressure boundary walls of the body or bonnet, as illustrated in Fig. 4.17, or by using a spindle

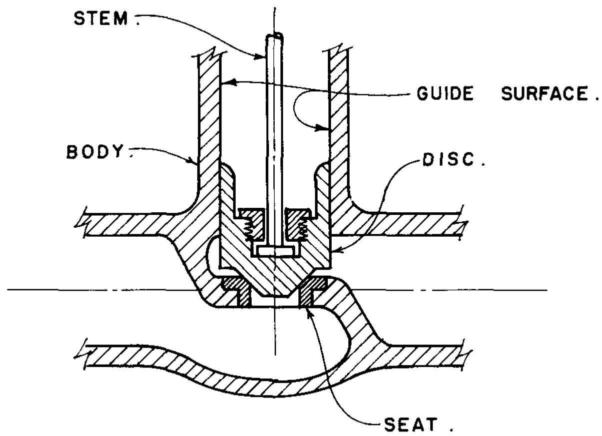


Fig. 4.17. Design for Body-Guided Disc Globe Valve.

arrangement, as illustrated in Fig. 4.18. Problems which may arise when these types of guides are used are of the cocking and sticking variety. Measures which may be used to preclude these problems include the provision of deep engagement between the guide surfaces at all times. In

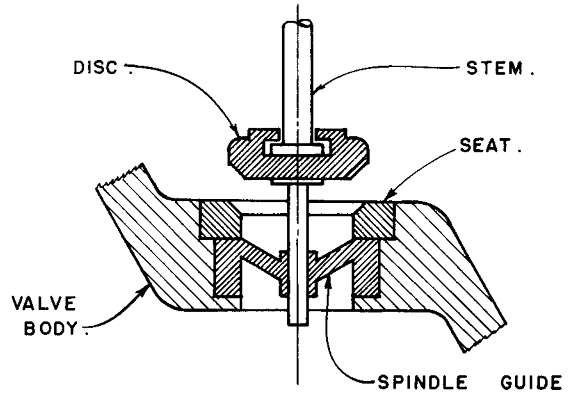


Fig. 4.18. Spindle-Guided Disc.

addition, corrosion must be minimized and entrapment of corrosion particles must be reduced. Carbon-steel wear surfaces are not satisfactory in this application, and they must be supplemented with a corrosion resistant liner. Stainless steel may be more satisfactory in this application, especially if the wear surface is polished.

Guide spindles are frequently used in throttle valves, and this spindle arrangement for disc guidance is generally quite satisfactory and trouble-free. It allows for a tight dimensional tolerance and at the same time provides for disc swivel freedom. The parts involved are not pressure boundaries and any wear does not subject the valve to the possibility of body leakage. Since the wear surfaces in this arrangement are smaller, it is not as difficult to provide hard-facing or a special liner for wear surfaces. On the other hand, the spindle type of guide is somewhat more difficult to arrange properly within the valve because of the complicated geometry involved. Unless the possibility of casting defects in the pieces can be accepted, internal fasteners and locking devices are necessary and these are undesirable because of the possibility of their coming loose and being swept into the flow stream.

Some success in disc guidance in globe valves has been achieved through the use of disc tailpieces that align off the valve seat. This arrangement has the big advantage that the disc is guided off the part to which it must be aligned, as illustrated in Fig. 4.19. If the tailpiece is not deep and tapered at the end, it can be subject to cocking and sticking caused by thermal cycling. In general, good results can be

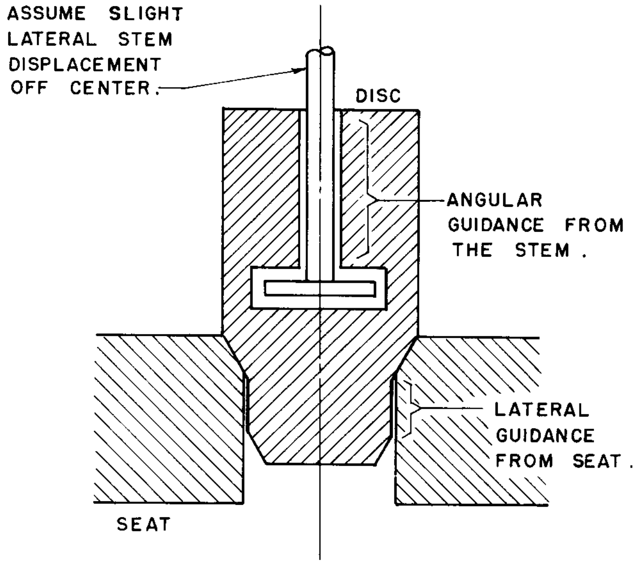


Fig. 4.19. Disc Guidance Off the Stem and the Seat.

obtained with this type of guide without the expense of providing a disc spindle arrangement. However, this type of guide may not be as desirable as the spindle type of guide for trip valves since the disc tailpiece disengages and the spindle does not. Engagement during rapid tripping action may either damage the seat or increase tripping time. These considerations can largely be eliminated through proper geometrical dimensioning and tolerance design, with proper control over the dimensions during the life of the valve.

4.2.4 Seat and Disc Design for Gate Valves

The seats for gate valves are shrink-fitted, screwed, or welded in place. In high-temperature and high-pressure gate valves, the seats are normally shrink-fitted and welded in place. The weld does not deteriorate in the operating environment, and it guarantees almost complete assurance of no leakage around the back of the seat. Once fitted, this welded shrink type of seat cannot be replaced, nor is seat replacement recommended for high-temperature, high-pressure valves. In fact, the geometry of all but the largest flanged gate valves does not allow accessibility for seat replacement. Design attention should therefore be focused on the hard-facing of the seat to permit the seat to be lapped without replacement although it is difficult to lap the seats of gate valves whose sizes are less than about 4 in.

Valve seats of high-temperature gate valves must be designed to resist thermal stresses, and the requirements are similar to those for seats in globe valves discussed in Subsection 4.2.3. The general approach is to design a seat with a generous axial length and to provide for a certain small amount of axial expansion or contraction during thermal transients. In shrink seats, the degree to which the seat is compressively stressed should be limited to allow some freedom of expansion axially. When a seal weld is used between the seat and body, the weld should not provide a constraint on axial expansion. The seat is normally designed so that the end rests against the body through a shoulder, and axial thermal expansion is taken up by the body rather than the weld. This is illustrated in Fig. 4.20. When fitting the seat in the body, the seat should be in full contact with the body before welding. In large gate valves, it may sometimes not be necessary to provide a shoulder and weld tab if the thickness of the seat is generous and the weld penetration is less than one-fourth of the seat thickness.

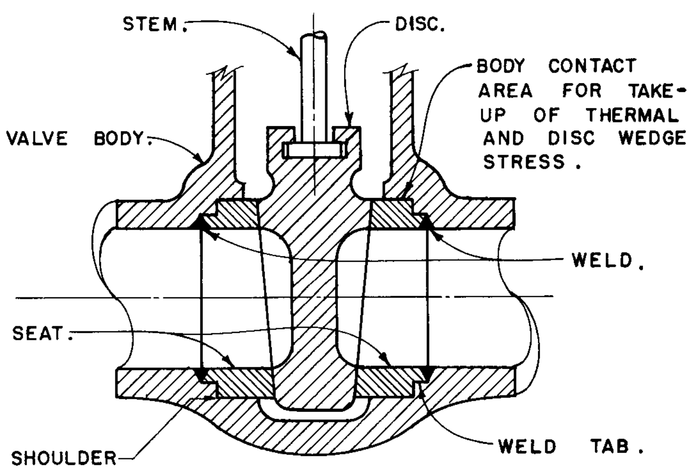


Fig. 4.20. Typical Gate Valve Seat.

Radial thermal expansion of the seat must be carefully controlled to preclude seat warpage and loss of leak-tightness. Seats are usually well constrained by the valve body. Thermal expansion in this direction is taken up by internal stress with perhaps some strain ratcheting. Such strain ratcheting in gate valves may not be as serious a source of seat warpage as is the case in included-angle globe valves because the strain does not alter the mating dimensions.

Gate valve closure is normally of the area contact type. It is difficult to obtain line-contact pressure seating unless thin seats are used, and these are subject to deformation and warpage under severe conditions. Difficulties are also encountered with the area contact type of seats since the larger seat area can accumulate greater quantities of foreign particles and dirt. Since there are difficulties in either design approach, the designer usually compromises by selecting a seat width which favors one or the other line of reasoning, depending upon the anticipated use conditions. That is why gate valves may be seen with relatively wide seats or relatively narrow seats, as illustrated in Fig. 4.21. In general, all of the preceding information is also applicable to flat-faced swing check valve seats and discs.

A note might be made here on seating surface wear in wedge and parallel face types of gate valves. Unlike globe valves, gate valve contact

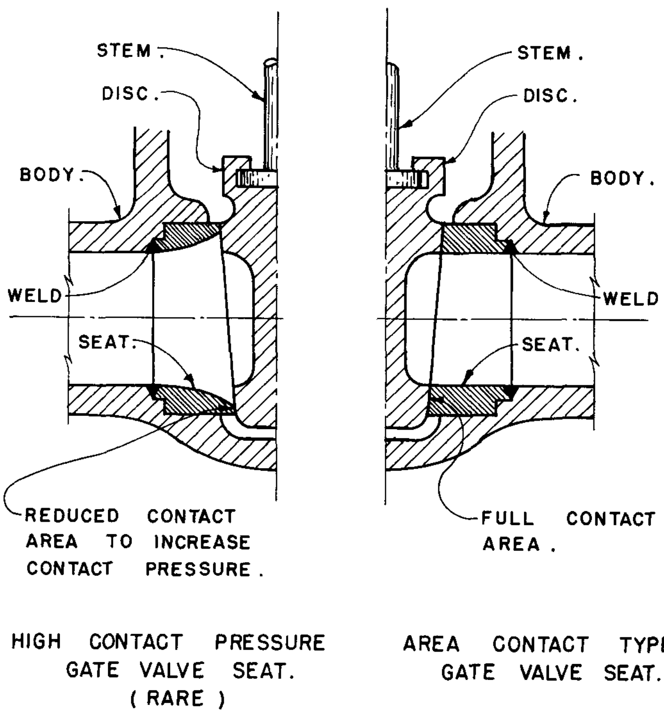
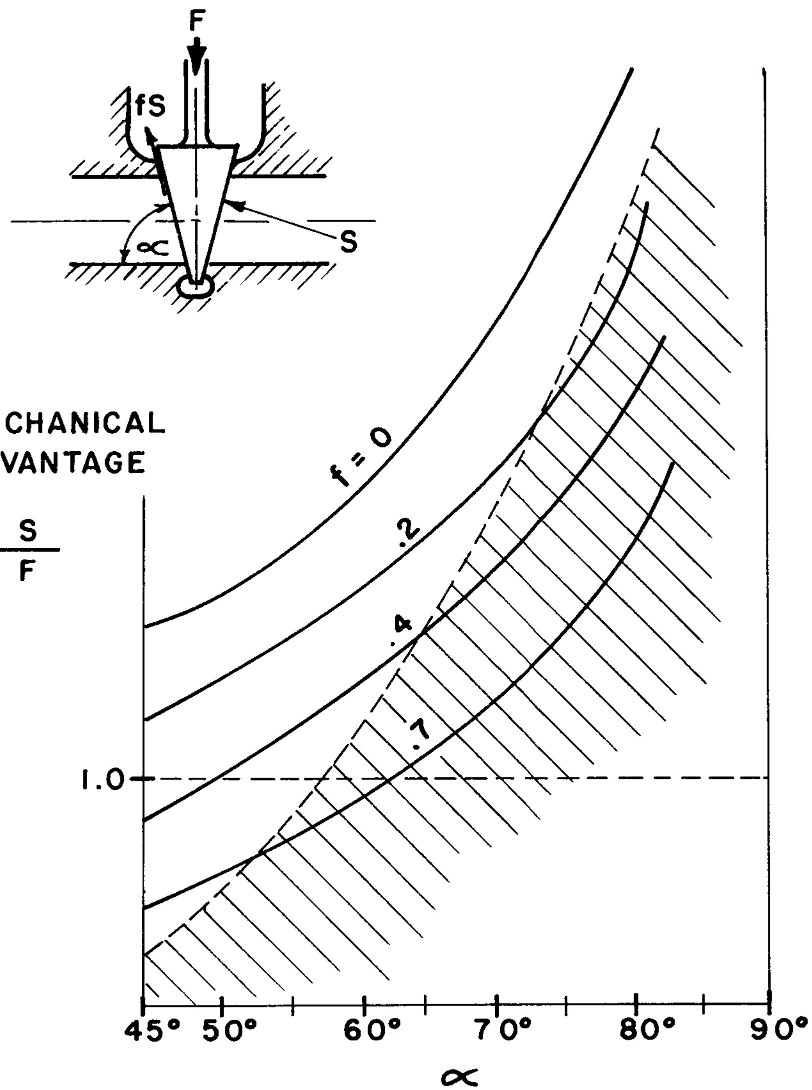


Fig. 4.21. High Contact Pressure and Area Contact Types of Seats for Gate Valves.

surfaces experience a certain amount of sliding contact under high bearing forces. Therefore, given the same number of open and closed cycles and the same material surfaces, the closure contact surfaces on gate valves may wear somewhat faster than those on globe valves. Normally, the use of good materials erases most of this disadvantage and the rest is probably masked by such effects as erosion across the seat areas. This wear disadvantage is also the reason why it is preferable not to depend on gate valves to sustain a large number of open and closed cycles throughout the life of the valve.

The amount of contact pressure being brought to bear strongly affects leak-tightness. In gate valves with a solid disc, the contact pressure for wedge shaped discs is higher by a large factor than for parallel-faced discs because stem force can be brought to bear to increase contact pressure. This is an important fact to remember when designing a valve with a solid disc since the wedge shaped disc can be effectively tightened to aid in reducing leakage while a parallel-faced disc cannot. However, either type can be reseated by cycling the valve open and closed to allow the momentary flow to flush the seat and also achieve a slightly modified seat-disc contact.

(a) Wedge Gate Valves. The wedge angle design in gate valves to eliminate sticking and increase the mechanical advantage to bring a higher contact pressure to bear on the seating surface does not follow the same considerations as in globe valve design. In gate valves, it is desirable and often mandatory to provide a spring or other elastic device between the discs to take up the squeeze imposed by the wedge angle. Dire consequences await the gate valve that does not have some such device. If the gate valve disc is a solid piece or back to back, the only salvation from disc sticking is in having a wide wedge angle. This makes for a larger valve, less mechanical advantage, and a difficult seat fabrication job. For such a design, a graph of the sticking and mechanical advantage possibilities of various wedge angles is illustrated in Fig. 4.22. As a final comment on this type of wedge disc design it should be noted that if the discs stick within the seat, thermal effects or excessive stem forces could cause an excessive splitting stress within the valve body



$$\frac{S}{F} = \frac{1}{\sqrt{1 + f^2}\,(\cos(\alpha - \arctan f))}$$

$$f_{out} = \cot \alpha$$

SHADED AREA INDICATES AREA IN WHICH THERE IS A POSSIBILITY OF DISC STICKING WITHIN THE SEAT.

LEGEND:

F = STEM FORCE.

S = DISC SEATING FORCE NORMAL TO DISC FACE.

α = WEDGE ANGLE. (SEE FIGURE)

f = FRICTION FACTOR.

fS = FRICTION FORCES PARALLEL TO SLIDING CONTACT FACES.

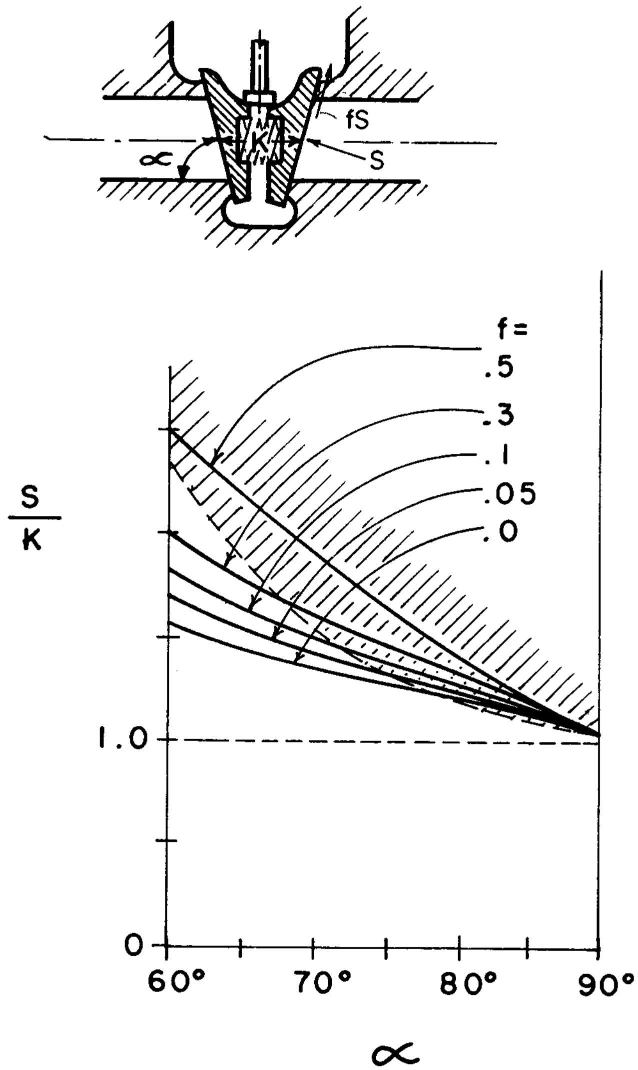
Fig. 4.22. Mechanical Advantage as a Function of Wedge Angle for Solid Disc Type of Wedge Gate Valves.

between the two seats. This limits the use of solid disc or split disc gate valves with no elastic member between the discs.

Normally, the large, high-pressure, high-temperature gate valves have a split wedge disc design, as illustrated in Fig. 2.4 on page 11, to provide flexibility between the two disc faces. The discs are also designed to be slightly flexible, acting like two heavy disc springs to force the disc faces against their seats. This helps to limit both the stem force and seat contact pressure force and helps preclude disc sticking caused by thermal effects or excessive stem forces. The spring force must be heavy so that seat contact pressure force will not be seriously reduced. The approximate effects that spring forces and spring constants have on contact pressures are illustrated in Fig. 4.23.

The design for a split disc type of wedge gate valve involves a close balance between mechanical advantage considerations and the elimination of disc sticking. High seat contact pressures involve a narrow wedge angle and a high spring constant. Elimination of disc sticking involves a wider wedge angle and a less heavy spring. By using the information illustrated in Figs. 4.22 and 4.23, by obtaining accurate friction factor information, and by determining the variation in the friction factor with constant pressure, the wedge angle and spring constant can be accurately designed for this type of wedge gate valve. The design is subject to verification, as a part of qualification testing, by a disc-sticking test. This would involve tests for the maximum closure forces required for all anticipated thermal transient conditions, both hot to cold and vice versa.

(b) Parallel-Faced Gate Valves. Parallel-faced gate valves may have a solid disc or a split disc. Most parallel-faced gate valves designed for high-temperature and high-pressure service employ a split disc design augmented by a wedge-shaped seating mechanism or spreader, as illustrated in Fig. 2.5 on page 12. The spreader forces the disc faces against their respective seats as the stem drives the wedges together. The relationship between the wedge angle, stem force, and seating force for the parallel-faced disc with spreader is illustrated in Fig. 4.24. This design has the advantage of increasing the mechanical advantage without requiring the wide V of a full wedge disc.



$$\frac{S}{K} = \frac{1}{\mathrm{SIN}\alpha - f\ \mathrm{COS}\alpha}$$

$$f = \mathrm{COT}\alpha$$

SHADED AREA INDICATES AREA IN WHICH THERE IS A POSSIBILITY OF DISC STICKING WITHIN THE SEAT.

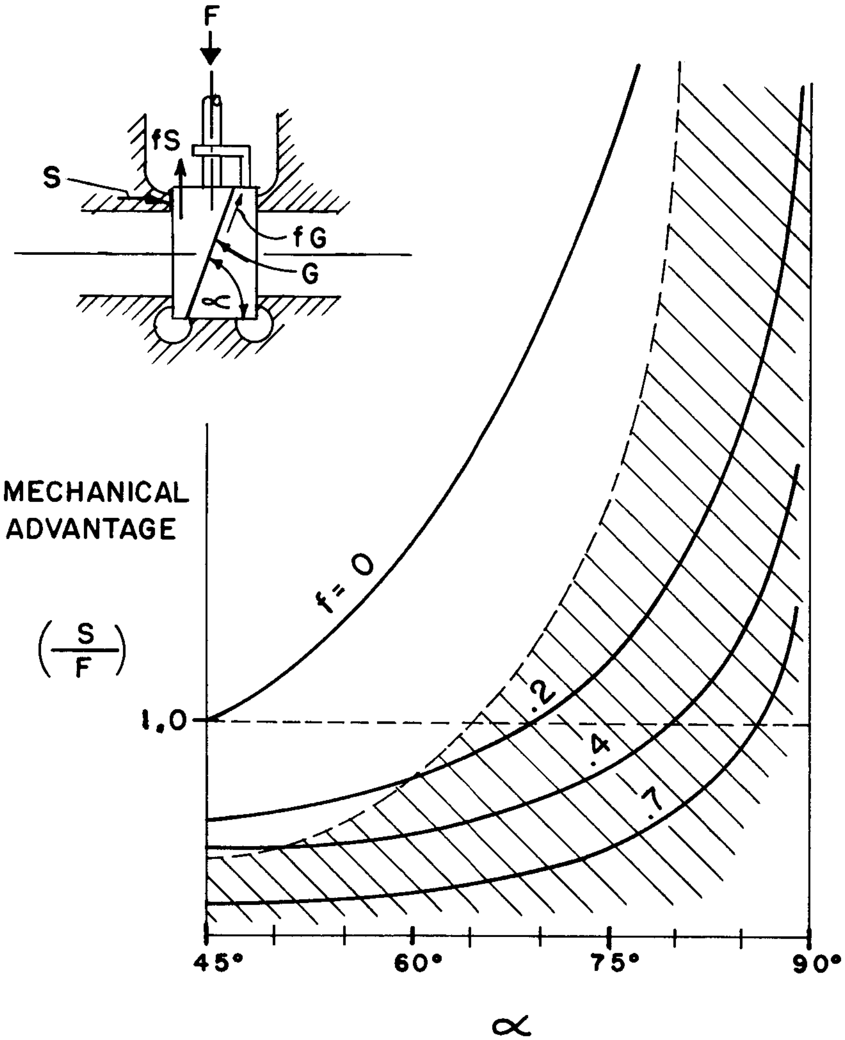
LEGEND:

K = HORIZONTAL FORCE BETWEEN DISCS DUE TO THE SPRING.

NOTE:

MECHANICAL ADVANTAGE IS DEFINED HERE AS SEATING FORCE NORMAL TO THE SEAT / SPRING WEDGE FORCE.

Fig. 4.23. Mechanical Advantage as a Function of Wedge Angle for Disc Spring Type of Wedge or Parallel-Faced Gate Valves.



$$\frac{S}{F} = \frac{\text{TAN}\,\alpha - f}{1 + f\,(2\,\text{TAN}\,\alpha - f)}$$

$$f = -\text{TAN}\,\alpha + \sqrt{\text{TAN}\,\alpha + 1}$$

SHADED AREA INDICATES AREA IN WHICH THERE IS A POSSIBILITY OF THE DISC STICKING WITHIN THE SEAT.

LEGEND:

F = STEM FORCE.

S = HORIZONTAL SEATING FORCE.

α = WEDGE ANGLE (SEE FIGURE)

f = FRICTION FACTOR (ASSUMED EQUAL AT ALL LOCATIONS WHERE SLIDING OCCURS.)

G = NORMAL FORCE BETWEEN DISC WEDGES.

fG, fS = FRICTION FORCES PARALLEL TO SLIDING CONTACT FACES.

Fig. 4.24. Mechanical Advantage as a Function of Wedge Angle for the Split Disc Type of Parallel-Faced Gate Valve.

Parallel-faced solid disc gate valves cannot utilize stem forces to provide assistance in contact closure. Seal contact pressure is obtained through differential pressure across the disc. This type of closure is typical of most check valves in general, and it becomes more attractive with large valves. Large valves have a high disc area-to-perimeter ratio and the disc differential pressure can apply a high seating force. The closure force from differential pressure in these large valves can amount to several hundred thousand pounds. In such cases, it is difficult to add significantly to this force with a high stem force unless the stem and operator are very large. In smaller valves, these high differential pressure closure forces drop off very rapidly and need to be assisted by stem forces aided by the mechanical advantages provided by seat geometry. In general, parallel-faced solid disc gate valves are large and are used for low-temperature and low-pressure service.

The large parallel-faced gate valves with a solid disc have a small amount of dimensional freedom between the disc and seat to allow for thermal expansion, as illustrated in Fig. 4.25. This means that the disc

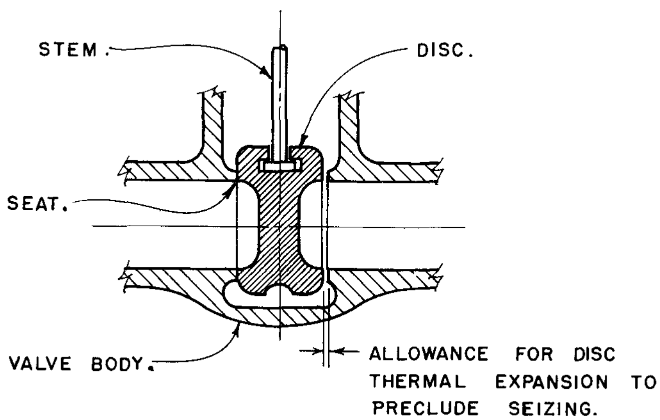


Fig. 4.25. Parallel-Faced Gate Valve With Solid Disc.

"flaps" back and forth when pressure is applied from alternate directions. In some designs, a spring-loaded false seat plate is used to hold the disc seated against the downstream seat during minor oscillations in system flow. This spring need not be as heavy as that used in the split disc wedge type of gate valve. In cases where the disc is designed so that the allowance for thermal expansion is limited and no elastic member is included, the valve should be functionally tested during qualification testing to confirm that the disc will not seize under thermal cycling conditions.

(c) Disc Guidance. Disc guides for gate valves usually consist of slots on either side of the valve body, as illustrated in Fig. 4.26. In wedge gate discs, the guides provide disc restraint after the disc has been lifted above the seats and has lost contact with them. Since there is usually some small pressure drop or flow across gate valves during valve operation, the valve stem could possibly be bent if the guides are not designed properly. The clearance of the guides, as increased by wear allowance, should not be great enough to permit stem deformation during disc motion. If this clearance criterion cannot be met without allowing a possibility of sticking, the disc guide surfaces may require hard facing to reduce wear. In any case, reasonably accurate calculations of the order of magnitude of wear expected can be made by considering the system requirements and obtaining wear information from such sources as the "Corrosion and Wear Handbook".[27]

Naturally, measures to preclude sticking of the guided disc should be used. The geometry of the guide and the guide pins or lugs should be

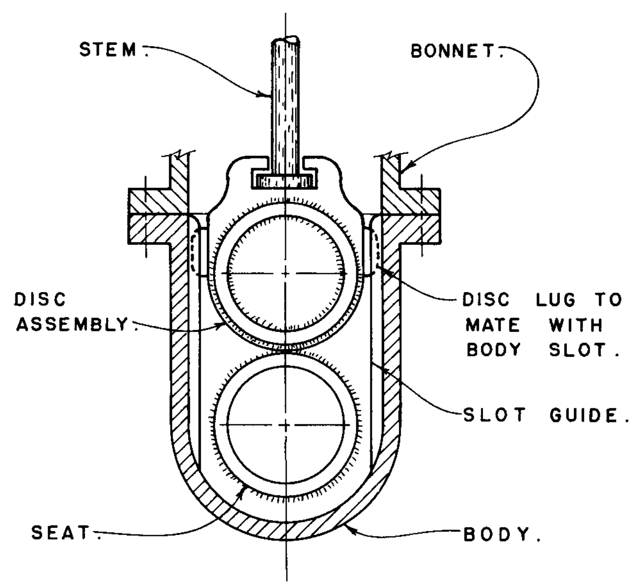


Fig. 4.26. Slot Disc Guides in Gate Valve Body.

rounded in a manner that will not permit cycling of the disc in any direction to cause seizing. This would be important if the valve was called upon to operate under high flow conditions where the disc would vibrate or flutter greatly. Unless the stem is large and the valve small, it is not advisable to guide the disc off the stem in gate valves. The large transverse forces upon the disc caused by fluid motion could distort or bend the stem, causing leakage through the packing. If there is any doubt, calculations should be made to prove the design. These problems are not shared by the parallel-disc gate valves because the disc is normally guided off the valve seat. Such guidance is sure, has a secondary benefit of wiping the valve seat clean, and makes use of the valve seat hard-facing as the wear surface.

4.2.5 Seat and Disc Design for Check Valves

In swing check valves or stem-assisted swing check valves, there is little which can assist the disc differential pressure in providing contact closure. Like parallel-disc gate valves, check valves depend upon fluid differential pressure to provide the required seal contact pressure.

A problem in swing check valves is that seat deformation can easily occur when the valve is closed quickly. After a sufficient number of closure cycles, the dents which form in the seat can compromise the leaktightness of the valve. It is desirable in stem-assisted swing check valves to provide some sort of dash-pot action in the operator to help take up the high closure energy. Another related consideration in check valve design is water hammer effects, which are discussed in Section 3.2.

The area type of seat contact is used almost without exception in swing check valves since the seats must be heavy to resist deformation caused by high transient forces during disc closure. Flat-faced disc and seat geometry is almost always used because some misalignment caused by disc hinge pin wear could conceivably damage the seat if an included-angle seat was incorporated in the design. The poor seating characteristics of the included-angle seat in a swing check valve are illustrated in Fig. 4.27. With respect to misalignment, the disc of a swing check

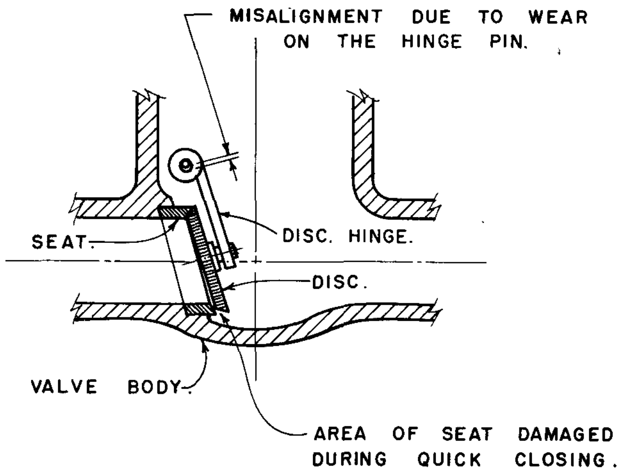


Fig. 4.27. Poor Seating Characteristics of Included-Angle Seat in a Swing Check Valve.

valve must have the largest amount of freedom possible. The disc should be connected to the disc hinge with a swivel joint which allows almost as much angular freedom as a knuckle joint. This allows the disc to freely "pancake" up against the seat for good closure. The disc hinge should only assure that the disc is well centered on the seat.

The only means of disc guidance used or necessary for swing check valves and stem-assisted swing check valves is the hinge pin which guides the swing arm. The disc can have a large degree of angular freedom, but its transverse or lateral freedom should be restricted somewhat. The disc hinge can have some freedom with respect to the hinge pin, but it must be capable of centering the disc within limits as it is seated. This is necessary to assure adequacy of seat-disc contact and retain satisfactory leakage characteristics during valve life. Because the hinge pin and disc hinge are wear surfaces, they should be hard faced and treated to resist wear. The hinge pin is the ultimate disc guidance device in swing check valves, and as such, its material and corrosion characteristics must be excellent.

A less common type of arrangement, a lift check valve, used for high-pressure and high-temperature service could be designed as follows. If leak-tightness is important, an included-angle seat design with a very narrow angle might be used. This can make up for the loss of stem force. Since line contact generally gives better service, the design would require some carefully engineered seat type of alignment. If flow stream interference is not important, a spindle guide above the disc might be used. However, preventive maintenance for this valve would include periodic disc replacement to preclude the possibility of wear-roughened contact surfaces increasing the friction factor and allowing disc binding in the closed position.

4.2.6 Backseats

"Backseat" in valve design is the designation given the valve seat within the bonnet that is provided to prevent leakage through the stem packing. The backseat is closed only when the stem is in the fully raised position and the valve is fully open. It is the third seat within gate valves. In addition to reducing or eliminating flow past the stem, the backseat performs the secondary function of providing a stem stop to decrease the possibility of stem ejection when the valve is opened fully. If the valve does not have a backseat, some type of stop should be included in the design to prevent the possibility of stem ejection.

As can be deduced from the preceding description, backseats are not subject to severe operating conditions. They are used to eliminate or reduce packing leakage or to provide a seat if the packing is removed and replaced while the system is filled with fluid. Backseats may be replaceable, as illustrated in Fig. 4.28, but more often they are not replaceable in that the bonnet material is used as the seat. In either case, it is important that backseats have hard facing. The non-replaceable backseats are usually made by hard facing the bonnet material directly. Sophisticated line-contact types of seal geometry are not often used for backseats, which are most commonly designed with a 45° angle and area contact.

The mating piece for the backseat is most often the stem itself, which is machined with a small angular shoulder at the proper angle to mate with the backseat, as illustrated in Fig. 4.28. Since this is part of the stem, which is usually of a corrosion-resistant and somewhat hard material, the shoulder is not often hard faced.

If the backseat is a separate piece, it should be welded into the bonnet, and the weld should be protected from thermal expansion stresses.

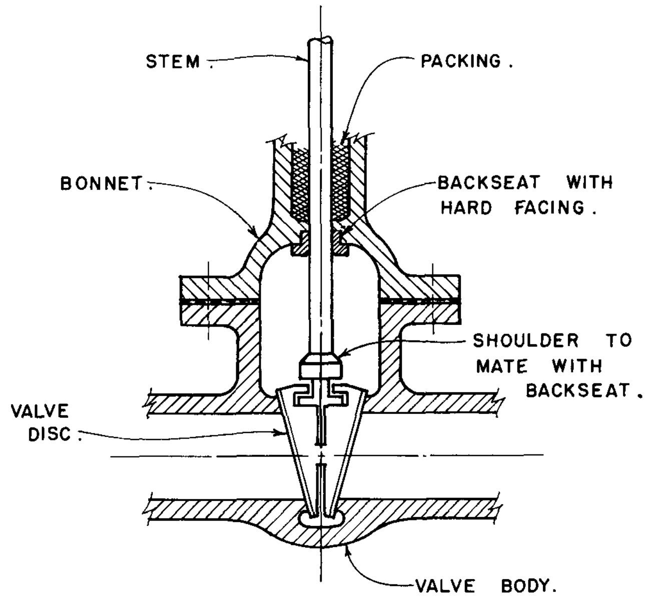


Fig. 4.28. Replaceable Valve Backseat.

In addition, it should not be allowed to contact the packing and receive stresses from packing tightening operations. Either shrink-fit or threaded backseat designs can be used, with recognition of the fact that a properly designed seat should never have to be removed. Maintenance can easily be accomplished by using lapping compound between the stem and the backseat mating surfaces while the bonnet is removed from the valve.

Backseats in globe valves receive more abrasive wear than those in gate valves because they are usually located above the swivel feature and seat with a rotating motion, while the stems in gate valves usually do not rotate. Utilization of the backseat during normal operation is at the option of the system operator. He may have the valve fully open and backseated or it may be turned in the closing direction a fraction of a turn from the fully open position. When the valve is used in radioactive fluid service, the stem shoulder is frequently seated hand-tight against the backseat to minimize or prevent leakage around the stem to the stuffing box. Seated in this manner, the backseat provides a backup barrier against leakage of radioactive fluid.

4.3 Stems

The valve stem is the mechanical linkage that translates external mechanical operation into opening or closing of the valve (seating or unseating of the disc), and it must penetrate the pressure boundary. This linkage may be of a rising-stem non-rotating shaft design, a rising-stem rotating shaft design, or of a non-rising stem design, as illustrated in Fig. 2.6 on page 13. In general, the rising-stem rotating-shaft design used in some globe valves is most likely to give trouble. Several different types of stem-to-disc connections are illustrated in Fig. 4.29 as being typical of the rising-stem rotating shaft designs in which a swivel disc feature is provided. It should be carefully noted that in such cases there is the potential for the disc to freeze to the stem, resulting in a loss of the swivel feature. Loss of the swivel feature would result in wiping of the disc on the seat with the consequent loss of leak tightness. Thus, the swivel feature is quite important to good leak tightness of

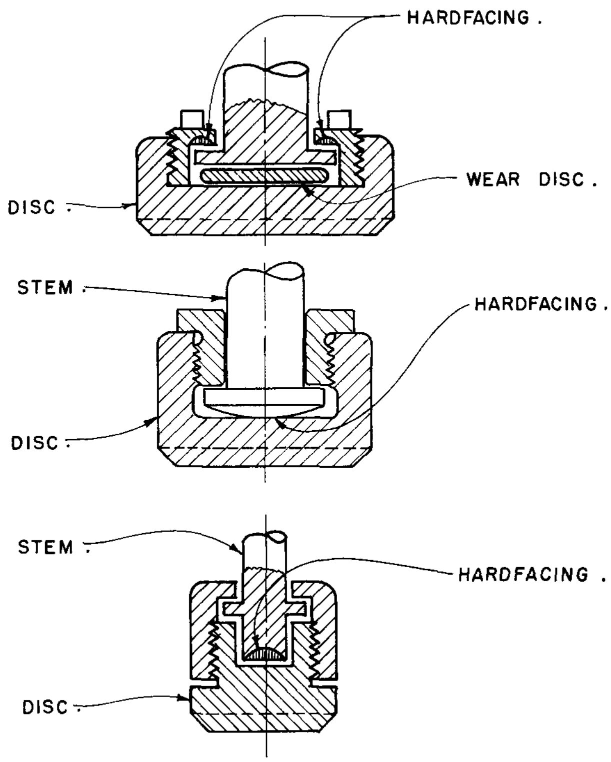


Fig. 4.29. Typical Stem-to-Disc Swivel Connections in Rising-Stem Rotating Shaft Designs.

globe valves, and it is recommended that some hard-facing or a wear resistant material be provided to reduce the amount of wear that takes place at this location.

The type of fastening between the stem and disc should be chosen to provide a high degree of simplicity and reliability. At the same time, it is important from a maintenance standpoint to allow the pieces to be unlocked and separated for replacement. In the environment of a system fluid, pieces in simple mechanical contact can gall and freeze, resulting in difficulties in separating these pieces. Generally, a screw type of locknut is used, and this locknut is locked with one of the locking devices discussed in Section 4.6. The clearances between the stem and disc within the disc cavity should be controlled in accordance with the design guidelines presented in Section 4.2 for good seating of the disc.

To perform its function effectively, the stem material must meet requirements for strength and ductility, machinability, corrosion resistance, and wear resistance. These requirements are discussed in the following subsections.

4.3.1 Strength and Ductility

The stem screw mechanism must translate the available torquing forces applied to a handwheel into an axial stem force sufficient to seat the disc. Thus, the thread design for the stem and yoke nut is selected to provide a mechanical advantage suited to this requirement. The yoke for large valves should be designed to permit ready removal of the yoke nut without removing the bonnet from the valve. Yoke nuts are made of a softer material than the stems, and this material is carefully chosen for ductility in the face of local yielding and usually it has self-lubricating qualitites. The majority of valves in high-temperature, high-pressure service have yoke nuts made of material such as nodular Ni-Resist iron, manganese bronze, or silicon bronze. Care should be taken to avoid the lesser grade brasses that may be subject to various forms of penetrating corrosion when stressed.

The strength and ductility requirements for the stem are imposed by several factors. These include the compressive load, tensile load, torsional load, and the impact load. The valve stem may also be subjected to excessive stresses by an overzealous operator or during installation. The stem must therefore be of a ductile material, and it must be capable of experiencing occasional and local overstress conditions without failure.

(a) Compressive Load. Compressive load is imposed when the disc is tightened against the seat, and this seating force is generally the largest compressive load on the stem. The high compressive load requirement results from the necessity for establishing leak-tight mating of the disc and seat and is dependent upon the

1. seat and disc material,
2. diameter of the seat,
3. angle of the seat,
4. contact area (a function of the width of the seating face),
5. quality of the surface finish,
6. alignment of working parts,
7. friction in stuffing box between stem and yoke, etc., and
8. operating pressure.

Minor compressive loads may also be imposed by fluid pressure or frictional drag forces in closing a valve. These tensile and torsional forces are in turn borne by the stem threads and transmitted through the stem threads to the yoke sleeve in the yoke. Acme or modified Acme threads are most often used to support the high axial loads.

(b) Tensile Load. The tensile load is imposed by the forces required to unseat a "stuck" valve, by the fluid pressure on opening a globe valve, and by the frictional drag forces encountered in opening a gate valve. The stem will also be in tension when the valve is back-seated.

(c) Torsional Load. Torsional loads are imposed by the turning of the shaft to raise or lower the disc. The maximum torque requirement is encountered when the disc is tightened against the seat, and a formula for computing this torque is given in Fig. 4.30. The acceptability of a stem design from the strength standpoint can be determined from appropriate formulae contained in such references as Marks' Mechanical Engineer's Handbook.

In a globe-stop valve, the torque on the handwheel rim must be sufficient to create a downward force in the valve stem at least equal to the line pressure to be held under the valve disc multiplied by the area of the disc. This force increases proportionally with the operating pressure. Deviations from the closing torques indicated may arise from differences in the trueness of the two mating surfaces, friction on the packing, and stem thread.

An average man can pull maximum of 200 lbs. on the rim of the handwheel.

The required torque to close the valve tight can be calculated from the following formula.

$$M = 1.2\ R\ T \tan(\alpha + \beta)$$

This formula is valid only for high pressures.

$$T = \frac{\pi}{4} p \left\{ d^2 + (D^2 - d^2)\ m \right\}$$

where M = required torque in in.-lbs. to close valve tightly with pressure action under the disc.

1.2 = coefficient for friction between stem and packing, stem, and disc.

T = axial force acting on stem in lbs.

α = thread angle

β = angle of friction between thread on stem and threaded bushing (the value of β is between 8° and 11°)

d = orifice diameter (inch)

D = chamfer diameter on the seat (inch)

p = pressure (psig)

R = radius of the middle of the area of contact between thread on stem and main nut (inch)

and m = sealing factor. This factor depends on the hardness, smoothness, and accuracy of the seating surfaces.

The Value of "m" in Standard Valves

m = 7 - 10 stellite to stellite

m = 6 - 8 for stainless steel to stainless steel

m = 5 - 8 for titanium to titanium

m = 4 - 6 for teflon to stellite

Fig. 4.30. Closing Torques for Globe and Angle Valves (from Ref. 30).

(d) Impact Load. Impact loads on the stem may be caused by the operation of an impactor handwheel or a "lost-motion" device when the valve is motor driven. These devices are described in Section 4.5, and they impose a torsional type of impact load on the stem above the screw mechanism. The pneumatic and hydraulic operators also impose an impact load on the stem as the disc slams shut against its seat.

4.3.2 Machinability

Ease of machining is important because of the need for a smooth finish to lower friction characteristics in both the threads and those areas of the stem that pass through the stuffing box and are in contact with the packing. A smooth finish is essential where the stem passed through the packing. Without it, the packing will have a short life and be difficult to keep leak-tight. Threads must be machined for operation of the stem in the bonnet or yoke and possibly for the disc and handwheel connections. In most cases, there is also the need to machine a backseat on the stem.

4.3.3 Corrosion Resistance

The need for corrosion resistance is self-evident. However, it is worthwhile to emphasize the importance of the corrosion resistance of valve stem material. The stem is subject to both the internal and external atmosphere and it must resist corrosion from both. Since corrosion would destroy the smooth surface where the stem passes through the packing, stem corrosion can also reduce the packing life. Where threads are internal, as in the case of a non-rising stem, the corrosion resistance becomes more significant because the threads cannot be lubricated and corroded threads would impair or possibly prevent operation of the valve. However, valves of large sizes generally have an outside stem and yoke design. The stem is also subject to crevice corrosion at the disc connection and from the crevice effect of the small clearances where the stem passes through the stuffing box.

4.3.4 Wear Resistance

Wear resistance is essential to insure that the stem can successfully withstand (1) the abrasive wear at the stem-disc connection when the disc is tightened against the seat, (2) wear from packing in the stuffing box, and (3) thread wear of the stem in the yoke. Good wear resistance characteristics are incorporated by selecting a material of sufficient hardness or by using suitable hardening techniques on the stem. Further discussion of material requirements is presented in Section 3.3.

4.4 Closures and Seals

From the user's viewpoint, the primary function of a valve is the control of fluid flow. In the performance of this function, the valve must also serve as a leak-tight pressure boundary for the contained fluid. In the case of gate valves, globe valves, and stop check valves, the flow control elements of the valve must be attached to some operating mechanism which penetrates the pressure boundary. In addition, the internals of all valves must be accessible for assembly and maintenance. Even check valves, which are automatically operated by flow conditions, must have provisions for accessibility to valve internals. These conditions require a minimum of one seal for all valves, and generally, two seals are necessary. The junction between the operating mechanism or stem and the valve body must have some type of dynamic seal to permit movement of the stem relative to the valve body. Since the access cover or bonnet has no movement relative to the valve body when the valve is in service, a static seal is provided at that junction.

As fluid containers, all valves are designed to minimize leakage. Nuclear valves often have special requirements for leak-tightness so that the maximum permissible concentrations of radionuclides will not be exceeded in the accessible areas of the nuclear plant. The restrictions on even very low concentrations of radioactive contaminants in the work areas or released to the atmosphere and plant effluent require that special attention be given this subject. The fluids handled in the various process systems of a nuclear plant will have vastly different levels of radioactivity. These possible levels of radioactivity in the process fluid in both normal operation and in accident conditions must be considered in establishing the leak-tightness requirements for the valve. For many reactor plant applications, the treatment of these closures and seals is directed toward the complete elimination of leakage; that is, achievement of a zero leakage condition.

In addition to the above requirements, valves incorporated within the containment isolation system frequently have a further special requirement. Valves and systems that penetrate the containment are required to exhibit leak-tightness against containment pressurization (ambient containment pressure must not leak into the valve and out to atmosphere through a system pipe), and the leak-tightness of packing and seals must be adequate to preclude leakage of radioactive gases as well as water or other system fluid. It has been pointed out in containment leak testing[31] that valve seals are typical offenders and require special attention during the leak test to insure that leakage does not occur as the containment is pressurized.

The two basic types of valve closures are covered in this section. The first type is the stem closure seal that is provided to allow the stem to penetrate the pressure boundary while minimizing leakage. Two types of stem seals are discussed. These are the packing stem seals and the sleeve stem seal. A third type of stem seal, the bellows, is discussed separately in Section 4.7. The second type of basic valve closure is the static seal between the access cover or bonnet and the valve body. Bonnetless valve designs, gasketed flanged joints, seal-welded joints, and pressure seal joints are discussed. This treatment of the static seal is also applicable to flanged joints connecting a flanged valve into the piping system although the use of flanged valves is held to a minimum in nuclear systems.[1] The structural (stress) analysis aspects of this static seal are discussed in Section 3.4.

4.4.1 Stem Seals

In designing stem seals, a balance is usually sought between frictional stem resistance and stem seal leak-tightness with additional consideration given to minimizing maintenance requirements. Resistance to the motion of the stem can be as high as 200 to 500 lb of force. This is undesirable in certain valves such as trip valves which must close quickly and surely. At the same time, leakage should be minimized or eliminated entirely for nuclear applications. The periodic maintenance requirements of the stem seal design chosen should also minimize valve and system outage. The types of stem seals considered to meet these requirements are both single and double packing seals and sleeve seals.

(a) Packing Stem Seals. The packing type of stem seal is the most frequently used dynamic seal and has the largest amount of operating history. Single and double packing seals are illustrated in Fig. 4.31. Packing materials should not be subject to decomposition at operating temperatures and should have self-lubricating qualities. These and other considerations, including the effect of radiation on packing materials, are discussed in Section 3.3.

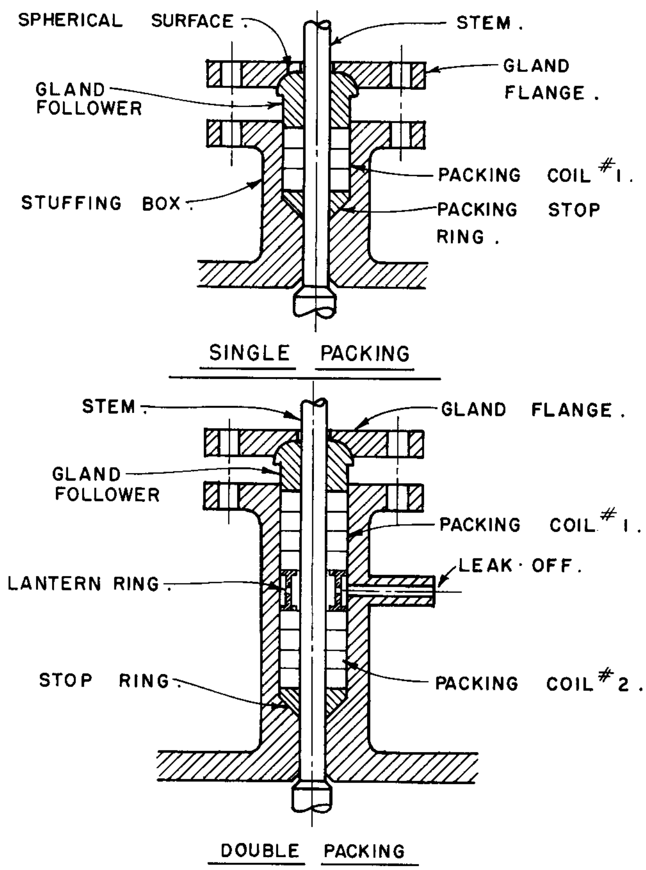


Fig. 4.31. Single and Double Packing Stem Seals.

The packing stuffing boxes should have ample depth for as many as six or more turns or packing. The depth of the stuffing box may be increased beyond this but the gain from additional packing diminishes beyond this point. This is because the greatest compressive stress in the packing occurs adjacent to the gland follower and diminishes throughout the rest of the packing so that the addition of more packing reaches the point of diminishing returns, as illustrated in Fig. 4.32. The bearing surface between the gland follower and the gland flange is spherically finished, as illustrated in Fig. 4.31. This is important in facilitating even tightening of the packing by torquing the gland flange nuts. A packing stop ring is provided at the bottom of the stuffing box in Fig. 4.31 to minimize the chance that the fibrous packing will be carried along with the stem and wedge between the stem and bonnet, binding the stem.

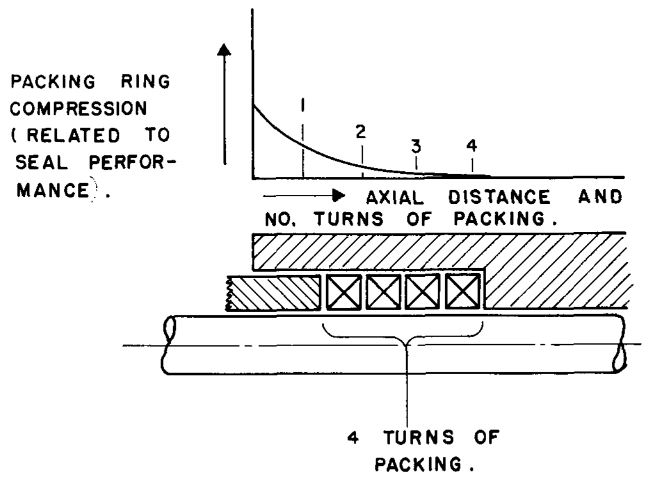


Fig. 4.32. Packing Compression.

The packing type of stem seal may be designed with a double packing arrangement provided to collect leak-off. This is accomplished by providing a special ring to separate the two sets of packing, as illustrated in Fig. 4.31. It is especially important that such packing be self-lubricating because the system flow may not be available to lubricate the packing in the upper part of the stuffing box. Older designs for packing did not have a self-lubricating feature and depended upon the system flow for lubrication. During periods of maintenance when the system was shut down and dry, the packing would dry out and permit leakage when the system pressure was reestablished. Self-lubricating packing improves the surface contact characteristics between the stem and packing and thereby enhances leakage characteristics.

Summarizing, packing stem seals must have good ductility and lubricity. The former permits close conformance of the seal material to the mating surface, while the latter permits this close conformance without undue friction. The ductility characteristic must be achieved while keeping sufficient structural shape to prevent any tendency of the packing material to extrude into narrow openings, such as between the stuffing box and gland follower. A good example of where these compromises must be achieved is a containment isolation valve that must pass the containment leak test while meeting the required operating time for containment isolation. The packing type of seal generally operates satisfactorily, and the size of the valve does not affect the efficiency of the packing in achieving leak-tightness.

(b) Sleeve Stem Seals. The necessary leak-tightness and sealing actions of sleeve seals are achieved through the use of very close dimensional tolerances and surface finishes between the hard metal sleeve and the surface of the stem, as illustrated in Fig. 4.33. As may be

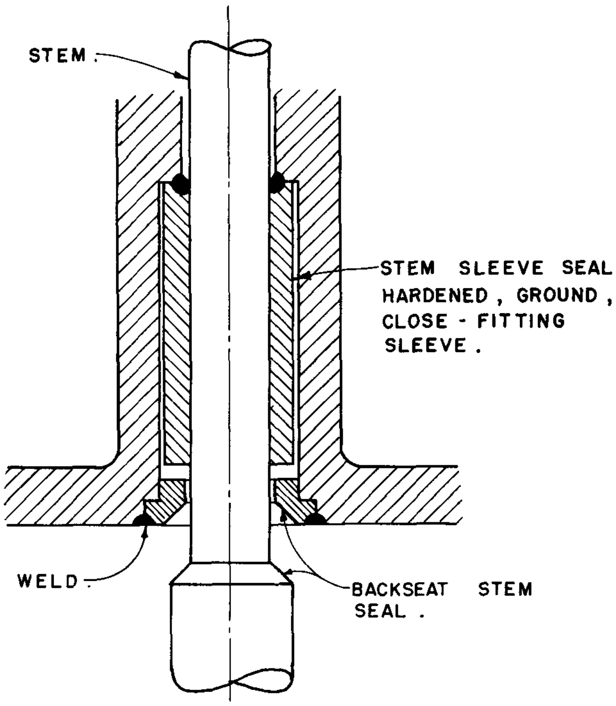


Fig. 4.33. Sleeve Stem Seal and Backseat Stem Seal.

expected, some leakage is always present and the use of backseats and leak-offs is necessary for improved leak-tightness. Sleeve seals offer considerably less resistance to stem motion than packing stem seals, but they have the disadvantage that thermal effects may play a large part in the degree of resistance. It is therefore important in designing sleeve stem seals to match the thermal coefficient of expansion of the stem sleeve with that of the stem.

From an upkeep standpoint, the sleeve type of stem seal is practically maintenance free. From a design standpoint, stem sleeves usually increase the bonnet length of a valve because of the necessary length of the sleeve required to obtain high leakage flow resistance. In addition, the tight clearances between the stem and the sleeve may permit crevice corrosion at that point.

4.4.2 Access Cover or Bonnet Seals

The leak-tightness requirements for the joint will dictate the design for the static seal between the bonnet and body of a valve. Depending upon the level of radioactivity in the system fluid, nuclear valves are often designed as one of the following three types.

Type I valves are hermatically sealed to prevent leakage of the system fluid to the ambient under normal operating conditions by seal welding all static seal closures.

Type II valves limit leakage of the system fluid to the ambient through the use of gaskets. This type of valve is also designed so that it may be converted to a Type I valve by seal welding all static seal closures in the event of leakage.

Type III valves limit leakage of the system fluid to the ambient through the use of gaskets, but they have no provision for seal welding the static seal closures.

The designs used to meet these leak-tightness requirements for the static bonnet-to-body seal include the bonnetless valve designs, gasketed flanged joints, seal-welded joints, and pressure-seal joints.

(a) Bonnetless Valve Designs. Some valve vendors have attempted to eliminate bonnet-to-body joints through the use of "bonnetless" valves, as illustrated in Fig. 4.34. This usage has been restricted to valves with sizes less than 4 in. NPS. Although these bonnetless valves appear to have potential, limitations on accessibility and certain experiences with them have caused avoidance of this design.

The reasons for this avoidance are clear upon study of Fig. 4.34. First, it should be noted that although there is a backseat in this design, there are two annular leak paths and the backseat seals only one of them. The leak path through the stem sleeve thread is not covered by the backseat. Such leakage is also in a location that makes corrective maintenance difficult to perform when a leak occurs. There is a

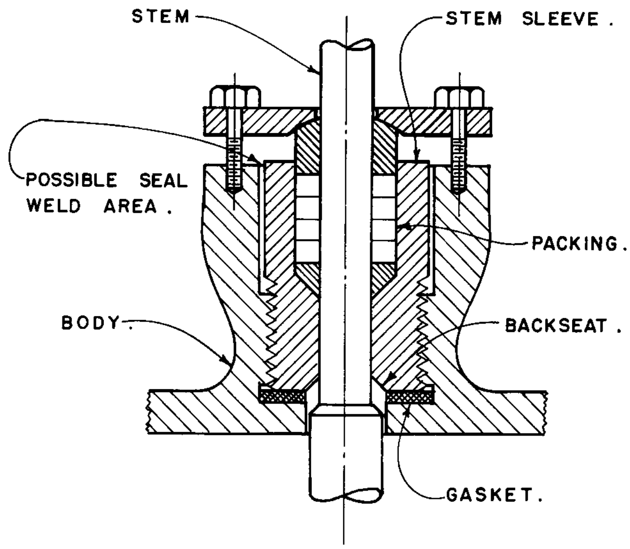


Fig. 4.34. Bonnetless Valve.

possibility that this leakage disadvantage of bonnetless valves can be remedied by welding the annular leak path in the area illustrated in Fig. 4.34. This seal weld would put an additional burden on the design and possibly make it uneconomical over the long run because of the problems involved in maintaining the internals of the valve.

Second, there is always some potential that the stem will unscrew during normal valve operation. Therefore, the stem sleeve must be provided with some type of locking device to prevent its rotation with stem movement. Third, the bonnetless design makes access to the seat difficult for initial machining and for any subsequent maintenance even in the absence of the seal weld previously discussed. This lack of accessibility is a major disadvantage of the design because it restricts the size of the internal components and limits access to them for maintenance. In an effort to provide more accessibility to valve internals, some vendors provide an access cover on the underside of the valve opposite the stem penetration. This access cover must also have a seal closure, so much of the advantage of a bonnetless design is erased.

In summary, the bonnetless valve design may appear to eliminate the problem of bonnet joint leakage, but its disadvantages in practice generally restrict its application to only the smaller valves.

(b) Gasketed Flanged Joints. By far the most common type to bonnet to-body closure seal is the gasketed flanged joint, as illustrated in Fig. 4.35. The gasketed flanged joint designs discussed here conform to the requirements for Type III valves. There has been extensive experience with gasketed flanged joints, and this type of closure seal is frequently used in power plant systems. In addition, the spiral-wound gaskets commonly used in these closure seal designs demonstrate a high degree of reliability.

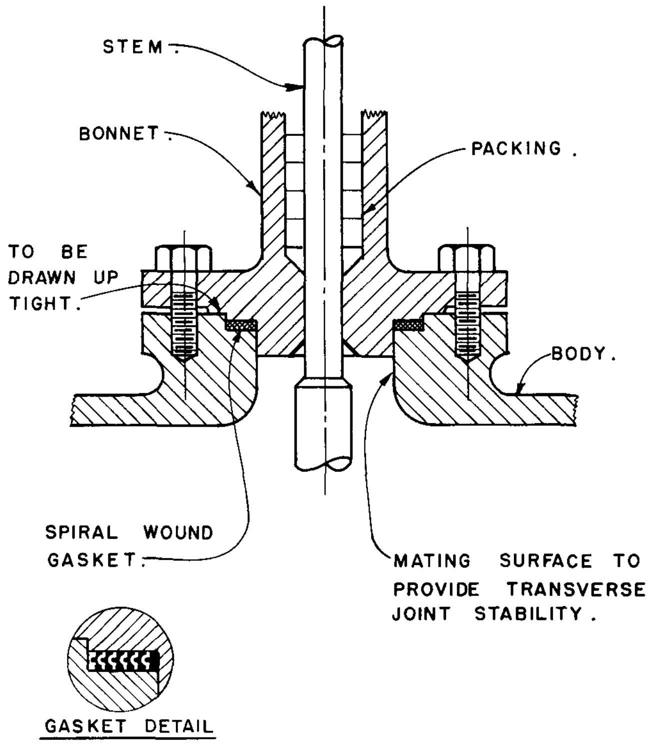


Fig. 4.35. Gasketed Flanged Bonnet-to-Body Joint.

The most common type of gasket used is the spiral-wound "Flexitallic" gasket that has either teflon or asbestos between the chevron-shaped stainless steel strip. These gaskets are cheap, rugged, and generally give good service. The chief cause of leakage or inability to seal is failure to use a sufficiently stiff gasket with adequate bolting to compress it to the proper seal thickness. These gaskets require a high compression pressure to seal leak-tight.

If this type of gasket is used, the joint should be of a proved design such as those given in the ANSI Standard B.16.5, Steel Pipe Flanges and Flanged Fittings, or the joint should be subjected to a prolonged seal leakage test to confirm its leak-tightness in service. These spiral-wound gaskets should not be used unless there is adequate bolting, and the tolerances of the body and bonnet gasket capture area should allow little or no transverse freedom. Any transverse motion of the bonnet with respect to the body while the valve is sealed and under pressure can start leakage. The flanged surfaces must be drawn up metal-to-metal (achieving rigidity) or the closure seal will not be leak-tight in all cases. The design illustrated in Fig. 4.35 shows the need for accurate machining of the flange surfaces which capture the gasket to provide a fixed amount of gasket compression when the flanges are drawn up. In very small valves where small bolt-circle diameters permit few bolts and even less tightening, there seems to be a weakness with spiral-wound gaskets in that slight over-compression on one side of the gasket relieves gasket pressure in other areas, opening up leakage paths. The resulting low compression areas are caused by an inability of the gasket to "follow" the flange surfaces accurately in small valves. This phenomenon is illustrated in Fig. 4.36.

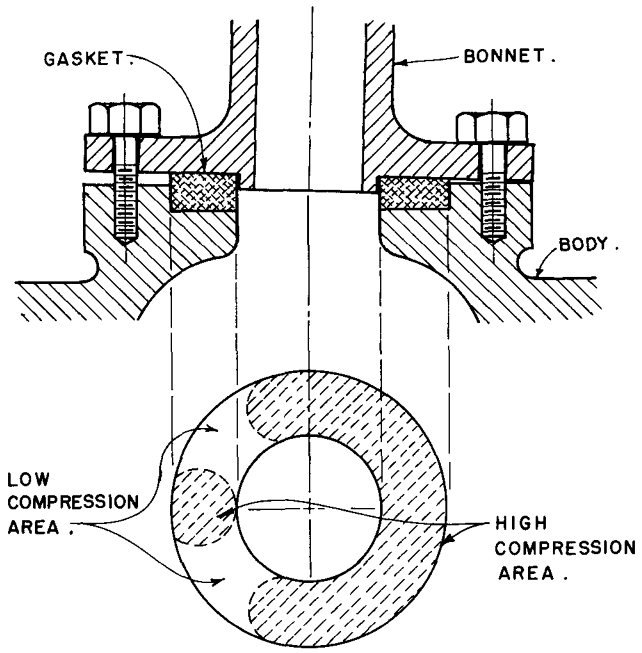


Fig. 4.36. Spiral-Wound Gasket in Small Valve.

Another type of gasket that has been proved in service is the hollow metallic O-ring. These gaskets are generally plated with a softer metal, such as silver over stainless steel, to improve their sealing properties. Often, they are self-energizing; that is, they have small passages from the pressurized side of the O-ring into the seal tubing and thereby use the system pressure to hold the gasket seating surfaces tight against the flange seats, as illustrated in Fig. 4.37. These gaskets require less

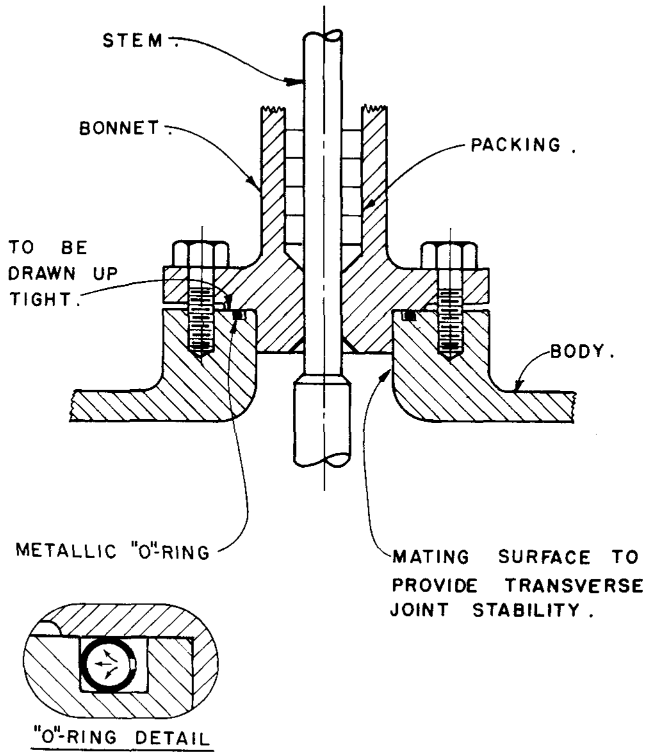


Fig. 4.37. Flanged Bonnet-to-Body Joint With Metallic O-Ring Seal.

space and bolting (compression) force than the spiral-wound gaskets. They also require close tolerances and a high surface finish (16 AA) on the mating flanges to effect their seal. The gasket manufacturer's recommendations on groove dimensions and tolerances should be followed. Metallic O-rings are suitable for high-pressure and high-temperature conditions and give good service when properly installed. However, the O-rings are easily damaged during handling, and considerable care must be taken to protect the flange sealing surfaces from scratches etc., whenever the closure is open.

To summarize, if flanged gasketed closure seals are properly designed, they are highly reliable. It should be noted that the closure joint of a valve bonnet differs from the closure joint between sections of pipe. Valve bonnets are subject to pressure forces and to stem forces that can be quite high under conditions where high closing force is being applied to the valve. The walls, including reinforcement, of the valve body are quite a bit thicker at this point than are those at piping joints. In the higher pressure and temperature ratings, the large flanges and heavy bolting necessary for valve bonnet-to-body closure add greatly to the size and weight of the valve. Perhaps the largest disadvantage of this type of closure seal is the fact that an increase in valve size and bonnet diameter and an increase in system pressure work against the reliability of the seal.

In large, high-pressure valves and especially in welded steam systems, the forces on the bonnet can be quite high so that even elastic deformation of the bonnet can open up gasket leak paths, as illustrated in Fig. 4.38. Further, at high temperatures and over long periods of time, the

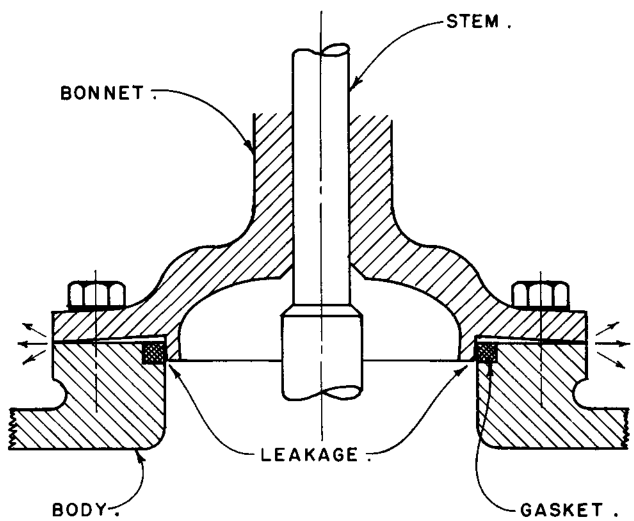


Fig. 4.38. Bonnet Flex as a Cause of Leaks.

studs are subject to a certain amount of creep. This in effect relaxes the axial stress of the stud that is relied upon to apply gasket compression for a good seal. Periodic torquing of the closure bolts with a shim inserted between the flanges to measure gasket compression will minimize this problem. If the stud nuts are not re-tightened periodically, there is the potential that eventually such closure joints will begin to leak because of stud relaxation. It is for this reason that the use of gasketed flanged joints is being suspended in high-pressure, high-temperature steam systems in many cases in favor of the pressure-seal type of bonnet joint.

(c) Seal-Welded Joints. The stringent leak-tightness requirements for some nuclear valve applications make it necessary to seal weld the static seal joint between the bonnet and body of the valve. Modifications to the designs for gasketed flanged joints to make them suitable for seal welding required of Type I and Type II valves are described in the following paragraphs.

Type I valves are hermetically sealed to prevent leakage to the ambient under normal operating conditions. This requires that the joint be seal welded on installation and that the seal be cut and rewelded each time the joint is opened. In the larger valves, a gasketed flanged bonnet-to-body joint backed with a canopy ring seal is generally specified. This design is illustrated in Fig. 4.39, and it provides a high assurance of leak-tightness. The canopy ring is not seal welded until after the

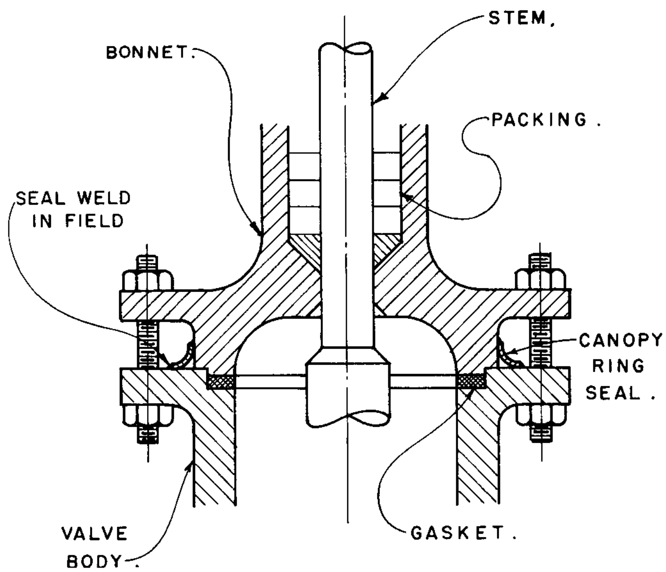


Fig. 4.39. Seal-Welded Backup for Bonnet-to-Body Closure in Type I Valves.

valve is installed and the system hydro-tested. The gasket provides a suitable seal until the system is ready for operation. The seal is achieved by first tightening the bolts to bring the flanges up metal-to-metal. Individual bolts are then removed one at a time, the canopy ring is seal welded at that point, and the bolts are replaced and re-tightened. The seal weld may be cut and rewelded several times. The canopy ring seal design has the advantage that the canopy ring may be cut off and replaced in the field should this become necessary.

Type II valves are not seal welded until system tests or operating experience confirm that the joint is leaking. A gasketed flanged bonnet-to-body joint frequently used for Type II valves is illustrated in Fig. 4.40. As with the canopy ring seal, the flanges must be drawn metal-to-metal before the joint is seal welded. This design is limited to the larger joints where the removal of one bolt does not compromise the metal-to-metal contact between flange faces at that joint.

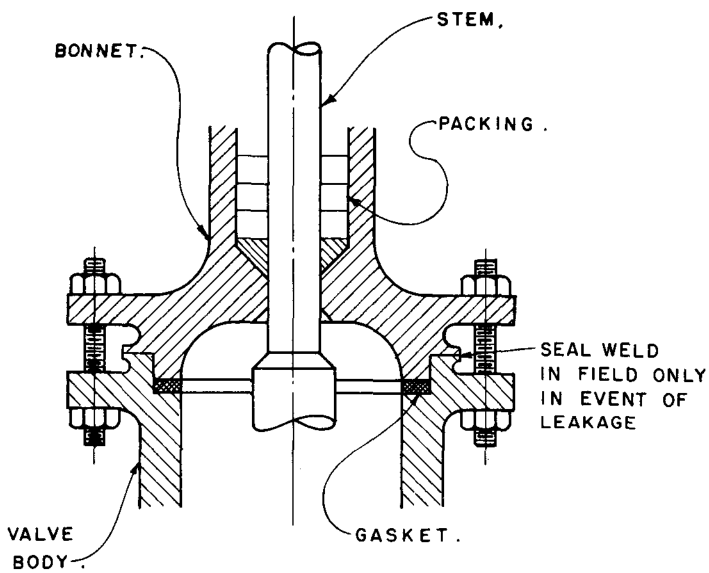


Fig. 4.40. Seal-Weld Backup for Bonnet-to-Body Closure in Type II Valves.

In the smaller bonnet-to-body closures, a seal-welded screwed connection is sometimes used. This design is illustrated in Fig. 4.41. By its very nature, the threaded type of bonnet closure cannot be relied upon to be completely torqued to compress a gasket properly. Thread

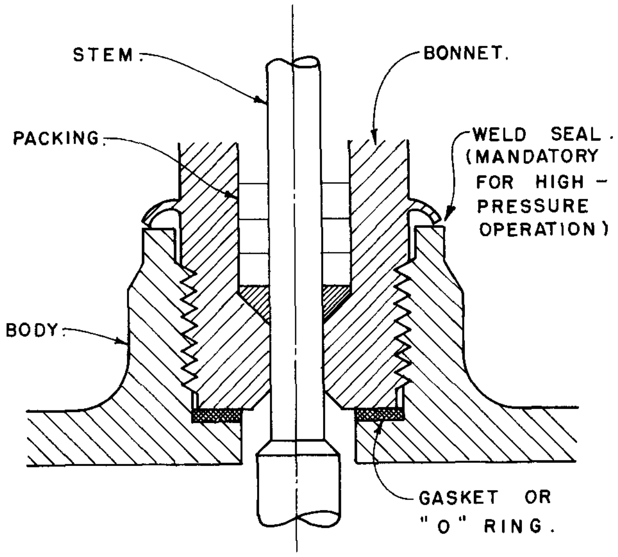


Fig. 4.41. Lip Seal Joint.

friction is high and unpredictable. The rotation of the bonnet when being screwed can result in abrasion of the gasket, and this can result in leak paths upon tightening. However, because of its simplicity, this type of design is used where the leak-tightness of the joint must be guaranteed by a weld seal; that is, where a Type I valve is specified. This type of design also reduces the size and weight of the valve.

Since the burden of leak-tightness does not rest on the closure design, the threads need not insure an interference fit or be required in any other way to minimize leakage flow. The threads may be coarse and relatively loose fitting. Nevertheless, a potential for thread galling does exist. For instance, it is recognized that over a period of time, cold welding of austenitic stainless steel pieces in contact can occur. This should be accounted for in the design by special hard-facing techniques or by the use of metals of dissimilar hardness.

Some alternates in screwed design do exist, and one is illustrated in Fig. 4.42. This design requires two annular seal welds. A more elaborate geometry such as this can usually be justified in taking high-pressure stresses off of the closure threads, thereby reducing or eliminating the possibility of thread galling.

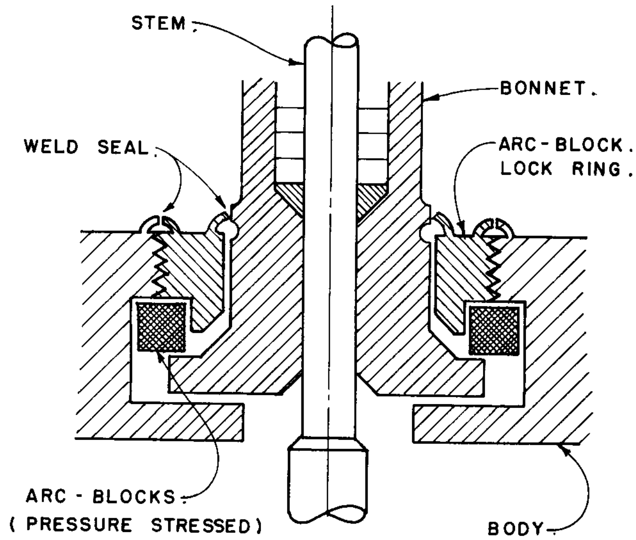


Fig. 4.42. Non-Stressed Thread in Screwed Bonnet Design.

(d) Pressure-Seal Bonnet Joint. The pressure-seal type of bonnet-to-body joint is illustrated in Fig. 4.43. Pressure-seal bonnet joints are preferred over gasketed flanged joints in large, high-pressure steam valves that do not require seal welding; that is, Type III valves. Study

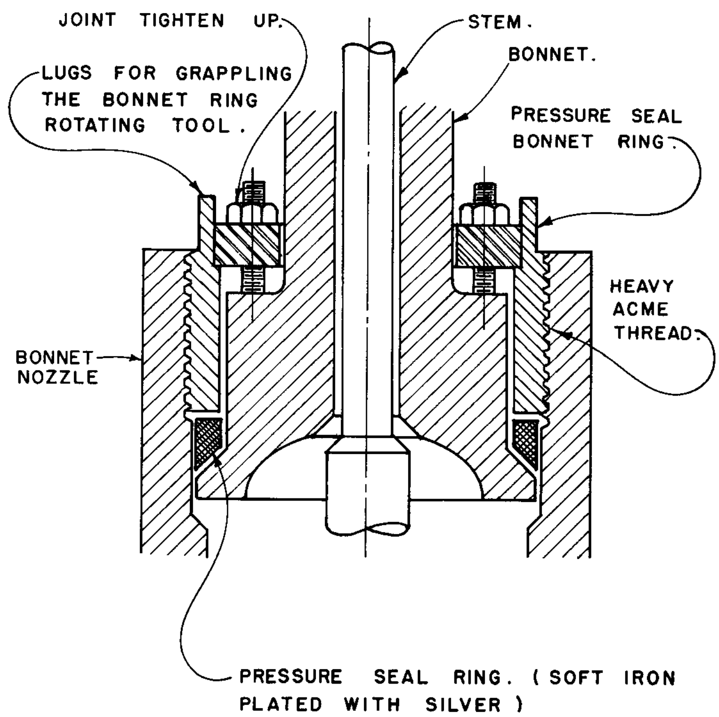


Fig. 4.43. Pressure-Seal Bonnet-to-Body Joint.

of Fig. 4.43 will show that in contrast to the gasketed flanged bonnet closures, the leak-tightness of pressure-seal bonnet closures is aided by increased system pressure. This is achieved by making the seal a reverse one with bonnet pressure increasing the sealing force. On the other hand, pressure-seal joints do not work well in small, low-pressure valves. This type of joint is not used in valves designed for the lower pressure ratings of 150, 300, or 400 lb. Pressure-seal bonnet joints may be used in valves designed for 600 and 900-lb ratings that are 2.5 in. NPS and larger, and this type of joint may be used in valves designed for pressure ratings of 1500 lb that are as small as 1 in. NPS.

Operation of the pressure-seal type of joint is straightforward. A soft metal seal ring with a corrosion resistant coating or plating is the basic sealing device. The ring is normally made of a soft iron and is plated with nickel or silver, the latter being the better. The closure is opened or closed by unscrewing the capture ring that engages the bonnet. Upon rotation, this ring draws the bonnet tight up against the seal ring, making the seal. The sealing pressure is reinforced upon pressurization of the system. Procedures for operating this closure usually specify that when the system is put back in operation after the closure has been opened, the seal ring compression ring must be periodically retorqued so the system can reach stress equilibrium.

Further precautions should be taken when designing pressure-seal bonnet joints. The thread surfaces should be properly designed to preclude corrosion and galling. These large threads should be of an Acme design to withstand the high axial force imposed on them. The sealing surfaces in contact with the pressure seal ring require a fine surface finish and are often hard-faced to improve their wear characteristics, particularly on the inside of the bonnet nozzle.

Pressure-seal bonnet joints are much more compact that gasketed flanged joints and result in a lighter valve. Although sharp hammer blows are sometimes necessary to unlock the seal, the pressure-seal bonnet joint is considered to be easier to assemble and disassemble than the gasketed flanged joint with its many studs and nuts. Mention should be made that they may require more access for assembly and disassembly than corresponding gasketed flanged joints. The most outstanding characteristic of the pressure-seal bonnet-to-body joints is their excellent record of leak tightness. Some of these pressure-seal types of closures may be designed for seal welding as required for Types I and II valves. However, the design is complex and requires at least two rings of seal welding to complete the closure. Pressure-seal bonnet joints therefore are seldom used if seal welding is a requirement.

4.5 Handwheels, Operators, and Position Indicators

Handwheels are the most common mechanism for operating a valve. They are simple, rugged, and need no other power source than a plant operator. However, the operator must be at the valve each time it is to be operated, and the operating forces and power are limited to the physical capabilities of the man operating the handwheel.

Valve operators are used where it is impractical to manually operate the valve either for reasons of required force, operating time, numbers of valves requiring simultaneous operation, valve accessibility, or some combination of these four reasons. The operating force requirements are generally proportional to the size of the valve so that the valve size in itself can give rise to a requirement for power operation. However, handwheels are specified on motor-operated valves as a reserve means of operating the valve when power is unavailable.

The governing factors in establishing the required operating time are generally the system conditions and the need to inhibit the severity of a possible accident. This may require the rapid isolation of a rupture, activation of a safeguard system, or both simultaneously. A rupture in the primary system of a pressurized-water reactor and a break in a steam line in a boiling-water reactor both have the potential of uncovering the core because of loss of water in the reactor vessel. Loss of water means loss of coolant and possibly fuel element damage caused by overheating.. In turn, fuel element damage would probably result in the release of fission products. Rapid closure of isolation valves is necessary to minimize the loss of water following a break. An external source of water (an engineered safeguard system) must also be activated quickly to supply additional water for more cooling.

The reactor coolant and portions of its auxiliary systems are enclosed in a containment structure to guard against release of radioactivity to the atmosphere in the event of leakage or breakage. The valves in the containment structure are not accessible during normal operation and must therefore be operated remotely. Since the use of reach rods is not practical, remotely actuated power operators are required.

Whether common mechanical types or sophisticated electronic displays, position indicators have the simple function of providing the operator with knowledge of the position of the valve. Remote indication is relied upon almost exclusively in reactor plants because the process functions are virtually all controlled from an operating room, while the plant components are located over a wide area both inside and outside the containment.

4.5.1 Handwheels

Handwheels are sized so that a reasonable force applied tangentially to the rim of the handwheel will effect a tight closure of the disc against its seat and, conversely, will move the disc free of its seat and out of the flow stream when applied in the opposite direction. The amount of force that may be applied by the operator is dependent upon the diameter of the handwheel, and the valve manufacturers vary the required tangential force accordingly.

For example, handwheels with diameters of 8 in. or less are designed to require less than 100 lb of force to operate. Valves with 12-in.-diameter handwheels are designed to require less than 135 lb of force for operation. When the diameter of the handwheel is from 20 to 24 in., the maximum tangential force is limited to 150 lb, and this force is considered a maximum for all larger handwheels. In large, high-pressure valves, the stem torque requirements are not satisfied with a handwheel of a reasonable size and the handwheel must be coupled to the stem through a gear mechanism to gain the necessary mechanical advantage.

As described in the previous discussion of stems in Section 4.3, the torque required to seat and unseat the disc may be quite large and unpredictable. In spite of all precautions, valves often tend to stick in service and require a shock load to start stem rotation. For this reason, large valves with high pressure ratings are generally provided with impactor handwheels. These are sometimes called hammer-blow handwheels. The design of the impactor handwheel permits the handwheel rim to turn freely through a large angle, building up momentum until the heavy operating lugs engage. The resulting impact load on the stem is quite effective in opening and closing the valve.

4.5.2 Valve Operators

The predominant types of remote operators for large valves are electric motor operators and pneumatic diaphragm operators. Hydraulic piston operators are sometimes used for large valves requiring rapid operation. The major characteristics of these three valve operators are given in Table 4.1.

Valve operators are designed to move the stem under all non-accident conditions of opposing force. The force requirements to move the stem vary greatly and are dependent in large measure on system pressure, fluid motion, and the type and tightness of the packing gland. For example, the stem force required to close a gate valve under full flow conditions where the fluid is forcing the gate against the guides and increasing frictional resistance to stem motion may be quite high and somewhat indeterminate because of the uncertainties in determining frictional resistance where an unknown state of wear exists on the friction surfaces.

On the other hand, stem force must be controlled so that any structural limitations in the valve are not exceeded. Hydraulic and pneumatic operators can be controlled by the operating pressure of the fluid and the area of the operating piston or diaphragm. Motor operators, which act through gear drives on the stem, are controlled by limit switches that are actuated by valve position and/or developed torque. These devices must have high reliability to prevent damage to the valve or operator.

Table 4.1. Major Characteristics of Electric Motor, Pneumatic Diaphragm, and Hydraulic Piston Valve Operators

Electric Motor Operator	Pneumatic Diaphragm Operator	Hydraulic Piston Operator
Slower operation, generally greater than 10 seconds.	Quick operation, generally less than a second.	Quick operation, generally less than 5 seconds.
Suitable for large stem movement.	Limited to short stem movement.	Suitable for large stem movement.
Provides high permanent seating force.	Seating force is limited and difficult to maintain; often combined with a spring to provide permanent seating.	High seating force available but difficult to maintain; often combined with a spring to insure permanent seating.
Size of operator reasonable, particularly for valves requiring considerable energy to operate but a slower closing time is suitable.	Size requirements limit it to the smaller of the large ($>$ 4-inch NPS) nuclear valves.	Small operator size a major advantage for large, high pressure valves.
Power readily available from plant electrical system.	Air readily available from plant air system.	Requires special hydraulic system to provide valve motive energy.

When the closing and opening speeds are high, the valve design must have adequate provision for shock loading. It may be important to provide a means of reducing the high transient forces through the use of energy absorbing devices. Dashpots or springs may be used to reduce the velocity just prior to disc contact with the seat.

(a) Motor Operators. Motor operators are suited to those applications where the energy requirements to operate the valve are large and a slower operating time is acceptable to the fluid system. Most motor operators in nuclear plant service are electric-motor driven although hydraulic and air motors are easily adapted to the operator mechanism. The availability of suitable electric power in the plant is a major factor in this selection. The normal attachment of an electric motor operator to a valve is illustrated in Fig. 4.44.

Electric motor operators consist of a reversing motor, reduction gearing, and the various switches needed to start and stop the motor.

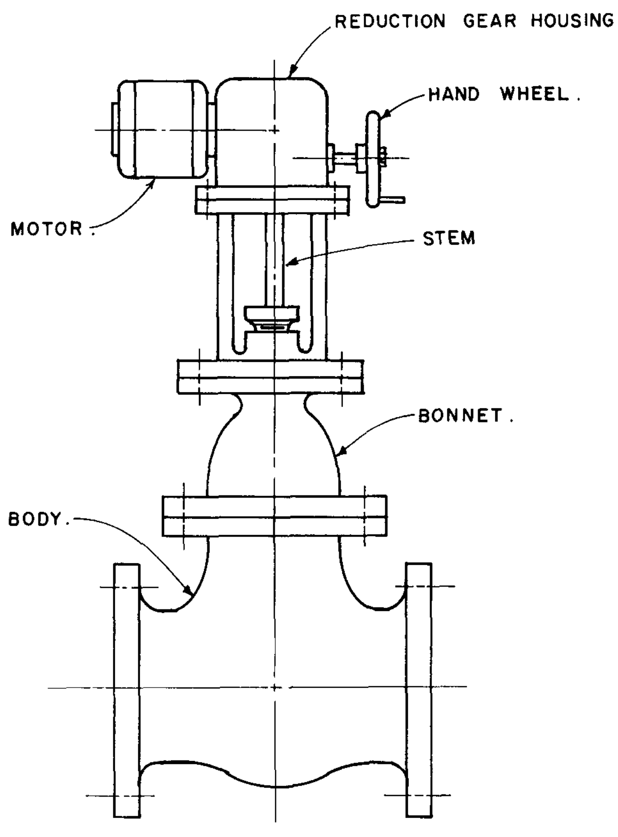


Fig. 4.44. Electric Motor Operated Valve.

The limit switches are capable of limiting travel in both the open and close directions, and most operators are also equipped with torque switches capable of limiting the applied stem thrust in both opening or closing. This affords protection for the valve and operator should any foreign matter obstruct movement of the disc. The motor is a special high-starting-torque motor designed for intermittent duty. The enclosure for the motor and switches must be suitable for normal service environments. Additional enclosure requirements may be needed for those valves located within the containment building that must be operated following a loss-of-coolant accident. The lubricant for the motor should be sealed in and adequate for the life of the motor. The radiation field in which the valve operator must function should be known to assure that lubricants and electrical insualtion will not suffer radiation-induced deterioration. Additionally, the operator installation must be such that the valve packing can be replaced without disassembly or removal.

A handwheel should be provided on the valve operator. The motor should be disconnected from the drive when the valve is operated manually, and the handwheel should be disconnected from the drive train when the valve is motor operated. The drive train should have an impactor or "lost-motion" device that will allow the motor and gearing to pick up full speed and momentum before torque is applied to the valve stem nut. Such a device will increase the effectiveness of the unit in unseating sticky valves, and it should be effective for both manual and electrical operation. As with manually operated valves, motor operators provide and maintain a high seating force because of the high frictional drag resulting from the stem screw and reduction mechanisms. They are particularly suited to applications requiring a single open and close operation of the valve and provide reliable service with little maintenance.

(b) Pneumatic Diaphragm Operators. Pneumatic diaphragm operators are widely used in nuclear power plants for remote operation of valves. A pneumatic diaphragm operator mounted on a valve is illustrated in Fig. 4.45. These operators provide high operative speeds in a single and reliable operator. The air supply needed to actuate the diaphragm is readily available from the plant air system, but care should be taken to

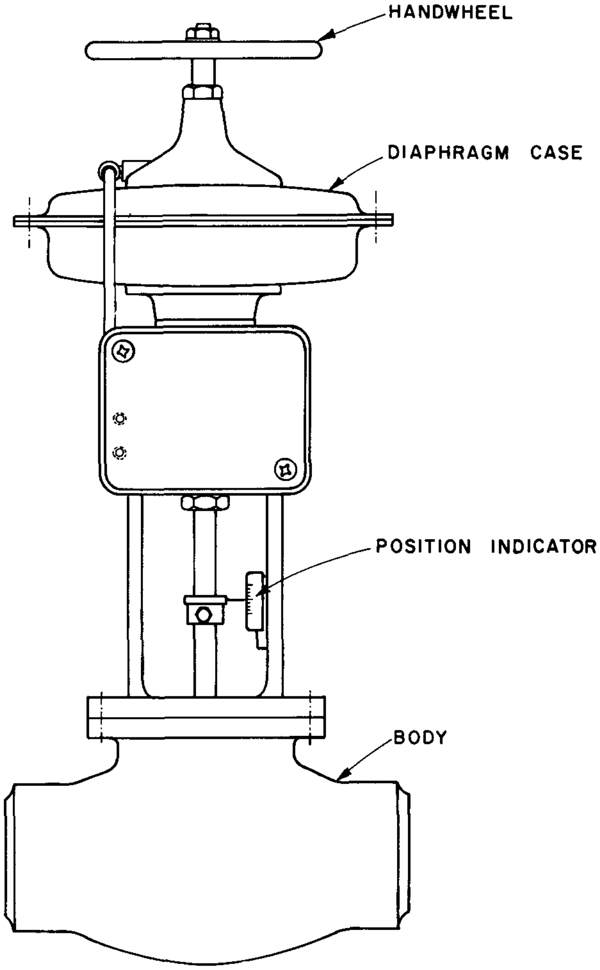


Fig. 4.45. Pneumatically Operated Valve.

operate with air which is free of moisture and oil. Otherwise, the maintenance requirements of the operator become excessive.

Pneumatic diaphragm operators are often used with throttling control valves, and they provide a responsive and accurate operator for this important function. When used with stop valves, these operators frequently have a spring to assist or provide the disc seating force and the diaphragm provides the energy to open the valve. Rigged in this manner, the valve disc will remain closed if the operator suffers a loss of air pressure.

Pneumatic diaphragm operators have two major limitations. They are primarily suited for valves which have a relatively short stem operational stroke, and the actuating force exerted by the diaphragm is relatively small for the size of the operator. As a result, the use of diaphragm operators is limited to the smaller stop valves in the higher pressure rating ranges. In the lower pressure rating range, they may be used on stop valves as large as 12 in. NPS. The pneumatic diaphragm operators are considerably more powerful than electric solenoid operators, which are often used to control the admission or release of air for the pneumatic diaphragm operators.

(c) Hydraulic Piston Operators. Hydraulic piston operators, such as the one illustrated in Fig. 4.46, are used on large, high-pressure valves requiring quick operation. They are not used as universally as the electric motor or pneumatic diaphragm operators. The hydraulic piston operators are small for their exerted force and simple in construction. A piston attached to the stem rides in a cylinder attached to the valve. The piston-cylinder arrangement is adaptable to a long stem travel

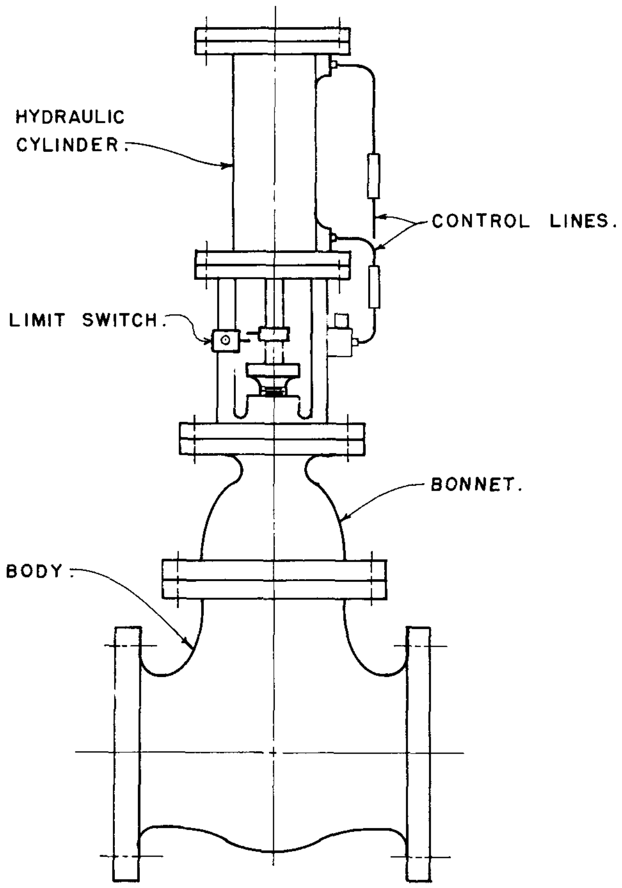
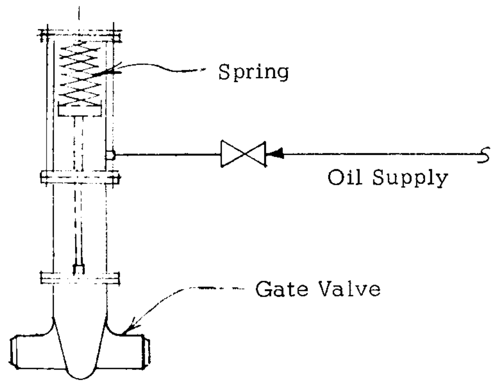
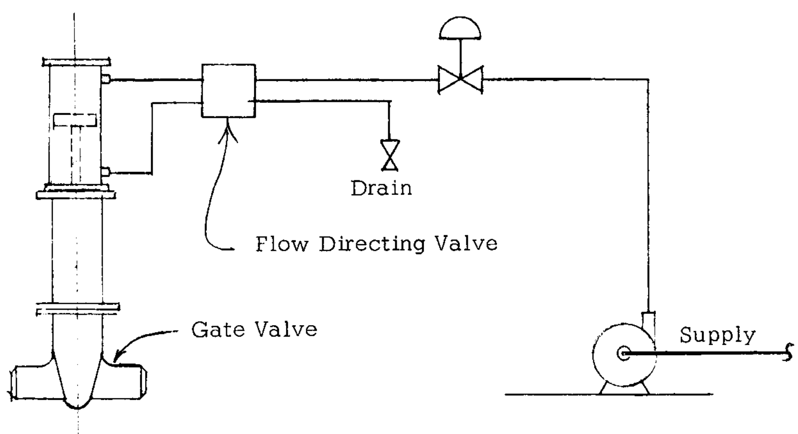


Fig. 4.46. Hydraulic Piston Operator Mounted on Valve.

requirement. The operating time of the valve is limited only by the capabilities of the hydraulic fluid system. Schematic diagrams of several hydraulic systems that may be used to actuate the valve operator are illustrated in Figs. 4.47 and 4.48.



Hydraulic Oil Fail Closed



System Fluid Fail As Is
Piston Latch Open and Closed

Fig. 4.47. Schematic Diagrams of Hydraulic Systems Used to Actuate Piston Valve Operator.

The feasibility of oil is limited, however, because of its potential as a contaminant and also because of the fire hazard. High-pressure reactor coolant can also be used to actuate the operator, and this has the advantage of permitting a free leakage path between the high-pressure valve and its operator. When either hydraulic oil or high-pressure

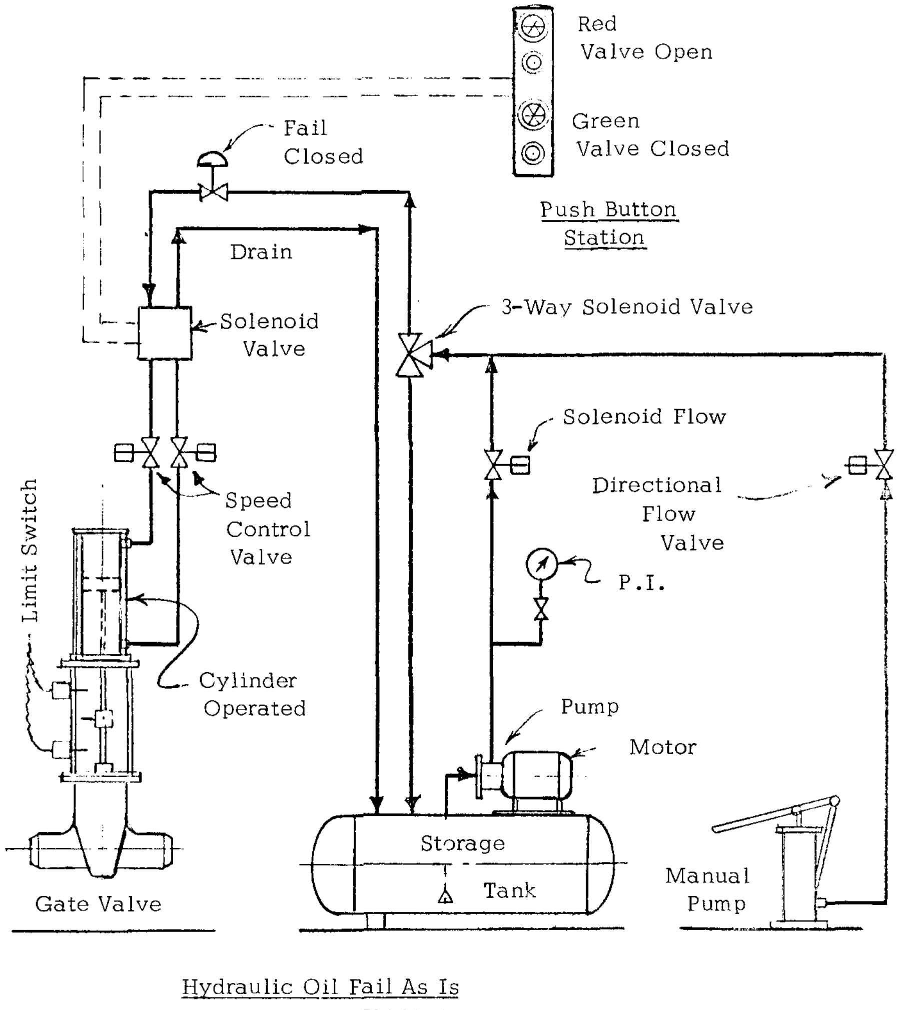


Fig. 4.48. Schematic Diagram of Hydraulic System Used to Actuate Piston Valve Operator.

reactor coolant is used to actuate the operator, a large, high-pressure system is required to provide the motive force.

4.5.3 Position Indicators

The function of a position indicator is to give the plant operator a readily intelligible display of the position of a valve. In the event of accident or suspected accident conditions, it is vital that the plant operator be able to tell immediately the position of significant valves to assess the effects of their position on plant conditions.

Position indicators run the gamut from simple mechanical devices observed locally at the valve to elaborate lighted panels that may display "open" lights, "shut" lights, or both. In many cases, the process system is outlined symbolically on the control panel with a lighted "O" indicating an open valve or a lighted bar indicating a shut valve. The position indication displays usually get their signals from the same or similar micro-switches used as limit switches to control valve operation. These same signals may also be used in interlocks or control circuits. An example would be a case where a reactor coolant loop had been isolated from the remainder of the plant and cooled down. The limit switch on the isolation valve would control a relay in the control circuit for the reactor coolant pump that would not allow the pump to be started unless the reactor inlet valve was closed. This would prevent a reactor power excursion caused by the introduction of "cold" water.

Thus, valve position indication and its related functions are of great importance to the safe and efficient operation of a reactor plant. It cannot be overemphasized that the position indication system must be reliable and the readouts must be clearly displayed.

4.6 Locking Devices

Locking devices are used to prevent fasteners or parts thereof from loosening. The fasteners are in turn holding together structural parts of a component. In the case of internal fasteners in primary coolant systems, the ability to insure tightness and to retain fasteners in the event of their failure are of paramount importance. Any loose parts in the flow stream could be carried back to the reactor core and block core flow passages. This could result in fuel element failure caused by overheating from blockage of coolant flow.

The locking device provides a mechanical obstruction to motion of the fastener in the loosening direction. The device must retain its locking ability under all anticipated environmental conditions and these may include vibration, shock, corrosion, and erosion induced by water flow and turbulence; thermal transients, gradients, or shock; vibration from water flow, primary coolant pump rotation, or operation of other machinery; and mechanical shock.

The material of the locking device is generally the same as that of the fastener or one that is compatible to insure its corrosion resistance in the environment. A similar coefficient of expansion is also desirable to prevent loosening caused by differential expansion. The locking device must also accept the deformation necessary to achieve locking action without incurring any damage or breakage beyond that deformation. Locking devices are generally expendable, and they should be simple enough to permit their installation by unskilled workmen.

Because of the varying consequences of fastener failure, locking devices are categorized as either Class A or B by their service requirements. A Class-A locking device is one that will capture and retain the fastener fragments in place in the event of fastener failure. The fastener must also be designed so that in the event of locking device failure, the fragments of the locking device will be held in place by a structural member. Locking devices exposed to primary coolant shall be Class A, and the use of lock wires is prohibited in Class A locking devices. The use of a Class A device is optional in secondary system fluid environments.

A Class-B locking device is one that will not capture fastener fragments in the event of fastener failure. This class of device is acceptable for use outside the primary coolant fluid environment and for use in secondary system environments where the passage of fragments of the fastener or device to other parts of the system is implausible and will not damage or interfere with the functioning of any part of the system.

4.6.1 Class-A Locking Devices

The Class-A locking device must be capable of retaining the fastener and fastener fragments in the event of fastener failure. In addition, the locking device must be retained by a structural member in the event of its own failure. Despite the unhandiness involved in disassembly, welding is a favored form of locking for internal reactor plant application because of the Class-A requirements. A welded self-capturing locking device is illustrated in Fig. 4.49. The preference for welding is attributable to its degree of positiveness in retaining the fastener in the event of fastener failure and to favorable cost considerations.

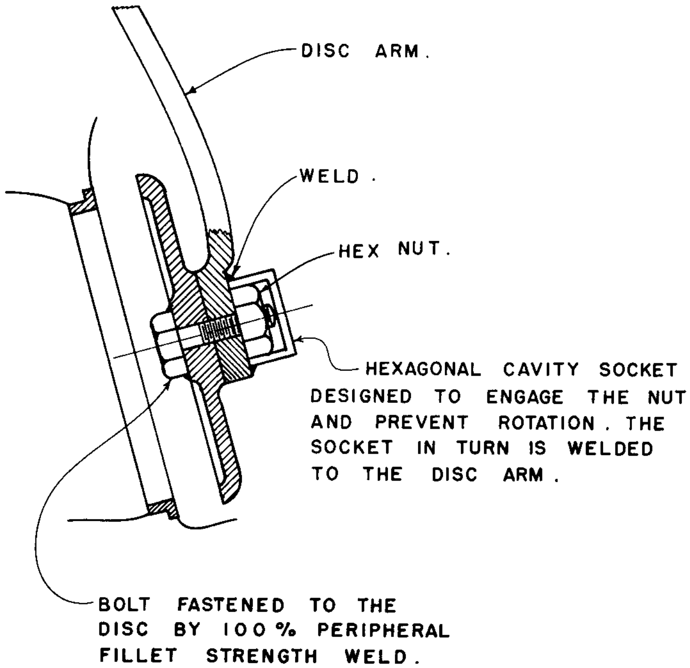


Fig. 4.49. Welded Self-Capturing Locking Device for Use in Primary Coolant Systems.

When welded locking devices are used, consideration must be given to the weldability of the materials, the effect of welding on the corrosion resistance of the assembly, the cleanliness of adjacent members, and the initiation of cracks. The size and location of the weld should be specified on the drawing, and where accessibility for welding may be questionable, it is desirable to qualify the welding involved by requiring satisfactory demonstration of welding accessibility through the use of a mock-up of the weld joint. Fillet welds should be a minimum of 1/8 in. and at least 1 in. long. There should be a minimum of two welds equally spaced. All welds should be dye-penetrant inspected to assure freedom from cracks.

A Class-A lock washer that does not require welding is illustrated in Fig. 4.50. The lock washer is forced into the enlarged section of the recessed hole in the structure for retention by the structure, and it can

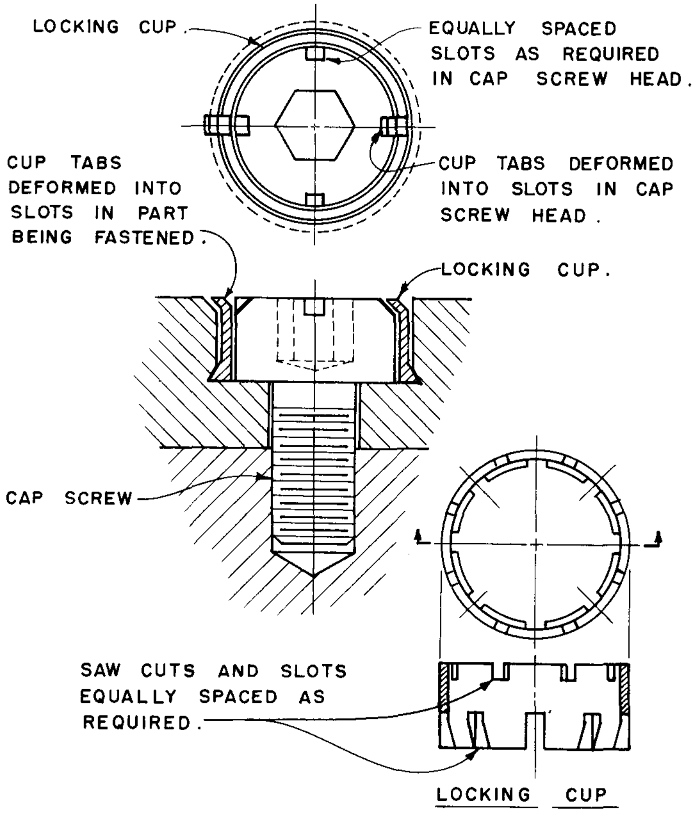


Fig. 4.50. Lock Washer for Use in Primary Coolant Systems That Does Not Require Welding.

be crimped over the upper portion of the bolt head to lock and retain this fastener. Other non-welded Class-A locking devices are illustrated in Figs. 4.51 through 4.54. The use of a locking cup with protruding and recessed head cap screws is illustrated in Figs. 4.51 and 4.52, and the use of a pin with recessed and protruding head cap screws is illustrated in Figs. 4.53 and 4.54. When pins are used as part of a locking device, the design should be such that a push fit is obtained. Where practicable, headed or tapered pins should be used. All pins should be mechanically secured.

Bolts that are tensioned with heating elements or tensioning machines are not required to have locking or retention devices. In cases where structural parts are pinned together, a device which will retain the fastener but not lock it may be used. Such a device may be a cap or disc welded over the pin hole or a disc extruded into the recesses over the pin in a recessed hole.

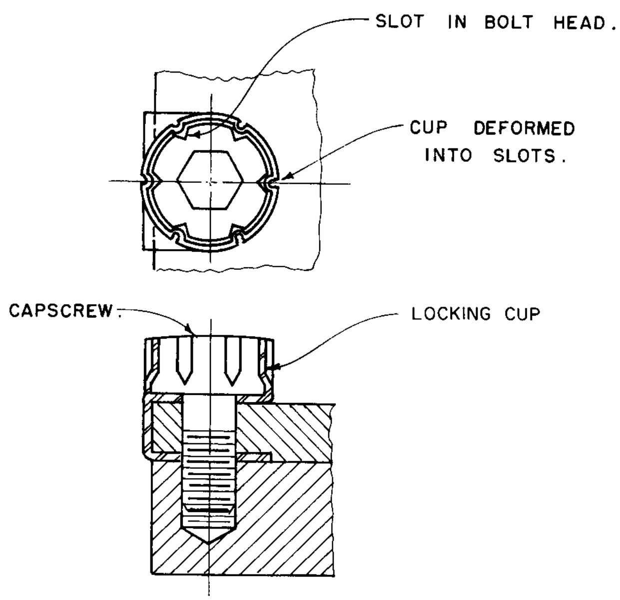


Fig. 4.51. Class-A Locking Cup for Protruding Head Cap Screw.

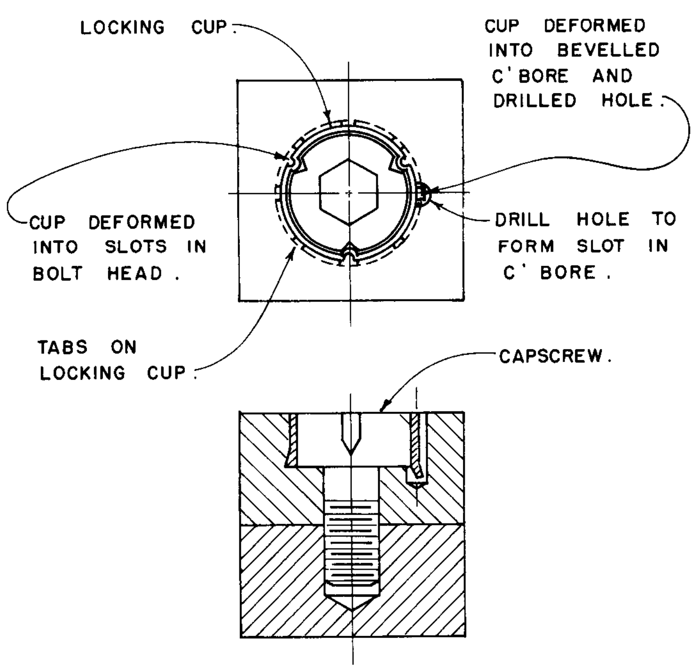


Fig. 4.52. Class-A Locking Cup for Recessed Head Cap Screw.

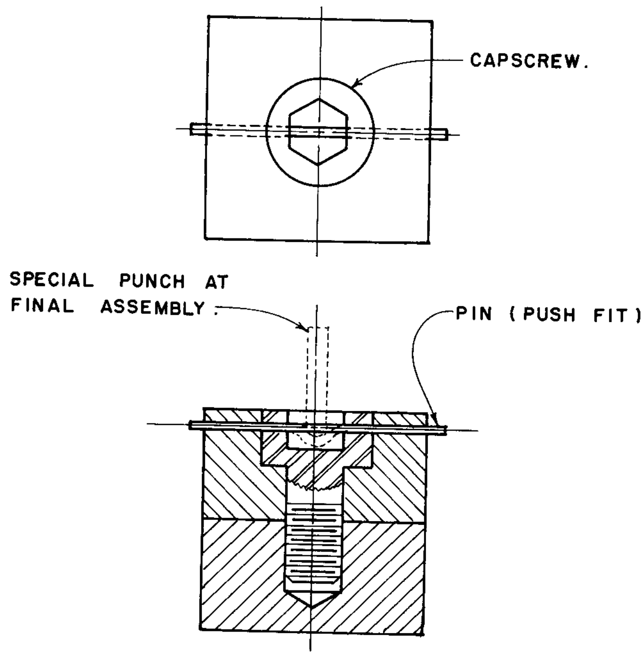


Fig. 4.53. Class-A Push-Fit Pin for Recessed Head Cap Screw.

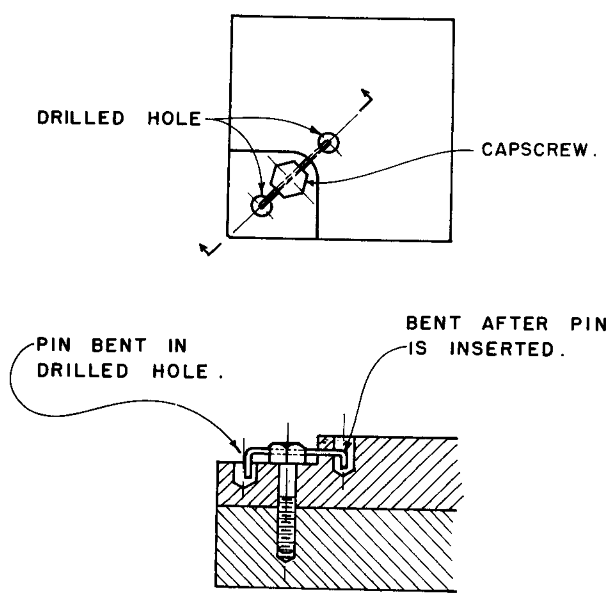


Fig. 4.54. Class-A Bent Pin for Protruding Head Cap Screw.

4.6.2 Class-B Locking Devices

Those locking devices not requiring fastener retention in the event of failure are identified as Class B. This class includes the more common devices such as the lockwire illustrated in Fig. 4.55 and the snap ring with keeper illustrated in Fig. 4.56. However, the use of lockwire

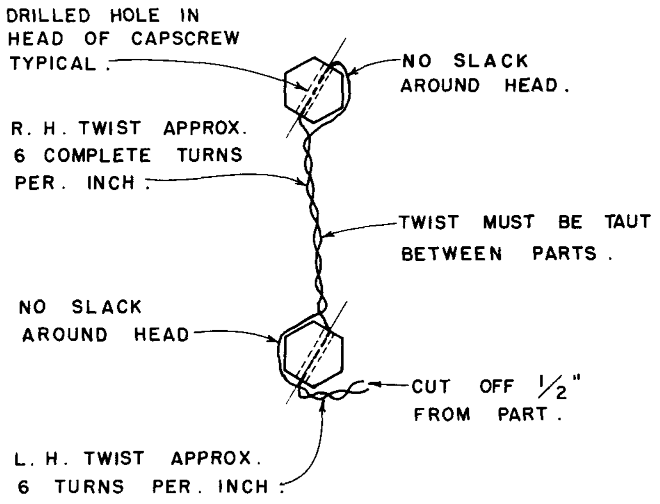


Fig. 4.55. Class-B Lockwire Locking Device.

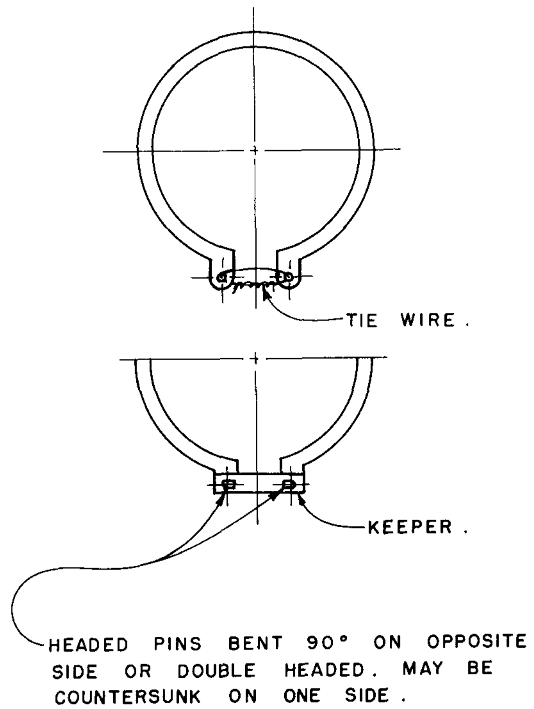


Fig. 4.56. Class-B Snap Ring With Keeper.

in high-velocity water is undesirable because of possible erosion. When used, lockwire should have a diameter of at least 0.050 in. The use of a pin with a recessed head cap screw as a Class-B locking device is illustrated in Fig. 4.57. If the head of the screw was held with a locking cap or by an adjacent member, it would be retained in the event of fastener failure and this arrangement would meet the requirements for a Class-A device.

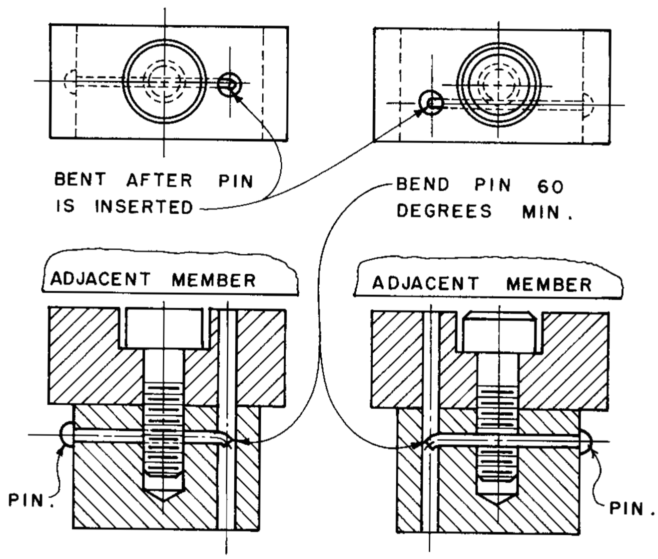


Fig. 4.57. Pin With Recessed Head Cap Screw Used As Class-B Locking Device.

Other Class-B locking devices are illustrated in Figs. 4.58 and 4.59. Locking plate and locking tab washer designs are illustrated in Fig. 4.58, and the use of an interference-fit pin with a stud bolt is illustrated in Fig. 4.59.

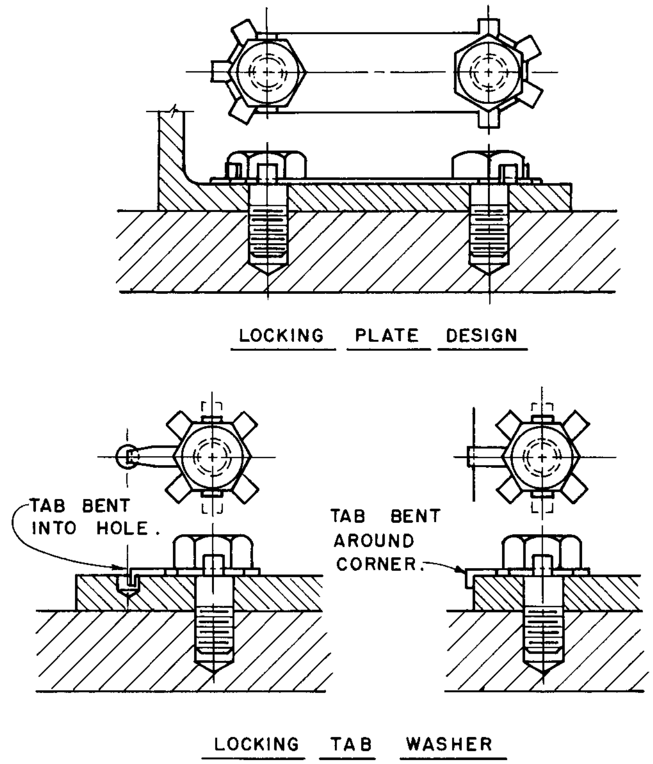


Fig. 4.58. Class-B Locking Plate and Locking Tab Washer Designs.

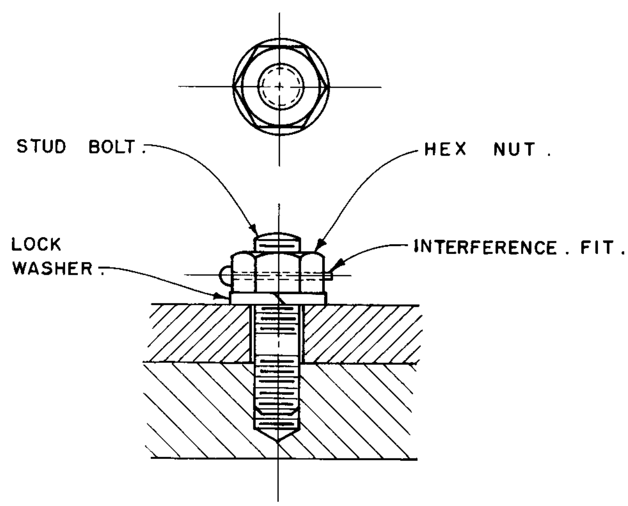


Fig. 4.59. Pin Used With Stud Bolt as Class-B Locking Device.

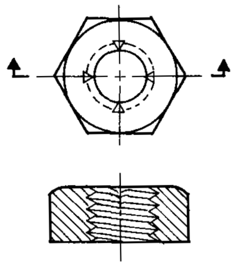
4.6.3 Standard Fastener Locking Devices

Standard fastener locking devices include fasteners themselves that incorporate locking features or that may be used in conjunction with a lock washer to achieve locking. Several types of standard fastener locking devices do not meet Class A or B standards because they do not have a sufficient degree of locking reliability and/or they do not have the fastener retention capability needed to meet Class-A standards. Among these are friction-type devices such as set screws, jam nuts, self-locking nuts, etc.

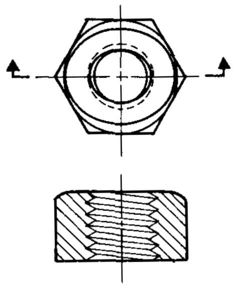
The ideal locking device should provide a positive stop against rotation, and this combined with sufficient elasticity in the fastener or adjacent structure would prevent relaxation of the fastening forces should any minute rotation occur. No matter how tightly secured, unlocked fasteners can eventually loosen if the load on the thread fluctuates. When the load increases, the nut dilates and the bolt contracts in the radial direction. Therefore, the threads slide back and forth on each other as the load fluctuates. This sliding is not purely radial but has a slight circumferential component which follows the unfastening direction of the thread helix. This phenomenon precludes the use of friction types of locking devices for critical applications involving long life expectancy.

Locking fasteners are internally or externally threaded mechanical fasteners which resist rotation by gripping the mating thread or connecting material. A locking fastener often has the same general dimensions and mechanical requirements as a standard fastener but with the addition of a locking feature. The most common types are locknuts and locking screws and bolts. Two general classes of locking fasteners are the prevailing torque and the free spinning.

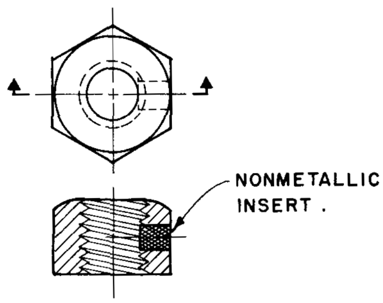
(a) Prevailing-Torque Locknuts. Prevailing-torque locknuts spin freely for a few turns and then must be wrenched to their final position. The maximum holding and locking power is reached as soon as the threads and the locking feature are engaged. Locking action is maintained until the nut is removed. Typical styles of prevailing-torque locknuts are illustrated in Fig. 4.60.



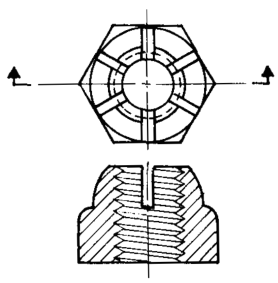
DEFORMED THREAD TYPE. DEPRESSIONS IN THE FACE OF THE NUT DISTORT A FEW OF THE THREADS.



OUT - OF - ROUND THREADED COLLAR ABOVE REGULAR LOAD - BEARING THREADS GRIPS THE BOLT.



NONMETALLIC INSERT GRIPS THE THREADS AND CAUSES A WEDGING ACTION BETWEEN BOLT AND NUT.



SLOTTED SECTION OF THIS PREVAILING-TORQUE NUT FORMS BEAMS WHICH ARE DEFLECTED INWARD AND GRIP THE BOLT.

Fig. 4.60. Typical Styles of Prevailing-Torque Locknuts.

Prevailing-torque locknuts are classified by their basic design principles whereby

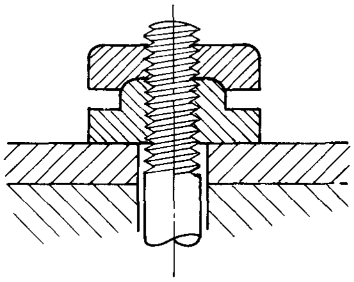
1. thread deflection causes friction to develop when the threads are mated and the nut resists loosening,
2. out-of-round top portion of the tapped nut grips the bolt threads and resists rotation,
3. slotted section of locknut is pressed inward to provide a spring frictional grip on the bolt,
4. non-metallic or soft-metal inserts are plastically deformed by the bolt threads to produce a frictional interference fit, or
5. spring wire or pin engages the bolt threads to produce a wedging or ratchet locking action.

Prevailing-torque locknuts can be either directional or nondirectional. The directional types can be started from only one end, and the nondirectional types can be started from either end. All remain in position until removed whether they are seated or not. To prevent damage to the locking element, they should not be used in applications where long nut-on-bolt travel is required. Blunt-end bolts should be avoided because the starting thread can damage the locking element of the nut.

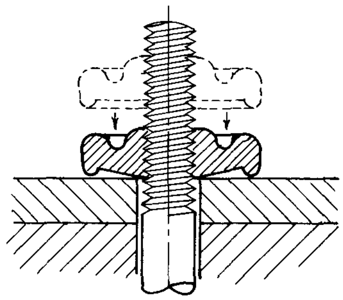
(b) Free-Spinning Locknuts. Free-spinning locknuts are free to spin on the bolt until seated. Additional tightening locks the nuts. Free-spinning locknuts are also classified by their design principles. Four basic types of free-spinning locknuts are illustrated in Fig. 4.61, and their design principles are as follows.

1. Two mating parts wedge together when tightened against each other or against a component of work, and an inward pressure on the bolt is developed. A recessed bottom face and a slotted grooved upper portion characterize this type. As these nuts are drawn up tight against the work, a spring action causes the threads of the upper portion to bind on the bolt. When pressure is relieved, most nuts resume their original shape and can be easily removed and reused. However, some take a permanent set upon seating and become, in effect, prevailing-torque nuts.

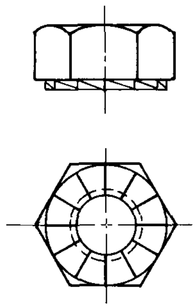
2. Metallic and non-metallic inserts in the bearing face collapse against the work when the nut is tightened to provide locking action.



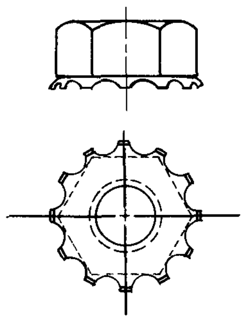
WHEN THE UPPER HALF OF THIS NUT IS TIGHTENED, IT PRESSES THE COLLAR OF THE LOWER HALF AGAINST THE BOLT.



TWO POSITIONS OF A DIAPHRAGM-TYPE LOCKNUT, BEFORE AND AFTER SEATING. BENDING ACTION CAUSES UPPER THREADS TO GRIP THE BOLT.



DEFORMED BEARING SURFACE. TEETH ON THE BEARING SURFACE "BITE" INTO WORK TO PROVIDE A RATCHET LOCKING ACTION.



NUT WITH A CAPTIVE TOOTHED WASHER. WHEN TIGHTENED, THE CAPTIVE WASHER PROVIDES THE LOCKING MEANS WITH SPRING ACTION BETWEEN THE NUT AND WORKING SURFACE.

Fig. 4.61. Four Basic Types of Free-Spinning Locknuts.

Thermoplastic inserts flow around the bolt threads to form a tight lock and an effective seal.

3. A deformed bearing surface provides a ratchet effect. Teeth on the bearing surface are designed to "ride" during tightening and are shaped to dig in and resist backoff.

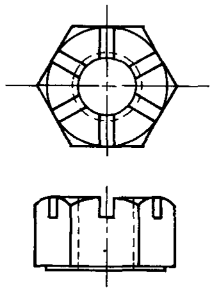
4. A captive locking member provides spring action between the nut and work surface. The locking member may be a captive toothed washer, a helical-spring washer, or a conical washer.

Free-spinning locknuts are often specified when long nut-on-bolt travel is unavoidable. Since most free-spinning locknuts depend upon clamping force for their locking action, they are usually not recommended for joints that might relax through plastic deformation or for fastening materials that might crack or crumble when subjected to preloading.

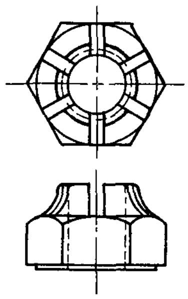
(c) Other Types of Locknuts. The other types of locknuts considered here are jam nuts, slotted nuts, castle nuts, and single-thread engaging locknuts. Jam nuts are thin nuts used under full-size nuts to develop locking action. The large nut has enough strength to elastically deform the lead threads of the bolt and jam nut. Thus, a considerable resistance against loosening is built up. The jam nut is considered ideal for assemblies where long nut-on-bolt travel under load is necessary to bring mating parts into position. However, the use of jam nuts is decreasing. A one-piece prevailing-torque locknut is usually used in preference to the jam nut at a savings in assembled cost.

Slotted and castle nuts have slots to receive a cotter pin which is passed through a drilled hole in the bolt, as illustrated in Fig. 4.62. The pin serves as the locking member. These nuts are essentially free-spinning nuts with the locking feature added after the preload condition is developed. Castle nuts differ from slotted nuts in that they have a circular crown of a reduced diameter. From an assembled-cost viewpoint, these nuts are expensive because of the extra operations involved in their assembly. A one-piece prevailing-torque locknut usually provides a better solution.

Single-thread engaging locknuts are spring-steel fasteners that may be speedily applied. Locking action is provided by the grip of the thread-engaging prongs and the reaction of the arched base. Their use



SLOTTED NUT USES A COTTER PIN THROUGH A HOLE IN THE BOLT FOR LOCKING ACTION.



CASTLE NUT IS BASICALLY A SLOTTED NUT WITH A CROWN OF REDUCED DIAMETER.

Fig. 4.62. Slotted and Castle Nuts.

is limited to non-structural assemblies and usually to screw sizes below 1/4 in. in diameter. When compared with multiple-thread locknuts, single-thread engaging locknuts are less expensive and lighter. The spring steel provides a resilient assembly which absorbs a certain amount of motion. However, the tightening torques for these nuts must be lower than those for multiple-thread nuts.

(d) Locking Screws and Bolts. Externally threaded fasteners with some sort of locking device such as those illustrated in Fig. 4.63 are classified the same as locknuts and use some of the same principles to achieve thread locking. For example, the prevailing-torque types of locking screws use such features as distorted threads, interference fit, or an insert.

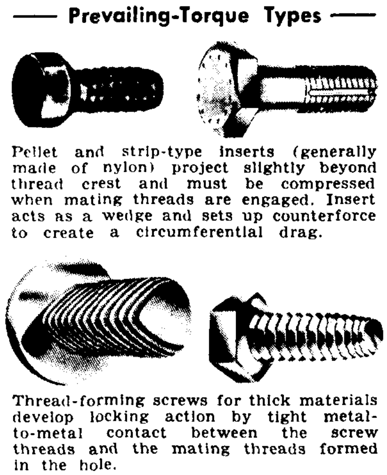


Fig. 4.63. Prevailing-Torque Types of Locknuts (from Ref. 32).

Free-spinning locking screws, such as those illustrated in Fig. 4.64, and bolts have a locking feature on the underside of the head so that locking is achieved when the screw or bolt is tightened.

(e) Lock Washers. Helical spring washers, such as those illustrated in Fig. 4.65, are made of slightly trapezoidal wire formed into a helix of one coil so that the free height is approximately twice the thickness of the washer section. They are usually made of hardened carbon steel but are also fabricated from bronze, stainless steel, K-Monel, and other resilient materials.

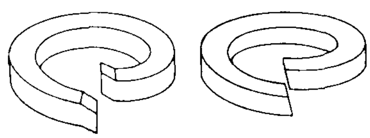
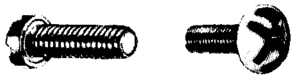


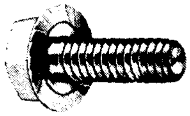
Fig. 4.65. Non-Link Positive and Plain Helical Spring Washers (from Ref. 32).

Helical spring washers are covered in the ANSI Standard B27.1-1958 as regular, extra duty, high collar, light, and heavy types for screw sizes from No. 2 to 3 in., inclusive. They are defined as (1) spring takeup devices to compensate for developed looseness and loss of tension between component parts of an assembly and as (2) hardened thrust bearings

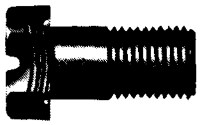
Free-Spinning Types



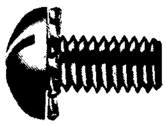
Reusable self-locking screw has cross-slotted head and recessed area adjacent to shank to form a spring element. Locking action results from spring tension in head when screw is tightened against workpiece to a predetermined torque.



Specially shaped, serrated tooth on underside of screw head resists rotation when fastener is tightened against mating surface. Teeth dig into material and resist counterrotation.

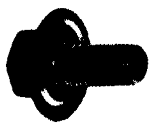


Hex bolt has depressed center and slotted head, with an undercut under the head designed to develop elastic stretch of the fastener. Bolt head acts like a set of controlled springs to absorb shock and vibration loads.



Preassembled washer and screw develop locking action upon tightening. Washer is tightened between the underside of the screw head and the work surface to provide a biting and spring action.

Standard screw or bolt coated with chemicals which are stable until mixed. Assembly into mating threads mixes epoxy adhesive and hardening agent. Epoxy cures to provide a locking bond.



Flanged-head screw has ratchet-like teeth and a circular groove on underside of flange. When tightened, teeth embed into the work surface, and the groove increases spring action (in much the same manner as a spring washer) to resist vibration and maintain clamping load.

Fig. 4.64. Free-Spinning Types of Locking Screws (from Ref. 32).

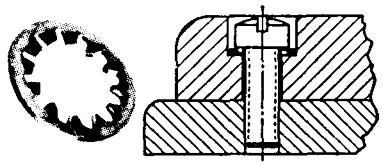
facilitate assembly and disassembly of bolted fastenings by decreasing the frictional resistance between the bolted surface and the bearing face of the bolt head or nut.

The plain helical spring washer is the most commonly used. The spring resistance of this washer comes into play when the tension in the bolt is reduced to a value equal to the force required to flatten the washer. At this point, the tension in the bolt is maintained by the expansion of the washer. In the flattened state, this washer is equivalent to a hardened flat washer with a small outside diameter. Because of this relatively small diameter, the washer is not recommended for use on soft materials or with oversize or elongated clearance holes. Special washers with wide cross sections for increased frictional area are available.

Various forms of tooth lock washers are illustrated in Fig. 4.66. A tooth lock washer has teeth that are twisted or bent out of the plane of the washer face so that sharp cutting edges are presented to both the workpiece and the bearing face of the screw head or nut. When compressed in position, most of the pressure is supported by the rim of the washer. However, any relaxing of the tension in the screw allows the teeth to bite into the workpiece and the face of the fastener, resisting relative motion. On hex-head screws or nuts where the teeth extend slightly beyond the hexagon, additional locking occurs on the edges. Each tooth acts as a spring of rather strong reaction because of its short length.

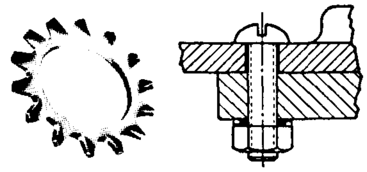
Tooth washers are used with screws and nuts not only to effectively add spring takeup for elongation of the screw but to increase the frictional resistance under the screw head or nut face. They bite into both the head of the screw and the work surface to provide an interference lock. Even at zero tension, the tooth lock washer will provide frictional resistance to loosening. In addition, tooth washers can be designed to make good electrical contact, span holes, seal, provide anti-shift features, reinforce sheet metal, and distribute loads. These special designs are possible because the locking teeth can be designed into many different shapes and sizes of metal stampings. Standard washers of internal, external, countersunk, and internal-external tooth design are listed in the ANSI Standard B27.1-1958.

TOOTH LOCK WASHERS



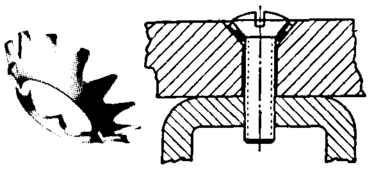
Internal

Used where projection of teeth beyond screw head or nut is objectionable. Presents a finished appearance, similar to plain washer. Standard sizes from No. 2 to 1¼ in. Up to 2⅜ in. available. Heavy washer made in ¼ in. to ⅞ in. sizes.



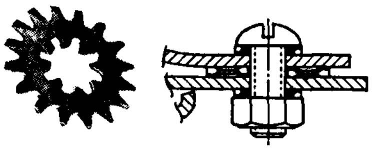
External

Most commonly used where security of assembly is critical. Teeth bite into standard screw heads and nuts to lock. Standard sizes from No. 3 to 1 in. Up to 2¼ in. available.



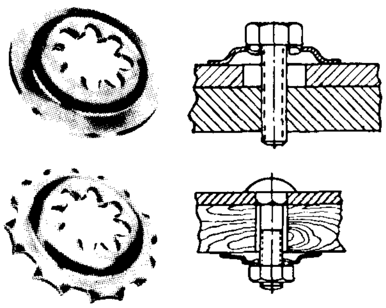
Countersunk

Used under countersunk screw heads. Standard sizes from No. 2 to ½ in. for 82 and 100 deg head styles.



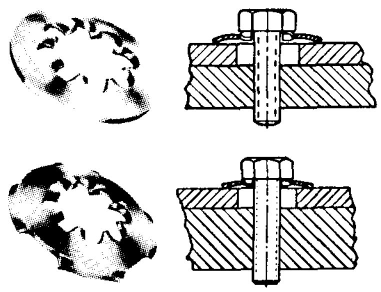
External-Internal

Cover oversize holes. Often used against soft materials and thin sheet steel because of large bearing surface. Can be used between two members which require adjustment before tightening. Resist rotation and sliding movements in final assembly. Can also be used for electrical bonding since teeth penetrate metal coatings to provide ground. Standard sizes from No. 4 to ⅝ in.



Dome Type

Recommended for use on soft or thin materials. Also suitable for use over large or elongated clearance holes. Plain-periphery type will not mar workpiece. Tooth-periphery type bites into workpiece and prohibits shifting. Standard sizes from No. 10 to ⅝ in.



Dished Type

Covers large bearing surface. Recommended for same general applications as dome type but where resilience rather than rigidity is required. Standard sizes from No. 4 to ½ in.

Fig. 4.66. Various Types of Tooth Lock Washers (from Ref. 32).

4.6.4 Other Locking Devices

In addition to fastener locking, locking devices for the stem and/or packing gland are incorporated in some valves. These are generally sheet metal stamped out in the shape of a 12-point socket to fit over the square or hexagonal cross section of the stem or packing gland nut. This type of device has a handle with a screw hole in it through which a screw is threaded into a stationary structural part of the valve, as illustrated in Fig. 4.67.

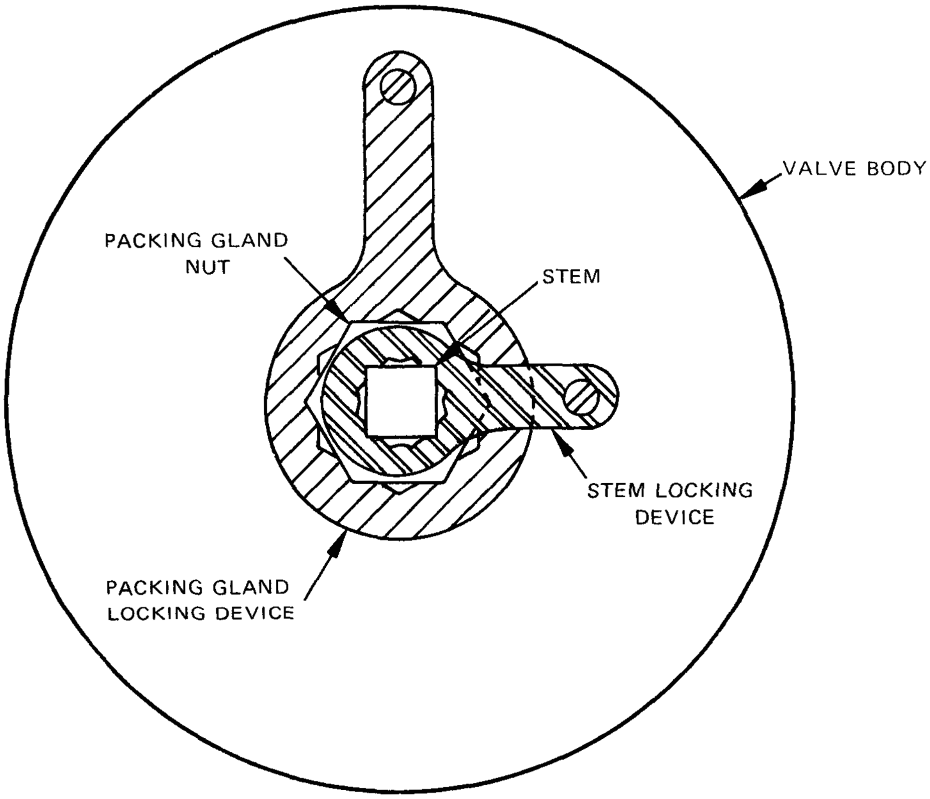


Fig. 4.67. Valve Stem and Packing Gland Locking Devices.

4.6.5 Selection and Application of Locking Devices

When selecting a locknut or locking screw or bolt for a specific application, the following factors should be considered.

1. Reliability. Locking fasteners are not uniform in their locking ability. Shape distortion, which is a common method of adding the

locking feature, will cause the degree of locking to vary from one type to another.

2. Vibration. If vibration is severe, a locking fastener with as large a frictional area as possible should be used. Depending upon the type, locking pressure can be evenly distributed or concentrated on a few threads.

3. Seating. Free-spinning locking fasteners lock only when they are tightly seated against the surface of the joint and are free to spin off if there is a relaxation in the joint. However, some free-spinning types take permanent set upon seating and become in effect prevailing-torque types. These will not move when pressure against the work is removed.

4. Reusability. If the bolted joint is to be disassembled for repairs or inspection, a locking fastener that can be easily reused should be selected. Some types jam a few threads together to obtain the locking action, and these can damage the thread so that replacement may be necessary.

5. Assembly. If speed of assembly is important, the use of a non-directional locknut may be indicated. A free-spinning locknut is preferable for use in inaccessible locations. Locking screws and bolts generally create more difficult orientation problems.

6. Temperature. Plastic and other nonmetallic inserts are used in some locking fasteners to obtain the locking action by frictional interference. These insert materials are not recommended for use where the temperature will be above 250°F.

Ninety percent of the torque required to correctly stress a non-lubricated assembly is dissipated by friction at the work surface and the threads; only 10% induces tension in the parts. When a locking fastener is lubricated with wax or cadmium plating, the frictional loss at the bearing surface and in the threads is greatly reduced. Considerably more applied torque is translated into useful tension. A $\pm 30\%$ variation in friction characteristics greatly affects the estimated induced tension. This friction variable is considered reasonable. However, there are no known methods of accurately controlling or predicting the friction values that will be encountered under assembly-line conditions. Even if design specifications keep these friction variables within control limits for preloading, the tightening methods induce significant error.

The most reliable tightening method of preloading bolts with locknuts to very high loads appears to be the turn-of-nut method. In this method, a wrench-tight or snug-tight condition is followed by a calculated amount of additional rotation. This achieves an assembly within calculated tension values. As long as these achieved tension values exceed the operating load, the assembly resists vibration and fatigue. The turn-of-nut method is reliable in solid joints, but flexible joints have factors, such as tubing deformation and thickness and compressibility of gaskets, that tend to cause a loss of initial tension. The turn-of-nut method is not considered reliable on nuts smaller than 3/8 in. and is not adaptable to most assembly lines.

In mass-produced assemblies, power tightening to predetermined values of torque is the common method of assembly. With this method, there is the normal friction plus the torque variation in power tightening. A torque variation of $\pm 15\%$ can be expected even under well-managed production conditions. This variation is caused by fluctuations in air pressure, inconsistent wrench dwell, and the tendency of wrenches to drift off setting as they become heated.

4.7 Bellows

The positive assurance of leak-tight joints provided by bellows is an important safety factor that makes bellows particularly suited to reactor plant applications where no leakage can be tolerated because of the toxicity, potential toxicity, or chemical reactivity of some fluids in the system. By the most general definition, a bellows is a device to seal a pressure boundary by the use of a thin, flexible membrane through which motion can be transmitted. The applications of bellows, their configurations, how they are made, what they are made of, how they are designed, and some of the problems and limitations in their use are considered in this section.

4.7.1 Applications

There are a wide variety of uses for bellows in nuclear plants. Bellows are used in piping systems to increase the flexibility of a run of pipe by accommodating the expansion and contraction of the pipe caused by thermal, pressure, and creep effects. Bellows may be used in valves to provide a leak-tight seal between the stem and bonnet. In this application, the flexibility of the bellows permits enough movement of the stem to operate the disc while maintaining a hermatic seal between the stem and bonnet. In the most common arrangement, the lower end of the bellows is joined to the stem near the disc and the upper end is joined to the inside wall of the bonnet. Since the movement of the bellows is limited to 15 to 20% of its free length, the height of the bonnet nozzle tends to be long in this design.

Bellows stem seals are usually backed up by a secondary enclosure and standard packing seal, both designed to full-system pressure. In fact, the ANSI Standard Code for Pressure Piping, ANSI B31.7-1969, prohibits the use of bellows for Class I piping. In this case, the bellows cannot be recognized as a satisfactory pressure boundary and must be backed up by a code-approved enclosure. A remote indicating device on the outer enclosure is used to indicate bellows leakage or failure.

A special application of bellows is in the stem seal of safety valves because the bellows resistance to stem movement is relatively small, unchanging with use, and predictable. The packing type of stem seal offers a high and unpredictable resistance force. The use of a bellows significantly simplifies the design of the operative spring mechanism and permits the accurate setting of the operative pressure for the valve. A valve cap is used to cover the spring mechanism to provide the approved enclosure.

Piping bellows and valve bellows are discussed in the ensuing subsections on the basis that (1) the technology is not significantly different and (2) the bellows expansion joint is being used in nuclear plant containment penetrations. These two applications utilize the same convolution design (dictated mostly by pressure considerations), same stress analyses, same materials, and many of the same reinforcements. They differ in that the emphasis in valve design is on reducing the diameter and length of the bellows per unit expansion and no erosion control device, such as internal sleeves, is required.

4.7.2 Configurations

Some of the various bellows configurations that are used in actual practice are illustrated in Fig. 4.68. Each configuration exhibits different characteristics. The parameters that are affected by varying the shape of the bellows are stiffness, flexibility per unit length and unit diameter, internal pressure capabilities, reverse pressure capabilities, and the type of flexibility.

Thin plate types of bellows are generally constructed in two forms: welded discs and convolutions formed from a thin-wall cylinder. Designs a through i illustrated in Fig. 4.68 are of the welded disc type, and designs j through l are formed from a thin-walled cylinder. It may be seen that more design variations are possible with welded disc designs, but this advantage is offset by the fact that so many welds are needed.

Designs a and b illustrated in Fig. 4.68 have the advantage of simplicity. They consist only of a large number of discs, flat in one case and conical in the other, that are welded peripherally to form the bellows design. The stepped-disc, design c in Fig. 4.68, is another popular version of a simple welded bellows. The basic advantage of these designs is their capability of providing a high degree of flexibility in a short axial length. However, the pressure-retaining capability of these designs is limited. Pressure applied either internally or externally to the bellows will cause overstress of the disc because of bulging of the centers of the bellows. In design b shown in Fig. 4.68, dashed lines illustrate how the discs deform with internal pressure. Depending on the angle of the conical disc bellows, there is some self-support between the bellows elements to provide pressure-retention capability when the bellows is at its natural length. The ability of the individual discs to provide support to each other is greatest when the bellows is fully compressed and diminishes as the bellows is expanded axially.

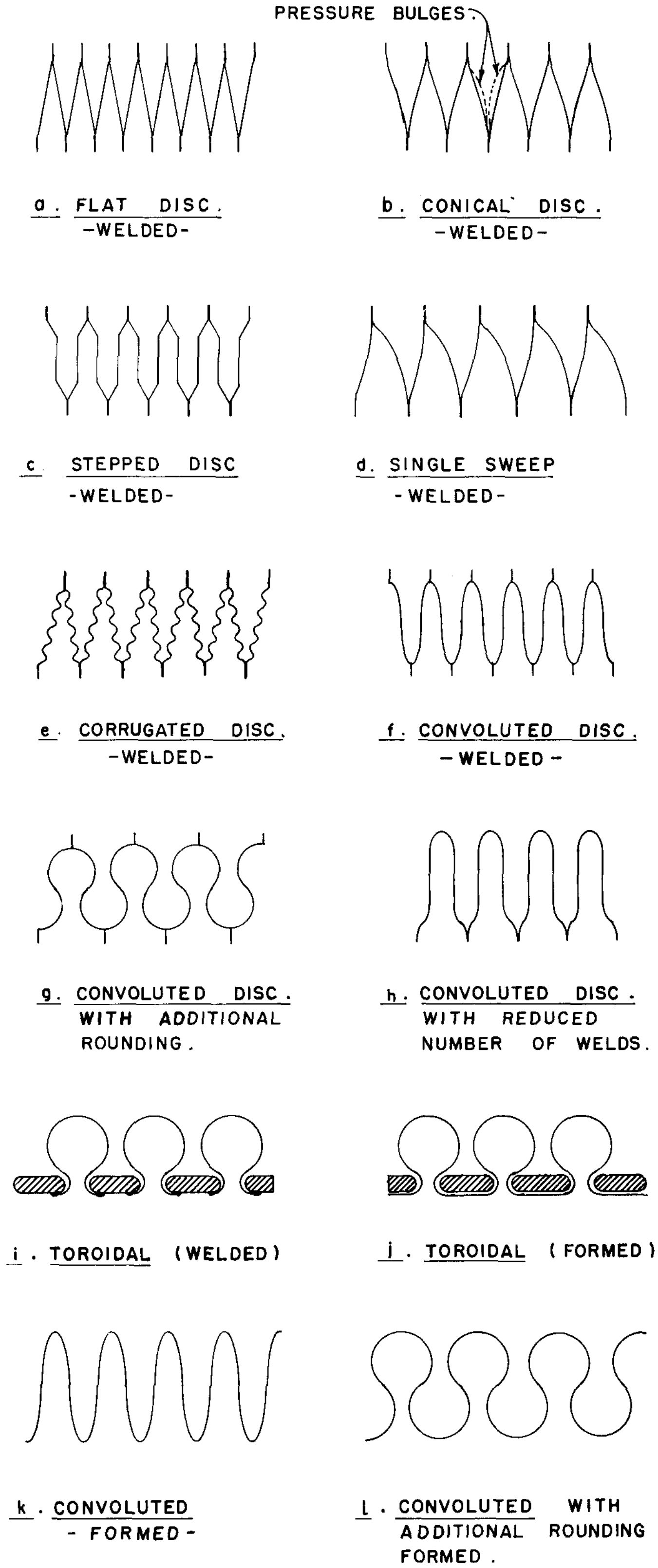


Fig. 4.68. Typical Bellows Configurations.

Where pressure retention is a design factor to be considered, the use of limit stops can be a very important means of avoiding bellows failure.

Welded bellows are used in applications involving moderate pressures and large axial movement at low spring rates. In contrast to formed bellows, the maximum pressure and deflection stresses in welded bellows of standard design always occur near the root and crown welds. This is undesirable since it means that the maximum stresses occur in a notched heat-affected zone. The change in section resulting from the weld bead also represents a possible source of stress concentration.

One of the more recently proposed designs for welded bellows is that of the single sweep illustrated in Fig. 4.68(d) in which the weld location is shifted so that the stresses near the crown and root welds are virtually eliminated. This design change involves tilting the bellows flats with respect to the axis of the bellows. By reducing the stresses near the welds so that the maximum stresses occur away from the weld areas and in an area where the metal has the properties of the original sheet material, the fatigue life of the welded bellows should be improved significantly. It is believed that this slight design change alone will result in a major improvement in the operating characteristics of welded bellows if optimum tilted flat configurations can be found for most shapes.

In the corrugated disc, design e in Fig. 4.68, concentric annular corrugations are provided in each disc element to further increase the flexibility of the bellows. This design has even greater flexibility than design a, the flat disc design. However, the pressure-retention capability of this design may be correspondingly less because a new mode or pressure failure is introduced. Because of the added corrugations, not much support is provided in the thin elements against stress applied from one edge of the disc to the other. High stresses can result in failure of the inside edge of the corrugated discs since the corrugations essentially eliminate the support of the remainder of the disc toward alleviating the stresses at this point.

Designs f, g, and h illustrated in Fig. 4.68 are all variations of the convoluted type of bellows. Many variations are possible, but these three exhibit the basic features of the convoluted design. Design f is a basic sinusoidal type of configuration. It does not have the flexibility per unit length of the earlier designs, but it is a popular design that uses the best features of the previous designs without suffering to a large degree in any respect. It has adequate flexibility and a fair amount of pressure-retention capability. The pressure-retention capability is enhanced in its alternate design, g, in which the additional rounding minimizes bending stresses caused by pressure. The number of necessary welds is minimized in design h in that only one weld per bellows convolution is required. Here again, the stress support at the weld is not as great as in other designs, and this may result in failures at the weld.

Designs i and j illustrated in Fig. 4.68 are of the toroidal type of bellows. Design i requires welding of each convolution, and design j is of the formed type. The toroidal type of bellows has a substantial pressure capability and an added resistance to hoop stress not present in other designs.

Designs k and l illustrated in Fig. 4.68 are substantially the same as designs f and g except for the method of fabrication, which is to form a cylinder into the desired shape. It should be noted that some shapes are possible through the use of this technique and some are not. Design h, for example, cannot easily be formed this way.

In summary, the most commonly manufactured welded bellows are made up of shaped diaphragms that are alternately welded together at the inner and outer diameters. Although more expensive to manufacture than formed bellows, welded bellows offer three significant advantages: (1) a wider choice of materials, (2) more deflection per unit length that results in shorter assemblies or longer strokes, and (3) a wider choice of performance characteristics because of a greater variety of convolute dimensions and shapes. Unfortunately, the large amount of welding required makes fatigue failure less predictable for welded bellows than for other types.

A general comment relevant to the shapes of bellows just discussed made on page 215 in Ref. 33 is that the early preference for close-pitch elements has largely disappeared in favor of better pressure capacity and a lesser amount of critically located welding for shaped contours. Close-pitch elements afford less self-cleaning and are more prone to collect sediment which may interfere with compressive movement. Nor are any of the bellows contours self-draining for joints in a horizontal position. Drain connections in bellows involve serious stress intensification and possible weld flaws and should be avoided.[33]

4.7.3 Reinforcement

Various bellows reinforcement techniques are illustrated in Fig. 4.69. Although there are other techniques possible, those illustrated indicate some of the methods used to reinforce bellows. Various internal and external rings, linkages, and cylinders can be used to provide effective reinforcement of bellows, and several types of reinforcement may be necessary. The bellows may bulge or instabilities may occur because of high internal pressures. Thus, reinforcement may be needed to allow the bellows to withstand the hoop stresses. Axial stresses may also exist whereby the bellows will try to extend axially until fully extended under pressure. A limit stop is used to regulate axial travel. Without such a stop, the internal pressure will force the bellows open and, in combination with the hoop stresses and bending stresses, will yield the bellows to failure. Bellows are sometimes called upon to withstand reverse pressure. Such is the case in penetration joints. In these cases, internal reinforcement may be necessary to keep the bellows convolutions from being caved in by the external pressure.

In general, reinforcement can be applied to bellows in various ways to simplify the design of the bellows so that it can perform its function more readily. The most popular types of reinforcement are the ring reinforcement used in the convoluted types of bellows, as illustrated in Fig. 4.69(a) and (b), and the cylinder reinforcement, illustrated in Fig. 4.69(d), used with many different types of bellows.

Bellows are sometimes designed with two-ply construction. Instead of one thin sheet providing the pressure-retention medium for the bellows, there are two such sheets back-to-back. If each sheet is capable of full pressure retention, a double backup is provided for the pressure-retention capability of the system. A connection can be made to the space between the two plies and brought out to an external monitoring system to detect

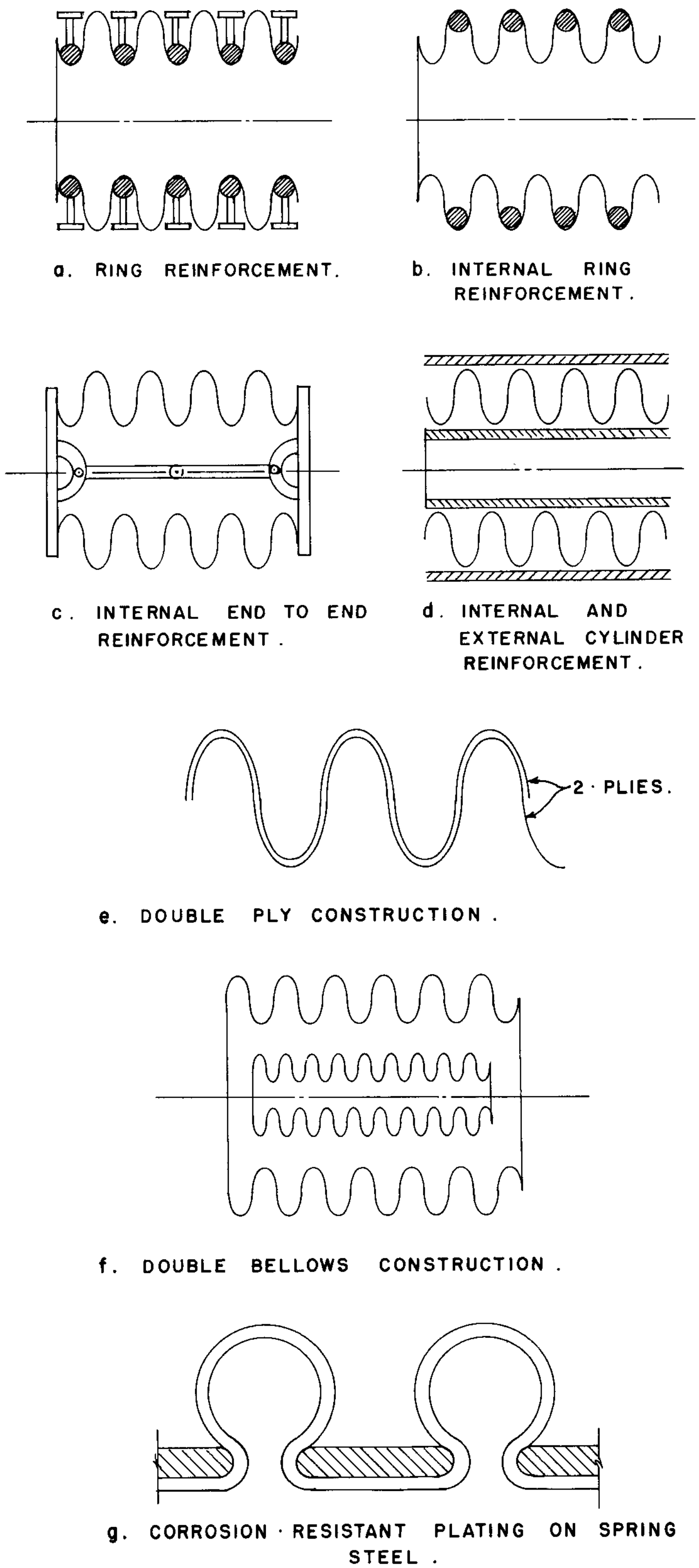


Fig. 4.69. Various Bellows Reinforcement and Limit-Stop Techniques.

failure of the first ply. A more complete discussion of plies and layered construction of bellows is given in Ref. 33. The advantages of providing two or more plies are that additional pressure support is provided to the first ply and that severity of a rupture is reduced. However, more development of the state-of-the-art of bellows will be necessary to determine fully the interaction between plies.

4.7.4 Fabrication

A metal tube can be convolute-shaped by continuous roll forming, a single convolution can be rolled from a metal strip, and convolutions can be formed hydraulically. Heat treating may be an important part of the fabrication process. If stainless steel is the material of fabrication and if subsequent welding must be done to assemble the bellows, provisions must be made through heat treatment to prevent intergranular corrosion. During fabrication and welding, the temperatures may reach those in which sensitization occurs and the following temperature quench may not be quick enough to preclude chromium migration to the grain boundaries. Normally stabilized steels should be used but an annealing heat treatment may still be called for as an additional safeguard. It is important that such heat treatments encompass the whole bellows and that quenching not cause significant warpage, especially where ply construction is used. Such an annealing heat treatment can be instrumental in relieving any internal stresses which can be the source of possible stress corrosion and intragranular cracking characteristic of this type of corrosion.

Welding is an important part of bellows fabrication. The geometry of the weld should establish a good weld that is not subject to tearing during flexing of the bellows. A particular problem associated with bellows welds is that the final thickness of the weld plays a large role in the stresses which the weld will experience. If the final thickness of the weld is slightly less than that of the bellows material, deflection will tend to induce a concentration of stresses at this point. On the other hand, if the thickness of the weld is greater than that of the bellows material, the bellows will be stiffened and deflection will be reduced at the weld. The resulting stress may be less at the weld, but it may have been moved to another point on the surface of the bellows and concentrated there. Therefore, the weld must be carefully made within a depth specification so that the deflections and stresses will not be redistributed in the assembly beyond what is allowable in the stress design. Normally the weld should be of the butt type, using an insert and drawing it up into the joint. Only one pass is usually used because of the limited thickness of the bellows material.

For weld quality which will least affect the cyclic life obtainable from the base material,

1. minimize the extent of welding;
2. locate welds away from areas of maximum bending stress;
3. avoid corner, fillet, blind root, and similar welds of undetermined root quality and use only butt welds where possible;
4. minimize heat-affected zones;
5. use full heat treatment on ferritic materials and an homogenizing annealing treatment on austenitic welds;
6. avoid root oxidation on welds, emphasize elimination of inclusions, and minimize undercuts with inert shielded welds where possible;
7. emphasize soundness and physical properties as nearly identical to the parent metal as possible through weld procedures;
8. weld surface must be as smooth as that of the sheet steel, which should be pickled and No. 1 finished and preferably ground;
9. use all applicable nondestructive examinations, including pressure and movement tests, to test the quality of welds; and
10. a careful fit-up is essential since no interruption in the assembled surface is tolerable.[33]

The different types of welds used in fabricating and assembling bellows are discussed in greater detail in Ref. 33.

The inspection of bellows is often difficult, and the problems are represented in the following excerpts from Ref. 33 on the testing and quality control of bellows joints.

> "Expansion joints must properly be classed with special rather than mass-produced equipment with regard to the extent of inspection and testing during manufacture and installation and of other quality controls applicable to the fabrication details and materials involved. In common with specialized

equipment, the advisable degree of control is also related to the reliability of the producer.

"Structural tests on bellows expansion joints include independent pressure and movement tests and also combined pressure and movement tests.

"Tightness tests at operating pressure are routine with each cycle of operation or period of maintenance; low and intermediate pressure air tests are often used at various stages of fabrication for the same purpose. Such tests essentially only check for leaks. Movement or flexure tests are desirable, particularly in combination with pressure in assessing the structural capacity of expansion joints and assuring that gross fabrication flaws are not present. If repeated a few times, the latter aim is more definitely assured. When extended to an appreciable number of cycles, the fatigue strength of that specific joint and also, to a lesser extent, of the general design is verified, but the joint tested is sacrificed.

"Flexure elements, in view of their critical nature, are properly subjected to all nondestructive examination applicable to the material analysis. Usually, a radiographic examination is of limited value due to the tin material, and ultrasonic examination is not sufficiently developed for general use on the configuration involved. Magnetic powder examination is applicable to ferritic materials, and the various crack detection aids, such as fluorescent penetrant oil viewed under ultraviolet illumination, penetrant dye suspensions, and volatile liquids as absorbed by chalk powder, may be used on all materials. It is preferable that the examination follow a few flexings and where practical be performed at both extreme and neutral positions since tensile stress would render flaws more readily detectable. The examination should cover the sheet material, the welds, and parts to which the bellows is directly attached."[33]

4.7.5 Materials

The stainless steels are most often used in bellows fabrication because their inherently greater strength permits them to be used in lighter thicknesses in contact with a wider range of fluids at higher temperatures and pressures than many other materials. Strength, ease of forming, and weldability are also important considerations, but the corrosion-and-erosion problem plays a major role in the selection of bellows material. Pure stress considerations such as ultimate and yield strength are of secondary importance. As a result, materials with vastly greater strength capabilities are not used.

The spring in a self-actuating safety valve may be designed with an exceptionally tough material capable of very large stresses, but the bellows used in the same safety valve for stem sealing purposes must be made of a comparatively ductile and pliable material that is much less capable of withstanding high stresses. The bellows is subject to corrosion by the system fluid and the spring is not.

> "A knowledge of the flowing medium is important from the standpoint of potential corrosion or erosion. On light-gage bellows elements, even mildly corrosive conditions may seriously affect service life in view of the high stress levels present with conventional movement ratings. Condensate corrosion during standby periods or when metal temperatures are below the dew point in service is a notorious contributor to bellows joint failures.
>
> "Deterioration of bellows material is influenced by its thin sheet material form and severe demands involving cyclic strain. Cold work, variations in analysis or structure, thermal history, inclusions, and segregation all contribute to increased sensitivity. Corrosion, particularly as associated with stress and fatigue, may result from contact with ordinary noncorrosive media so that weight-loss data or conventional corrosion tests are not reliable guides. Where a material of assured resistance to such attack is not economically available, it is necessary to reduce the stress range to purely elastic action (within twice the yield strength) and in extreme cases to no more than twice the basic system allowable stress. Superficial overall corrosion and initial traces of concentrated attack or pitting are sufficient to cause accelerated fatigue failure. The possibility of intergranular corrosion on austenitic steels, as associated with chromium depletion at the grain boundaries, should be avoided by the use of stabilized materials, with maximum resistance to other accelerated attack generally obtained if the composition is completely austenitic."[33]

It can be postulated that there is a potential, although quite minimal at this stage, for the development of plated or treated spring steels for bellows applications. Potentially, the spring steels can be plated with a corrosion-resistanct material, and use can be made of the much greater strength capabilities of these steels. However, the major obstacle which stands in the way of such designs is that the forming of spring steels into bellows configurations is not nearly as well developed as the forming of the much more ductile stainless steels.

4.7.6 Basic Stress and Strain Characteristics

The fatigue failure problem of bellows is perhaps the area that requires the most design analysis. Design analysis to preclude fatigue failure over the lifetime of bellows is a subject that demands major attention from both the procurer and the supplier. However, analysis can only go so far in assuring good bellows, and there also is large dependance upon fabrication procedures and the absence of flaws in the finished product. In its broadest sense, design analysis must therefore encompass adequate factors of safety based on thorough testing with programmed flaws. There is a certain amount of similarity between this and the case of a nuclear reactor core wherein the core heat transfer capabilities are established with analyses which include multiplication factors greater than one for so-called hot-spot conditions. The location and magnitude of hot spots within a core are established through long experience with reactor core operation and testing. Bellows design may also require the application of similar factors to the locations of supposed weak spots.

Large study efforts have been undertaken to solve the problem of fatigue failure sufficiently to permit the more widespread use of bellows. A basic, simplified treatment of bellows stress analysis is presented here to indicate bellows performance under various conditions and the important dimensionless parameters. A basic summary of bellows fatigue criteria is provided in Ref. 33 (pages 220-221).

> "As a rule, the manufacturer's ratings for bellows joints are not clear cut and do not indicate the number of cycles to failure. Ordinarily, with the high stress range attendant to the manufacturer's rated maximum movements, the number of useful cycles is apt to be only a small fraction of the 7000 cycles established for pipe in the 1955 Piping Code rules. Manufacturers are inclined to cite similar service applications in justification of their ratings; such statements must be discounted in the absence of direct data of actual service or test data, including the number of cycles."[33]

The author[33] goes on to state that it is believed that a rational approach to the bellows design problem is attainable through the application of a stress range approach to fatigue performance of bellows material accompanied by recognition and evaluation of the effect of local stress raisers that cannot be eliminated. Fatigue testing is advanced as being the soundest method for establishing the characteristics of a basic approach or of a specific design. Inasmuch as tests of each specific size of each design are impractical, the results of test performed must be translated to other joint sizes. The following approximate approach is presented[33] as one that gives reasonably useful results that parallel those obtained from tests on other piping components in the relation of strain range to cycles to failure.

$$N \propto \frac{1}{\epsilon^n} ,$$

where

N = number of cycles to failure,

ϵ = total range of unit strain caused by movement and pressure, and

n = a constant for the material used.

The exponent n ranges from 3.3 to 4 on stainless steel bellows, usually nearer the lower value. A value of 3.5 is therefore assumed as a reasonable one for stainless steel bellows elements.[33]

It is advocated[33] that a safety factor of at least 2 be provided on stress or strain in the bellows at rated conditions. This means a margin of $(2)^{3.5} \simeq 11$ on the number of cycles to failure since the change is rapid with stress variation. The establishment of a minimum performance is recommended for cases where the number of cycles in operation is expected to be rather low, and it is suggested that the same minimum number of 7000 cycles used for piping also be applied to the joints. The following approximate relation can be used in assessing or extrapolating test data from probable performance of stainless steel bellows.

$$N = \left(\frac{1,600,000}{S_R}\right)^{3.5}$$

where

N = number of cycles to failure and

S_R = calculated range of stress, psi, given by stress analysis information developed for the deflection and pressure required (varies with the design).

The design service rating should not be assumed higher than about 10% of this or

$$N_s = \left(\frac{800,000}{S_R}\right)^{3.5}$$

where N_s = the number of cycles which the joint should be expected to endure safely in service.[33] This relation can be used to combine the effect of extreme emergency movements with normal operation into a single equivalent condition. If the extreme movement produces a stress range S_1 and is expected to occur at most times N_1, the fractional part of the fatigue life used for the condition is

$$\frac{N_1}{\left(\frac{800,000}{S_1}\right)^{3.5}} ,$$

and the design number of cycles for the normal movement must be increased by the fraction to secure the equivalent number of cycles at normal movement.[33]

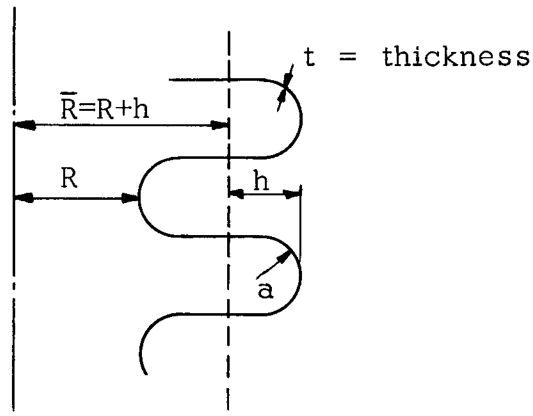
The calculation of the maximum stress within the bellows, S_R, for at least some cases of bellows design is available in the literature as a result of studies performed by various groups for the United States Atomic Energy Commission. Information about and results of such analyses are given in Refs. 34 through 37. These reports are quite thorough, even to the point of providing a sample proposed bellows procurement specification which includes detailed stress analysis design requirements.

A good way to synthesize the results of bellows analyses is to find the dimensionless parameters of interest and indicate what the gross effect is when various bellows dimensions are changed. There are at least two dimensionless parameters of interest for essentially all bellows. These indicate the basic form or geometry with regard to the proportions of various dimensions to each other in the bellows. Thus, the size of the bellows can be increased or decreased, keeping these dimensionless parameters constant, with no net change in the resultant stresses. These parameters are especially applicable to the convoluted and toroidal shapes; other parameters may be needed to define the shapes of corrugated discs and flat-plate geometries. These two dimensionless parameters are defined as follows.

$$A = \frac{a}{(\bar{R}t)^{1/0.778}}$$ indicates the basic size of the convolutions with respect to the radius and thickness of the bellows.

$$B = \frac{a}{h}$$ indicates the ellipticity of the bellows convolutions

where



Making the parameter A large means that the size of the convolutions is large as compared with the radius and thickness of the bellows. Conversely, making the parameter small means that the bellows conforms a little more in shape to a straight cylinder. Making the parameter B large means that the convolutions are somewhat flattened and shallow. When B is equal to 1, the bellows convolutions form a circle, and in the case or toroidal convolutions, are perfectly doughnut shaped. When B is small, the convolutions become very wide; and this is the typical case in the convoluted type of bellows.

Bellows theory indicates that, in general, the optimum value of the parameter A is 1.0 for both the convoluted and toroidal types of bellows with or without reinforcing rings. This more or less establishes an optimum bellows configuration in which the size of the convolutions is in good proportion with the other dimensions. With regard to pressure and the dimensionless parameter B, the more nearly spherical shape always withstands pressure forces better. The membrane theory indicates that the more stresses which can be absorbed as membrane stress rather than bending stress, the better. In any case, the optimum value of parameter B with respect to pressure stresses is that which gives a toroidal configuration of convolution.

From the standpoint of deflection stresses, the picture is much more complex. In some cases it may be desirable to increase the ellipticity of the convolutions, and in others it may not. One thing should be noted clearly. Increasing both the radius of the bellows and the thickness of the wall material, t, is usually always desirable from the standpoint of stress reduction. In fact, it is difficult to design a bellows with satisfactory deflection in a small-radius configuration. In a differential pressure cell, for example, optimization of bellows design may call for a wide large-diameter bellows to assist in reducing the number of necessary convolutions. The same thing applies to the design of a bellows to be used in sealing the stem of the valve; the diameter of the bellows may have to be larger than actually necessary for stem clearance to assist in reducing the necessary length of the bellows.

A similar situation exists with respect to the wall thickness, t, of the bellows. A greater thickness rapidly increases the deflection spring force of the bellows. However, when applied evenly, the increased thickness tends to even out the bending stresses in the bellows wall so that the net result is a reduction in the maximum stress experienced by the bellows even though the spring force is much greater, making the bellows stiffer.

There are perhaps more sidelights which are not immediately evident intuitively that can be obtained from a thorough review of bellows stress analysis theory. Most aspects of standard bellows design are covered in Refs. 34 through 37. Unusual design configurations, such as variations in wall thickness of bellows, uneven convolution sizes, the effects of more than one ply, and various other unusual features, do not lend themselves to definitive analysis within the present state-of-the-art.

There is a particular sidelight of bellows analysis that should be noted carefully. Given the usual fabrication procedures in a practical situation, weak spots of various sizes may very well exist within the bellows wall wherein the stregth of the wall may be slightly less than in other portions of the bellows. These weak spots may be caused by slight welding irregularities, corrosion pits, or the forming process itself; and they have a significant bearing on the fatigue life of the bellows. In most static stress design problems, a weak spot may be tolerable if its strength is above the minimum design requirement. However, an additional complicating factor arises in bellows design. Weak spots distort the stress pattern so that the bellows deflection is redistributed and deflection is increased at the weak spot, compounding the problem. If the weak spots are severe, the bellows design analysis may be rendered invalid even though sufficient conservatism was applied to the numbers to allow for the existence of these weak spots within normal design practice.

Stated another way, suppose a bellows was fabricated in a perfectly uniform manner so that the thickness of the wall material was exactly equal to the minimum required thickness for the design conditions. Adding additional thickness to this in various places, even to the point of adding thickness almost completely uniformly, may result in bellows failure because of the redistribution of deflection which unavoidably occurs. Unlike a chain, which is only as strong as its weakest link, a bellows is weaker than its weakest link.

The application of a bellows to a valve stem in high-temperature service poses a similar design problem. Bellows can be designed to withstand very high temperatures, and they can generally be designed to withstand any temperature which the surrounding piping can withstand. However, the temperature of the bellows must be uniform or else the higher temperature convolutions will take a disproportionate amount of travel and stress because they will flex more readily than the colder convolutions. In a dead-end branch application such as a valve stem, special techniques may be required to assure an even temperature for the bellows. One such technique is the use of an insulated housing over the entire bonnet to distribute the heat.

A few words should be said about the general capabilities of bellows as presently designed. The capabilities referred to are percentages of deflection, pressure, and reverse pressure. For convoluted and toroidal types of bellows where each convolution has an axial length significantly greater than the thickness of the wall material, deflections of as much as 10% are difficult to attain. This includes, by geometrical correlation, angular and offset deflections. About a 20-in. axial neutral length bellows would be required for stem seal purposes in a 2-in. gate valve. The more compact bellows that have more convolutions per unit length, such as the flat plate and corrugated types, may reach deflections of an order of magnitude of 50% or more, but this is offset by the generally poor pressure-retention capability of these bellows designs. For high-pressure valves that require a large stem movement, such as gate valves, the space requirements quickly become prohibitive in larger valves.

The instabilities and "unusual" behavior of bellows should also be considered. Generally, the various phenomena noted as having to do with bellows "squirming" all have a common cause. The large pressure stresses within a bellows tend to attempt to increase the length of the bellows and hence the effective pressure area. If at any condition of its use the bellows is capable of increasing the internal effective pressure area on the bellows wall, a squirming instability can occur. This can easily be visualized in the case of a long bellows where the ends are held fixed with varying degrees of freedom and the bellows is subject to internal pressure. The middle of the bellows will tend to move off the axial center line in a sideward direction, as illustrated in Fig. 4.70, in such a way that will initiate an unusual deflection leading to a blowout. This can happen all the more readily if the ends of the bellows are not fixed from the standpoint of angular rotation. It is mandatory that such an instability in bellows be precluded as even small "squirms" can be detrimental to the life expectancy of a bellows. The onset of this instability can generally be precluded by giving proper attention to this facet of bellows behavior in the design analysis. This effect is discussed in Refs. 34 through 37, and design measures directed toward precluding it are provided in the sample specifications.

4.7.7 Frequent Problems

As previously noted, the most basic problem of bellows design is associated with the fact that bellows fatigue can vary over a wide range even with strict control of fabrication processes to insure uniformity of wall material. This is the problem of weak-spot fatigue. Deflection

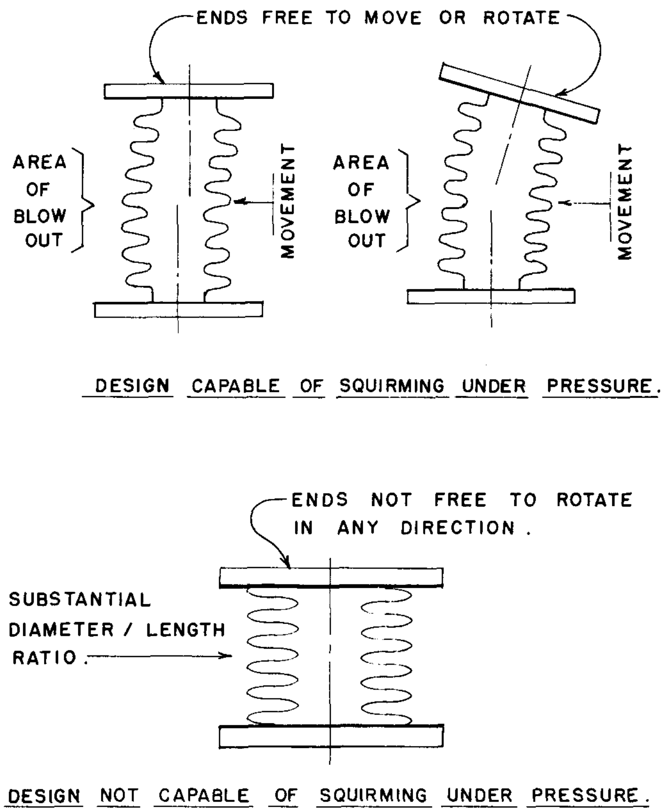


Fig. 4.70. Bellows Squirming Instability.

tends to become uneven and concentrated at the weak spots, increasing the stress at these locations. The principal characteristic of bellows failure is that it tends to be a major one. Bellows generally do not leak; they blow out.

It is apparent that at this point in the development of bellows and at least with the single-ply construction, the potential for a bellows blowout should be considered as part of the accident design for the system so that appropriate safeguards can be provided. If a double bellows arrangement, such as one bellows within the other for double backup protection, is provided, it may be desirable to shift the neutral deflection positions of each bellows so that failure of the inside bellows at a maximum deflection position will occur with the backup bellows in a more nearly neutral deflection position and therefore a lower stress condition.

As another example, it can be important in some bellows applications to provide flow restricting devices to minimize the hazardous effects of a bellows blowout. When the bellows is used as a valve stem seal, the annular opening between the stem and bonnet should be made as narrow as possible to restrict flow in case of a blowout. The hazard of bellows blowout in this application can be further reduced by using a gasketed stem seal as a backup. In such a case, the bellows serves only to assure the absolute leak-tightness demanded by the operational requirements for the system.

In any fluid system and any component of a fluid system, the effects of corrosion and erosion on the pressure boundary must be considered. These effects are applicable to valves, pumps, and pipes but are invested with a new significance in the case of bellows. In connection with weak spots, corrosion and erosion processes can provide many sites within the bellows where failure can occur. From a procurement standpoint, the particularly difficult part of the problem is brought about by the difficulty or impossibility of gauging the extent to which corrosion and erosion can affect the lifetime of bellows by any reasonable testing program. Reliance must be placed purely on corrosion data for the material and on the provision of safeguards to eliminate erosion.

With respect to corrosion, there is a particularly susceptible area in bellows design. This is the problem of water or moisture collecting in the bellows convolutions. The toroidal and convoluted bellows are prone to accumulate moisture even in a vertical orientation. It is impractical to attempt to drain all convolutions even if it could be done without developing stress concentrations.

Exposure to penetrating types of corrosion should be avoided at all costs. This includes stress corrosion cracking and intergranular corrosion cracking. In cases where a protective film, a passive film in the case of stainless steel, is required on the inside surfaces of the bellows for corrosion protection, special system operations to establish this film should be considered early in the plant life to assure that this protective film is provided on all bellows surfaces, including those normally quite inaccessible to system fluid at temperature.

A particular problem of bellows used in liquid-metal systems is the possibility that the liquid metal will freeze within the convolutions. Normally, bellows used for sealing purposes are of such a length that the temperature decreases as the heat proceeds up the seal. If the temperature at the cold end of the bellows is not quite adequate to sustain the temperature of the liquid metal above its freezing point, such freezing could occur. If the bellows is used for a valve stem seal, the valve could be rendered inoperable by this freezing. From this standpoint, a bellows design that has a minimal volume per convolution and convolutions which are drainable is desirable. A flat disc type of bellows or a conical disc type of bellows may be more desirable than convoluted bellows, and convoluted bellows may be more desirable than toroidal bellows.

There are many lesser problems that can occur during bellows operation. These include such things as the loss of flexibility caused by changes in the material or the formation of kinks or distortions in the shell of the bellows. Many of these smaller problems are unusual and are normally precluded by using prescribed methods of selecting the proper bellows configuration and the best stress analysis principles available.

One practical consideration related to the use of bellows in nuclear reactor plants is the formation of crud traps within the convolutions of the bellows. If the bellows is used for containment of highly radioactive material, there is a good possiblity that a large amount of crud will develop within the convolutions and become permanently embedded within the bellows because of the inaccessibility for decontamination. Rather than attempting to provide adequate cleansing flow into and through the convolutions, it is better to provide for ease of bellows replacement during design.

4.7.8 Design Areas Needing Further Investigation

There are a number of areas which can perhaps be profitably investigated to further advance the state-of-the-art of bellows design. Some may require major study and analysis and some may require major advances in fabrication technique.

More emphasis can possibly be placed on the use of more than one ply in bellows construction. The use of double or even triple ply construction does not seem to have been fully investigated. The potential for providing additional outside support through the use of stronger materials for the outside plies may merit more consideration. An inside ply of stainless steel and outside plies of spring steel may very well increase the capabilities of bellows, and the use of a corrosion-resistant plating on the inside is also seen to have possible merit.

From the standpoint of tampering with the geometry of bellows, the problem of bellows drainage could well be expedited by the use of a spiral design for the convolutions if the bellows is to be oriented with its axis vertical. This could be particularly important for the toroidal bellows, which are susceptible to drainage problems. The fabrication of spiral bellows appears to be possible by using the hydro-form technique. Stress analysis techniques for bellows would have to be revised, especially in the smaller radius bellows, to cover a spiral design. Other geometrical changes could be developed to provide better reversed-pressure capabilities. An inverted toroidal design might be feasible where the external pressure is high.

5. VALVE RELIABILITY REVIEW

The objective of the valve designer is to achieve a high degree of reliability and functional effectiveness at an economically justifiable cost. Both general and detailed information on gate, globe, and check valves of 4-in. size and larger and considerations relevant to the use of these valves in systems carrying radioactive fluids have been presented in the preceding sections of this document as guidelines which may be used by the designer to achieve this high reliability. Techniques which can be used to determine how well these guidelines have been followed are presented here.

5.1 Reliability Requirements

The degree of reliability required for a valve is a function of its application in a system. If the designer requires a high degree of system reliability, it may be necessary when evaluating a system design to establish a very low failure rate for a valve to achieve this value of system reliability. Some flexibility with respect to the required reliability of a valve in the system may be obtained by the proper application of redundancy. However, a detailed system reliability analysis is required to determine the best combination of components to achieve the desired result.

System quantitative reliability analysis is a technique used by system engineers to determine the comparative reliability and maintainability attributes of two or more candidate system configurations composed of identical or similar parts. This technique is based on combining the failure rates and repair rates of all system components in a predetermined manner which best describes the configuration. These failure rates and repair rates are usually generic rates based on a general category of valves rather than a specific valve.

When applying the comparative technique to a system design, it is necessary that the system engineer satisfy himself that the assumed failure rate for the component is consistent with that which can be achieved in actual practice. High reliability cannot be achieved without requiring a high-quality valve. Therefore, adequate provisions must be made to insure that quality is designed and built into the valve.

The sensitivity analysis is a useful engineering tool for identifying critical components. Use of this analysis permits determination of the sensitivity of the system reliability to the reliability of a component or group of components. The analysis is performed by varying the failure rate of the component being examined over several orders of magnitude to determine whether this variation significantly affects the overall reliability value of the system. The remaining step is to identify the degree of quality control necessary to achieve a level of performance consistent with the assumed failure rate. This is necessary to insure that the configuration selected through use of the system reliability analysis is not degraded as a result of unrealistic assumptions for component reliability.

Accurate quantitative values for the reliability of various classes of valves used in nuclear plants are not available. Two approaches should be considered when establishing the failure rate value to be used in determining system reliability.

1. In those cases where the sensitivity analysis indicates that the failure rate can vary over several orders of magnitude without affecting system reliability significantly, the user can rely on data available in the marine and aircraft fields since the valves manufactured for those applications are of higher quality than those used for normal commercial applications. The availability of such data is discussed in Subsection 5.2.3.

2. In those cases where the sensitivity analysis indicates that the failure rate is extremely critical in its effect on overall system reliability, it will be necessary to conduct a valve test program under the environmental conditions to which the valve will be subjected during its normal life.

The failure rate exhibited by a component is the result of the various environmental and operational stresses to which it is subjected. In this context, stress is a measure of any parameter (pressure, temperature, number of cycles, fluid purity, etc.) that becomes a function of the valve failure rate. Each of these stresses may be responsible for a part of the total component failures recorded. Thus, the failure rate of a component may be looked upon as the sum of the partial failure rates corresponding to each stress level. Components are designed to withstand certain nominal stresses or combinations of operating conditions. They are therefore rated for this particular combination of stresses (pressure, temperature, etc.), and a certain failure rate may be observed in a statistical test of valves operated at these nominal conditions.

An increase in any particular stress level will likely result in a higher failure rate; that is, lower reliability. For example, if the speed of a chemical reaction is approximately doubled for each 17°F rise in temperature, as stated by the Arrhenius equation, the failure rate attributed to thermal stress could very well double for each 17°F use in component temperature. If thermal stress failures are a significant part of the failure rate of a component, temperature effects would be critical to the reliability of the component. Similarly, increased stress levels attributed to pressure, vibration, shock, etc., may contribute significantly to higher failure rates.

Conversely, the failure rate may be reduced appreciably by operating components at one-half to three-fourths of their rated values. This use of components at significantly lower than rated stress levels is called "derating". Unfortunately, the correlation between stress levels and failure rate is seldom known for valves, thereby introducing uncertainties. The usual method of allowing for these uncertainties is the use of safety factors, which specify the percentage by which the net design rating must exceed the anticipated operating conditions.

Derating techniques and safety factors are important tools for designing reliability into a component. In large valves where the relationship between operating conditions and failure rate is generally unavailable, the use of ample safety factors is more popular than derating for designing components with high reliability.

5.2 Design Considerations

The failure mode and effects analysis, the check list used in this analysis, and the fundamentals of a valve test program by which absolute failure rate data may be obtained are discussed in this section. The failure mode and effects analysis technique is an engineering tool which can be used to evaluate the realiability of a valve. The check list is a management tool designed to insure that all the design aspects of the valve that have an effect on reliability have been given proper attention. The test program involves the use of the analysis and the check list to define a meaningful test. Valve tests are performed only when existing data are inadequate or the dependency on the valve to achieve high system reliability is critical. The failure mode and effects analysis should be performed if existing data are used to establish the valve failure rate or if the valve will be tested to establish the failure rate. In the former case, a high confidence level can be established to show that the data are being applied properly. In the latter case, the analysis is useful in establishing a more meaningful test and serves as a check to see that all aspects of the design have been considered.

5.2.1 Failure Mode and Effects Analysis

The failure mode and effects analysis is a procedure for systematically identifying potential failures and analyzing their effects with respect to the application of the valve or valves in a given system. Decisions regarding details of component design, system design, and of quality control requirements during manufacture and installation can be made as a result of performing this analysis. The analysis is usually facilitated by the use of a tabular form, as illustrated in Fig. 5.1. Normally, the table is completed by those responsible for the system design rather than the reliability specialist. The results should then be used to aid in decisions regarding the details of component design, system design, and of quality control requirements during manufacture and installation.

FAILURE MODE AND EFFECTS ANALYSIS

ITEM OR PART	FAILURE MODE	POSSIBLE CAUSES	EFFECT OF FAILURE	QUALITATIVE FAILURE PROBABILITY	REMARKS AND RECOMMENDATION

Fig. 5.1. Typical Tabular Form for Failure Mode and Effects Analysis.

(a) Failure Mode. Failure modes of valves can be categorized independently of the types of valve considered. A convenient categorization is outlined as follows.

1. Leakage Modes
   (a) Leakage through disc-seat interface
   (b) Leakage through stem seal, packing, or pivot pin seal (check valves)
   (c) Leakage through body-to-bonnet joint
   (d) Leakage through wall because of porous casting or welds
   (e) Leakage through wall caused by structural rupture
2. Improper Operation Modes
   (a) Fail to open
   (b) Fail to close

Each of these failure modes can be divided into various degrees or grades of failure. Each grade of failure may have a different effect, which will depend on the type of system, the valve design, type of fluid handled, etc. The only failures discussed here are those that may occur during the course of system operation. The type of failure that is directly a result of an engineering error, such as a valve that is insufficiently sized for the application, is not considered.

(b) Mechanisms of Failure. The mechanisms of failure can be categorized in terms of the stage of the valve lifetime at which they are introduced. These stages and the methods of deficiency prevention are tabulated below.

Stage	Deficiency Prevention Method
Design	Design review
Manufacturing	Quality control
Installation	Inspection and operational test
Operational	Procedure, inspection, monitoring, maintenance, and repair

The failures usually considered are those that occur at the operating stage. However, the causes of failure may be indicated by deficient performance at any one or more of the above four stages. A typical list of failure modes and possible causes is given in Table 5.1. The causes given in Table 5.1 should not necessarily be thought of as being introduced at the operational stage.

Valves usually fail by a variety of modes. However, one mode is usually more prevalent than others. A concentration of preventive effort on this mode will produce the greatest results in the shortest length of time. The elimination of failure causes should be conducted at all four stages by the previously listed methods, but the only method discussed here is with respect to the design stage. Critical areas that should be subject to design review are considered. The details of quality control and verification of proper installation and operation are more properly subjects of procurement and operation manuals.

Table 5.1. Failure Modes and Possible Causes

Failure Mode	Causes
Leakage modes	
Through stem seal or packing	1. Stem wear (rotational or linear)
	2. Pitting of stem (chemical corrosion)
	3. Packing wear
	4. Packing deformation
	a. Eccentric stem
	b. Misaligned stem
	c. Successive freezing and thawing of boric acid
	5. Seal wear
	6. Gland or seal retention stud or bolt structural failure
	7. Gland or seal tetention stud or bolt vibrates loose
	8. Packing not properly compressed
	a. Misaligned packing gland
	b. Not tight
	9. Bellows failure (special application)
Through body-to-bonnet joint	1. Seal leakage (through O-ring or gasket)
	2. Retaining stud or bolt structural failure
	3. Retaining stud or bolt vibrates loose
Through wall	1. Erosion
	2. Corrosion
	3. Porous casting or welds
	4. Rupture
Through disc-seat interface	1. Erosion
	2. Corrosion
	3. Scoring, spalling, fretting, wear, and cracks
	4. Particle interference
	5. Valve seat distortion
	6. Improperly seating disc (misaligned)
	7. Improper contact
Operational modes	
Stick in open or closed position	1. Stem binding
	a. Tight packing
	b. Misalignment
	c. Chemical precipitation
	d. Thermal expansion in full-open or full-closed position
	2. Differential thermal expansion of seat-disc
	3. Overtorqued in the backseat or closed position
	4. Improper travel limit devices

Table 5.1 (continued)

Failure Mode	Causes
Operational modes	
Disc detachment	1. Structural failure of retaining devices
	2. Retaining device loosens
Stem fracture	1. Fatigue
	2. Overtorquing
	3. Corrosion

(c) Effects of Failure. The same failure mode can have significantly different effects, which depend to a large extent upon the system application. For example, a slight leak to atmosphere is more critical in a radioactive fluid system that is operated at temperatures above 200°F than in one operated at temperatures below 200°F because of the possibility of airborne particulate contamination. In many cases, the effect of failure can be mitigated by specific design provisions. Some examples of this are redundancy of components, allowance for some minimal leakage, provision of backseats, special material requirements (such as ductility) that are contrary to failure growth, etc. It is impossible to include all possible situations in this discussion. The system design engineer must conduct a failure mode and effects analysis for each specific system to determine the effects of specific types of failure. Some examples of specific system applications are discussed in the following paragraphs.

For valves that are inaccessible during operation because of high temperature or radioactivity levels or both, a significant leak to the atmosphere through the stem seal or packing or through the bonnet-to-body joint may require cooling down or shutdown of the system to repair. A somewhat more critical situation will exist if the valve cannot be isolated from its source of fluid since this could force a plant shutdown.

Primary coolant or recirculation loop vent and drain valves should be designed for tight seats to prevent unnecessary processing of leaking fluid. The system designer may specify that the low-pressure side (downstream) be under the stem packing to minimize the possibility of leakage to the atmosphere.

Valves which are used for steam generator feed regulation, steam generator automatic blowdown, and turbine bypass (dump to condenser) are characterized by frequent operation for throttling. Thus, sticking is the most significant failure mode for those valves, and seat leaks or leaks to the atmosphere could be tolerated and generally would not force a shutdown of the plant.

Resin sluicing systems require valves that will not impede the flow of the slurry. This calls for valves with unrestricted flow paths, such as gate valves. Angle and check valves are to be avoided, but if check valves are necessary, the swing type is more desirable than the lift type. Similarly, crevices or pockets that may catch resin beads are undesirable, particularly where radioactive resins are being used because localized zones of high radiation can be created.

The reliability of containment isolation is increased by the use of redundant valves in series. Therefore, the effect of disc-seat leakage does not become significant unless one of the redundant valves sticks in the open position. The designer of this type of system has a choice in stressing high reliability. He can choose to require high reliability for closing and allow normal nuclear standards for disc-seat leakage, or he can require extremely low leak rates and accept the normal operational performance reliability. Consideration of stem and body-to-bonnet joint leaks becomes important if the valve is located outside the containment with the high-pressure side under the stem or in the bonnet-to-body joint.

Engineered safety systems such as coolant injection and containment spray are used infrequently. However, they must be highly reliable. Most of the automatic valves for these systems are such that failure to open or close is a more significant failure mode than leakage through the seat since most are open for operation. Considerations for stem or bonnet-to-body joint leakage are the same as for containment isolation valves if they are outside the containment. Since these systems usually have redundant paths which can be cross-connected, it is extremely important that the cross-connect valves have a high degree of structural integrity. The valves are common links between redundant paths, and a structural failure represents a failure of both paths, negating the redundancy.

In some systems, valves are positioned either open or shut at a given temperature and are required to operate at a significantly different temperature after cool-down or heat-up. These valves should be designed so that differential thermal expansion will not cause sticking, jamming, or misalignment.

5.2.2 Design Reliability Check List

Many qualitative factors are involved in the reliability evaluation of a given system. Some of these factors, both general and specific, are identified in the design reliability check list. A typical example of this check list is given in Table 5.2. This check list should be used in conjunction with the check lists normally used during the design review stages of a specific design. These check lists are usually prepared initially through the combined efforts of various groups, such as system and component designers, material engineers, quality assurance engineers, and purchasing, construction, operating, maintenance, and cost control personnel. Normally, these lists are made broad enough to cover nearly all applications of valves and they are continuously revised to account for problems encountered.

The check list is made up of a series of questions to be answered "yes" or "no" by the agency responsible for the system design. Recognizing that questions may be interpreted differently by different parties, it is mandatory that "yes" answers be substantiated by documentation. It is also recognized that "no" answers may be partially compensated for by other considerations which may be presented. Use of this check list requires confirmation of the "yes" answers by inspection of the documentation and judging the degree to which the "no" answers may result in an unreliable design.

TABLE 5.2

DESIGN RELIABILITY CHECK LIST FOR VALVES

		YES	NO
1.	Are the specific design criteria for reliability and safety of each valve class specified?		
2.	Has the required failure rate for various parts been established?		
3.	Are component parts selected to meet reliability requirements?		
4.	Are maintenance requirements and maintainability goals established?		
5.	Have acceptance, qualification, sampling, and reliability assurance tests been established?		
6.	Are standardized high-reliability parts being used?		
7.	Have "below-state-of-art" parts or problems been identified?		
8.	Has shelf life of parts chosen for final design been determined?		
9.	Have limited-life parts been identified, and inspection and replacement requirements specified?		
10.	Have critical parts which require special procurement, testing, and handling been identified?		
11.	Have all vital adjustments been classified as to factory, pre-operational, or operator types?		
12.	Are mechanical support structures adequate?		

Table 5.2 (continued)

		YES	NO
13.	Are heat transfer devices or designs adequate?		
14.	Is there a concentrated effort to make the developmental model as near to the production model as possible?		
15.	Is there provision for improvements to eliminate any design inadequacies observed in engineering tests?		
16.	Have mechanisms of failure for critical parts been identified?		
17.	Has the need for the selection of matching parts been eliminated?		
18.	Can all maintenance be conducted in the field?		
19.	Have prototype designs and random samples been tested with respect to the following conditions?		
	a. Significant number of open-shut cycles (at least 200% of estimated cycles to wear in for non-destructive tests of large valves).		
	b. Flow (system design)		
	c. Pressure (system design) and pressure differential		
	d. Temperature (system design)		
	e. a, b, c, and d above simultaneously.		

Table 5.2 (continued)

	YES	NO
20. Has consideration been given to the valve design to eliminate crud traps which will have an adverse effect on operation and maintenance?		
21. Has consideration been given to surface finish to minimize corrosion?		
22. Has the selection of lubricants considered the effects of radiation on the breakdown of the lubricant?		
23. Has the design considered the environmental extremes to which the valve will be exposed?		

5.2.3 Valve Reliability Data

Quantitative reliability data on valves resulting from field experience in reactor systems is for all practical purposes nonexistent within the present state-of-the-art. As a consequence, the reactor designer must turn to valve reliability experience in other applications and extrapolate the results of this experience to the nuclear reactor enviornment. The accuracy of this extrapolation will depend upon the accuracy of failure mode and effects analyses of nuclear valves and the valves on which the data were based. Unfortunately, valve reliability data are limited in all applications. However, significant data are available to the designer who is willing to spend considerable effort to retrieve it. The following data sources are recommended as starting points.

1. Rome Air Development Center. The Rome Air Development Center (U. S. Air Force, Air Force Systems Command, Research and Development Center, Griffiss Air Force Base, New York) reports "Data Collection for Nonelectronic Reliability Handbook" and "RADC Unanalyzed Nonelectronic Part Failure Rate Data, Volumes I and II" (NEDCC I & II) contain

information on the location, collection, classification, organization, and analysis of the reliability of nonelectronic parts. A final report on this study performed under Air Force Contract AF 30(602)-4242 was scheduled for release in 1968. This report includes failure data for a variety of valve types, associated stress level data, and failure mode distributions. Over 90% of the data resulted from airborne experience, but the remaining naval and ground-based system data can be of some value to reactor designers.

2. United States Navy. The USN Maintenance Data Collection System (MDCS) reports all failures of naval systems. Data may be requested for given parts through the use of an Equipment Identification Code (EIC). Specific data on valves used in USN ships may be obtained through the USN Maintenance Support Office in Mechanicsburg, Pennsylvania. Because this data system is in a formative stage and because of its complexity, more analysis is required after data extraction.

3. FARADA. FARADA is a tri-service and industry data collection program managed at the United States Naval Ordinance Laboratory in Corona, California. It is similar to NEDCO I & II except that less analysis of the raw data is made.

4. Holmes & Narver, Inc. Holmes & Narver, Inc., has studied the operating and safety experience of a number of power reactor plants under contract to the USAEC. Reports prepared under these contracts provide a further source of data on valve reliability.

5. Valve Manufacturers. Many valve manufacturers have reliability data that are available to reactor designers upon request. This data are usually more qualitative than quantitative and are often biased by the manufacturer's viewpoint.

From a long-range standpoint, it may be desirable for system designers to initiate a data collection program. The essential information elements for such a program are

1. data source,
2. valve identification number,
3. manner of valve malfunction,
4. symptoms,
5. effect,

6. operating time or cycles at failure,
7. date of malfunction,
8. disposition of failed valve,
9. shop findings as to cause of malfunction,
10. environment,
11. time to repair, and
12. system down time.

Valve reliability should be expressed in terms of valve failures per unit operating time and valve failures per unit operating cycles.

5.2.4 Valve Test Program

System reliability can be evaluated by using a comparison technique when two systems contain the same type of components. To establish a correlation between component reliability and quality control, it is necessary to establish failure rate data and correlate the data with the quality control requirements to which the component was designed and fabricated. In cases where reliability or failure rate information for a particular valve design is considered essential and where satisfactory information is not available from previous usage of the design under comparable conditions, the only alternative may be to obtain the essential data through a valve testing program.

The contents and basic considerations of a valve test program designed to obtain failure rate data are discussed in the following paragraphs. The types of tests conducted on a prototype for development purposes or on one or two samples for demonstrating basic functional capability are not discussed because each such case must be evaluated individually and generalization should not be attempted. This is not to say that tests conducted with limited numbers of samples cannot provide valuable information related to the reliability of a valve, but the results are heavily dependent upon technical judgment and provide little or no statistical confidence such as can be derived from extensive use of a design or from proper testing of an adequate sampling of the lot.

Test programs to determine the reliability of a particular valve design for a specific application or for a range of applications are invariably expensive and time consuming. Consequently, they usually are considered only after weighing the relative costs of changing the system design to reduce the absolute value of the reliability required and the cost of the test program required to determine whether the required reliability can be achieved for the valve.

Where a test program is desirable, careful and complete pre-planning is essential. Otherwise, the results are likely to be inconclusive because the proper combinations of variables were not considered, the valves tested were not representative of the lot, or the results lack statistical confidence. The planning for a test program should follow a step-by-step procedure to reduce the chances of obtaining inconclusive results. A typical procedure used in planning a valve test program is presented in the following paragraphs. This procedure serves only to highlight some of the more important considerations which should be included in a valve test program and is not intended to be totally comprehensive or restrictive. If the application for which reliability information is needed involves the use of automatic operators, the valve and operator are commonly tested as a unit. Therefore, some reference may be made to the operator in the following sample procedure.

(a) Understanding Purpose of Test. Tests of valves to determine failure rate are conducted to establish the reliability of the valve when subjected to the operational conditions it will experience during the service life of the plant. If it is understood before undertaking the valve test program that design changes will be made if necessary to meet reliability requirements, the test thought to be the most limiting should be conducted first to minimize the time expended between design changes.

(b) Defining the Variables. The first step in the definition of variables is to identify the functional requirements of the valve for which failure rate information is required. Typical functional requirements encompass the automatic features of valve operation, flow characteristics, leakage requirements etc.

The next step should be to define exactly what constitutes a failure of the valve since these are the conditions for which information will be collected. Examples of these conditions include failure to open at at least two-thirds of normal stem travel, leakage across the seat of more than 2 cm^3 per minute, leakage to the environment of more than 1 cm^3 per hour, etc. It may also be necessary to consider combinations of failure conditions wherein the levels will be more stringent than for single failures.

It is then necessary to define the operational variables and their ranges for the system in which the valve is to be used. It is advisable to take advantage of the experience of individuals familiar with the design, fabrication, operation, and maintenance of both the valve and the system when defining these variables. Typical operational variables are flowing fluid composition, flow rate, temperature, pressure, contaminants, cavitation characteristics, corrosion and erosion conditions, thermal and mechanical shock, operating torques or forces, rates of changes of all variables, etc.

There usually are numerous other variables that may affect the reliability of the valve during its lifetime. These include external environmental variables, installation variables, and maintenance variables. The external environmental variables may include such factors as valve orientation, forces exerted through nozzles, ambient constituents, ambient temperature and pressure, vibration, and the rates of change of all variables. Such installation variables as rigging forces, impact forces from dropping or forcing into position, heat from welding, and misalignment can often be important. Cleaning, painting, lubricating, repacking, adjusting, and the frequency of these operations are maintenance variables of interest.

(c) Designing the Test. Once the variables are fully defined, design of the test can be started. The first step in designing the test is usually to determine the amount of time available before the results are required. An accelerated test is often necessary because the design life of a nuclear plant is usually in the order of 40 years and acceptable reliability is commonly in the order of one failure per tens or hundreds of thousands of valve years. The accelerated test for many of the variables, such as corrosion, means an increase in the severity of the conditions. For other variables, such as wear from cyclic operation, the accelerated test means an increase in the rate of cycling. For still other variables, such as installation damage, the accelerated test may have little or no effect. Where compression of time does require an increase in the severity of one or more conditions, a judgment factor is introduced into the tests. It is common practice to scale the severity rate on the basis of the best information available from the failure mode and effects analysis and then double or triple it to avoid challenge of the judgment.

When the test time limitations are understood, the next step in designing the test is to select the variables that will be duplicated by the test. This can be done rather readily by using the lists of variables previously discussed, but some judgment may be necessary if consideration is given to elimination of the less significant variables. A convenient method for selecting variables is to prepare a table in which each variable, its actual range, frequency, and period where applicable are listed along with the proposed test range, frequency, and period. Whether the effects of the variables are regarded as being significantly cumulative should be indicated in another column of the table. Variables with cumulative effects are those contributing to corrosion, erosion, wear, and fatigue. Variables which are usually limited to a finite number of cycles, such as hydrostatic pressure, mechanical shock, thermal shock, and moments exerted through the nozzles, must also be considered carefully. Another column can be used to list variables with complementary effects or effects that contribute to the same type of weakening or failure of the valve. An example is failure of the valve body caused by the combined effects of corrosion, erosion, and hydraulic shock.

For the more complicated cases, a failure mode prediction chart is also helpful at this point in the design of the test. The previously defined valve functional failures are listed down one side and the test variables are listed down the other side of a typical failure mode prediction chart. Working from the functional failures, the various failure modes are developed. The test variables can then be more easily associated with the actual failure modes, and the chart will aid in showing which variables have cumulative or complementary effects. A simplified version of such a chart is given in Table 5.3.

TABLE 5.3. FAILURE MODE PREDICTION CHART (Typical Example)

VALVE FUNCTIONAL FAILURES	COMPONENT PARTS INVOLVED IN FAILURE	PREDICTABLE POTENTIAL FAILURE MODES	TEST VARIABLES* WHICH COMPLEMENT FAILURE	TEST VARIABLES WITH CUMU– LATIVE EFFECTS
Leakage across seat greater than 10 cc/min. when closed.	Seat	Scored	A, E	A, E
		Warped	D, I, J	
		Galled	E	E
		Eroded	A, J	A, J
		Pitted	A	A
		Dislodged from body	D, E, H, I, J	D, E
	Disc	Scored	A, E	A, E
		Warped	D	D
		Galled	E	E
		Eroded	A, J	A, J
		Pitted	A	A
		Dislodged from stem	D, E, H, J	D, E
		Misaligned (frozen to stem)	A, D, E, H, J	A, D, E, J
Leakage to environment greater 100 cc/ day	Packing	Hardened & cracked	A, B	A, B
		Porous	H	
		Loose around stem	E, F	E, F
		Loose against bonnet	F	F
		Extruded by fluid	C, D, E, F	C, D, E, F
	Stem	Bent	D, E, H	
		Scored	E, F	E, F
	Bonnet	Packing surface scored	F, H	F
		Gasket surface scored	H	
		Gasket surface warped	H	
		Cracked	C, D, H	
		Porous	H	

Table 5.3 (continued)

VALVE FUNCTIONAL FAILURES	COMPONENT PARTS INVOLVED IN FAILURE	PREDICTABLE POTENTIAL FAILURE MODES	TEST VARIABLES* WHICH COMPLEMENT FAILURE	TEST VARIABLES WITH CUMU- LATIVE EFFECTS
	Bonnet gasket	Insufficiently compressed		
		Deteriorated	A	A
		Misaligned	H	
		Extruded	C, D	C, D
	Body	Gasket surface scored	H	
		Gasket surface warped	H, I	
		Cracked	C, D	
		Porous	H	
	Nozzles & welds	Cracked	C, D, H, I, J	
		Porous	H, I	
Stem rotation torque greater 20 ft-lb.	Stem	Galled	E, F, G	E, F, G
		Rusted	A	A
		Worn	D, E, G	D, E, G
		Bent	D, E, H	D, E, H
	Yoke bushing	Worn	E, G	E, G
		Galled	E, G	E, G
		Misaligned	D, E, H	
		Cracked	D, E, H	
		Dislodged	D, E, H	D, E
	Packing gland	Misaligned	E, F, H	
		Galled	E, F	E, F
		Cracked	E, F, H	
		Bent	E, F, H	
	Packing material	Overcompressed	F	
		Deteriorated	A, B	A, B
	Bonnet bushing	Worn	A, E	A, E
		Galled	E	E
		Misaligned	D, E, H	
		Cracked	D, E, H	
		Dislodged	D, E, H	D, E

*Hypothetical Test Variables for Failure Mode Prediction Chart

A. Flowing Fluid: water with 1000 ± 20 PPM CaOH and 100 ± 20 PPM of 40-100 mesh silica sand.

B. Fluid Temperature: constant at $70 \pm 2^{\circ}$F.

C. Fluid Pressure: total head at inlet nozzle 200 ± 10 psia except during induced hydraulic shock.

D. Hydraulic Shock: induced at rate of one cycle per minute so as to raise pressure at valve inlet nozzle to 1000 ± 100 psia when wave front passes with valve closed.

E. Valve Cycling: valve closed every 10 ± 1 minutes for 1 ± 0.5 minute period at rate of one handwheel rotation per 2 ± 1 seconds with seating torque varied in 10 ft-lb increments between 40 and 100 ft-lb on successive closings.

F. Packing Gland Adjustment: tightened to 5 ft-lb on nuts every 100 valve cycles.

G. Stem Thread Lubrication: 5 drops of SAE 90 gear oil every 100 valve cycles.

H. Sample Selection: 20 valves randomly selected from lot of 5,000 valves.

I. Installation: each sample to be installed in test rig by a different qualified welder.

J. Orientation: at least one valve in each 90° orientation in both horizontal and vertical piping with flow in each direction in vertical pipe.

If the valve design is subject to change based on the test results, it is often desirable to make the first part of the test a destructive test. This will consist of exposing the valve to the most severe conditions of pressure, temperature, mechanical shock, etc., to determine whether it can survive these conditions before proceeding with the cumulative variable tests, which are more time consuming.

Planning the time-dependent portion of the test is usually more difficult because it is desirable to have all variables which could cause each failure mode reach their most destructive condition at the same time. A method for doing this can be recognized in most cases after study of the previously discussed lists and tables. If the situation is quite complicated, it may be necessary to develop an analytical scheme to assist in planning the test. This necessarily must be left to the discretion of the individual designing the test. At this point, the sequence of events to which a single sample will be exposed and the type of apparatus that will be used to control the appropriate test variables should be described by the design for the test.

The next question to be answered concerns the number of valve samples to be tested. Statistical methods are usually necessary to assure that the test results will be within the confidence limits desired, but several decisions must be made before the statistician can start to work. One decision is whether the results must show the actual failure distribution or whether it will be sufficient to indicate that at least a certain percentage of the valves will not fail before a given service life. If the actual failure distribution is necessary, the majority of the test samples must be tested to failure. If actual failure distribution is not required, it is not necessary that any of the valves fail. Economics of valve testing make it desirable to run tests only as long as required to demonstrate the desired reliability. One common test design involves a combination of these two conditions by testing at simulated operating conditions until a certain reliability is demonstrated and then increasing the severity of the conditions progressively until all or most valves fail. Although the failure distribution obtained is meaningless, the knowledge gained about the failure modes is often useful for either design improvements or for demonstrating safety margins.

The statistician must also be given the degree of confidence required for the test results. Another way of stating this is the probability that a randomly selected valve from the same lot will have a certain minimum life before failure or will fail between certain lifetimes.

The number of samples may be further dependent upon the potential variables in the lot to be sampled. An example would be a situation where valves in the lot were produced by different vendors or were made from different lots or materials. In such cases, the number of samples may depend upon the quality control programs and other variables that would distort random sampling techniques.

Occasionally, it is advantageous to design the test in parallel parts if some of the variables do not contribute in any way to the types of failures evaluated in another part of the test. This arrangement will require duplication of sampling. At this point, the essential information concerning test conditions, times, number of samples, etc., is sufficiently defined to permit writing the test program in detail.

(d) Special Considerations. Adequate instrumentation of the test rig is necessary to assure that the variables are being consistently exercised within their intended ranges and at the desired rate of change. It may also be desirable to instrument the test valves with strain gages, torque meters, etc., to better understand sequences and modes of failure. Consideration should be given to the use of recording instruments both as evidence of the testing and to permit more detailed analyses of the proceedings during and after the test.

Safety considerations are obviously very important, especially when testing for certain types of valve failure. Failure of the test apparatus must also be considered. Experience indicates that modification of the test procedure or the equipment is often necessary or desirable after only a short period of testing. The test design should therefore be made as flexible as possible, and some scheduled time should be allowed for such changes.

(e) Conducting the Test. Performance of the testing is relatively straightforward, but a few words of caution are in order. For long test durations, the monetary investment near the end of the test may be increased significantly. This should be considered when selecting personnel to operate the test and handle the data obtained to assure the protection of equipment and the remaining samples from destruction by unanticipated modes of sample failure as well as external detriment. In-process analysis of data is usually desirable to assure that the test is proceeding in a manner that will provide the desired results with the planned confidence levels.

(f) Evaluation of Test Results. The data from the test will be handled in much the same manner as was the data from actual valve applications. If the results should be slightly below the minimum requirements set forth, consideration should be given to additional sampling and testing to further evaluate the statistics involved before making design changes to the valve that would necessitate complete retesting. Care should be taken to see that the results of a test made for a specific application are not erroneously extrapolated to other applications with significantly different variables.

For tests that do not proceed to destruction, it is wise to retain the samples in view of the possibility of using them in additional tests where only non-cumulative variables are changed and the samples can be readily retested with consideration for the new variables.

In summary, the salient points to be kept in mind when designing tests to determine the reliability of valves are that

1. tests to determine specific reliability data are invariably expensive and time-consuming;
2. the accuracy of test results is dependent upon the concise and complete identification and duplication of all significant variables;
3. statistical methods are necessary in designing the test, determining the number of samples to be tested, selecting the samples, and evaluating the test results; and
4. tests rarely proceed in accordance with the original plans and scheduled time should be allotted for potential changes.

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GLOSSARY OF TERMS

acme: a truncated thread intended primarily for translating screws or traversing motion.

actuator: a device to convert control energy into mechanical motion.

adsorption: a state of physical adherence of molecules, such as O_2, to a metal surface. It is similar to chemisorption but is characterized by having a lower film energy, not qualifying as a chemical reaction.

anode: the electrode of an electrolytic cell at which oxidation occurs. In corrosion processes, it is an area that is usually corroded.

asperities: high points or irregularities that exist even on finely finsihed surfaces. These play a major role in friction and wear.

backseat: valve seat within the bonnet to prevent leakage through the stem when the valve is fully open.

bellows: a device to seal a pressure boundary by the use of a thin, flexible membrane through which motion can be transmitted.

body: main part of a valve that provides the structural housing for trim, principal pressure boundary for the line fluid, and connections to the piping.

bonnet: part of valve forming housing for retracted disc in the open position, a support for the operator and stem, and a pressure boundary at the stem seal.

bonnetless valve: a valve designed with the stuffing box as a part of the valve body, and the gland flange is fastened to the body.

buildup, crevice corrosion: accelerated localized attack observed at the perimeter of a contact area resulting from crevice corrosion.

canopy ring: a ring used to seal weld the bonnet-to-body closure.

cathode: the electrode of an electrolytic cell at which reduction occurs. In corrosion processes, it is the area that is usually not corroded.

carbon steel: steels which do not contain alloying elements and whose properties are determined by their carbon content.

caustic embrittlement: stress corrosion cracking of carbon steel which is observed in boiler waters treated with alkali.

cc/kg: cubic centimeters of gas per kilogram of water (STP).

chemisorption: a metastable state in the oxidation of a metal surface wherein the oxygen ion is attached to the surface but the metal ion has not left its location in the lattice.

choking: occurs in valves when local flow velocity at the greatest flow restriction approaches the speed of sound. The flow and mass transfer through the valve reach a maximum value and cannot be further increased.

Colmonoy: a nickel-base alloy used as a seat material.

contact area: area of actual contact between valve seat and disc.

contact corrosion: see buildup, crevice corrosion.

corrosion: the transfer of atoms in a metal from the metallic to the ionic state. In aqueous solutions, it involves the formation of soluble and insoluble compounds of the metal being corroded.

corrosion fatigue: accelerated localized corrosive attack which causes cracking to occur in areas where a metal is subjected to cyclic stresses.

creep: plastic deformation over a period of time of a metal subject to steady stress lower than yield. This phenomenon usually occurs at elevated temperatures.

crevice corrosion: crevice corrosion occurs when stagnate areas of solution (as in crevices) develop differences in oxygen concentration, setting up metal-ion concentration cells. These cells cause corrosion of the metal where the ion concentration is low.

crud: the insoluble corrosion products formed in the primary coolant of water-cooled nuclear reactor plants.

crud level: the quality of crud per unit volume of the primary coolant. It is normally expressed as parts per million (ppm).

decontamination (for radioactivity): the process employed to reduce the activity resulting from radioactive deposits in components or a system to a tolerable level. Mechanical and chemical cleaning methods are employed.

degassed water: low oxygen-bearing water employed for corrosion tests to simulate low oxygen conditions in the reactor system. The oxygen content for degassed water is normally 0.05 to 0.1 cc/kg.

demineralized water: high-purity water obtained by the use of ion exchanges.

design reliability check list: a series of questions that is used to identify many qualitative factors involved in a relaibility evaluation.

disc: moving seal piece in a valve that stops or throttles the flow.

disc guide: a part used to align the disc with the valve seat to insure even contact.

double packing: two sets of packing separated by a packing leak-off ring (lantern ring).

erosion-corrosion: the loss of metal resulting from the combined action of corrosion and erosion.

failure mode and effects analysis: a procedure for systematically identifying potential failures and analyzing their effects with respect to the application.

fasteners: any part used to hold structural parts of a valve together.

feedwater: term employed in boiler water technology to signify the relatively pure water which is condensed from the steam in the turbine and fed back to the main system.

fretting corrosion: accelerated attack of metal that occurs at the interface between two surfaces in contact and is directly attributable to the presence of vibrations which produce slip between adjacent surfaces.

galvanic corrosion: accelerated corrosive attack occurring at the point of contact between two dissimilar metals that results from the difference in electrical potentials produced by the two metals in contact.

general corrosion: the uniform loss of metal as a result of corrosive attack.

gland: the moving part of a stuffing box by which the packing is compressed; a device for preventing leakage of fluid past a joint.

gland flange: that part of the packing gland that is engaged by bolts and is torqued down by nuts.

gland follower: bushing which is forced down between the stem and stuffing box by the gland flange, thereby compressing the packing.

hard-facing: materials incorporated in valve trim parts for their hardness and wear resistance.

hermetically sealed: sealed so that no leakage of primary fluid to the ambient is permitted. All closures in the fluid pressure boundaries must be seal welded.

high-pressure flow throttling: situation where a throttle valve is required to throttle flow at very low flow rates and very high differential pressures for long periods of time.

high-purity water: arbitrarily defined as water with a minimum resistivity of 500,000 ohms-cm and a total solids content of less than 2 ppm and a pH ranging between 6 and 8.

hydrogen embrittlement: loss in ductility of a metal or alloy resulting from the absorption of nascent hydrogen.

hydrogenated water: test water used to determine the effect of dissolved hydrogen on corrosion and wear in a simulated reactor environment. This water is prepared by adding known amounts of hydrogen to degassed water.

intercrystalline cracking: see intergranular cracking.

intergranular cracking: cracking which occurs at the boundaries between the grains of a metal.

intergranular corrosion: a special form of localized corrosion in which accelerated corrosive attack occurs only at the grain boundary of a metal.

lantern ring: metal grooved ring separating the primary and secondary packing in a double-packed valve with leak-off.

lip seal: a welded bonnet-to-body joint seal for a hermetic seal in addition to the bolted joint.

localized corrosion: nonuniform loss of metal as a result of corrosive attack, such as pitting.

locking device: a device designed to prevent fasteners or parts thereof from loosening; a mechanical obstruction to motion of the fastener in the loosening direction.

low alloy steel: steels which possess special properties that are developed by the addition of alloying elements; for example, less than about 3% of alloying materials such as Cr and Mo.

low-oxygen water: test water employed to simulate conditions where low oxygen-bearing water was expected in service. Normally range between 0.1 and 1.0 cc/kg.

MDM: term employed to represent the corrosion rate of a material, milligrams of metal corroded per square decimeter per month.

oxygenated water: test water employed to simulate conditions where high oxygen-bearing water was expected in service. Normal range between 1 and 30 cc/kg.

packing: a dynamic ring seal of self-lubricated compressible material used to seal a moving shaft.

packing stop ring: a metal ring at the bottom of a stuffing box to prevent the packing from extruding between the shaft and journal.

packing leak-off ring: see lantern ring.

passivity: the property of metals enabling them to generate a thin tenacious film on the surface, thereby retarding general corrosion.

polarization: the extent of potential change caused by net current to or from an electrode measured in volts.

precipitation-hardening stainless steel: stainless steel which is hardened by a constituent precipitating from an alloy during a specific heat treatment.

pressure boundary parts: those elements of the valve which are stressed by the pressure of the contained fluid; normally those components restraining the fluid.

pressure seal bonnet joint: a valve bonnet-to-body joint that incorporates a soft metal ring which is sealed by internal system pressure making a self-energized seal.

pressurized-water reactor: term describing a water-cooled nuclear reactor in which the water in the reactor is pressurized to prevent boiling.

seat: the fixed seal area in a valve.

seat angle: the included angle between the seat and disc that increases contact pressure by the mechanical advantage of the wedge design.

seat bridgewall: structural section extending from the inner walls of the body to support the valve seat.

seat diameter: the smallest diameter at which the disc touches the valve seat.

sensitization (chromium carbide): exposure of an unstabilized austenitic stainless steel to temperatures which will cause the precipitation of chromium carbide at the grain boundaries. The material is thus rendered susceptible to intergranular attack in certain environments.

sensitizing: the susceptibility of stainless steels to intergranular corrosion when held at temperatures of between 900 to 1500°F caused by the depletion of chromium content along the grain boundaries.

squirm (bellows): deflection of a bellows trying to move off its axial center line in a sideward direction.

stainless steels: stainless steels are divided into three broad categories: the austenitic, ferritic, and martensitic types. The austenitic stainless steels generally refer to the 18 chromium-8 nickel type such as ASTM Type 304 or 347. The ferritic stainless steels generally contain 12% or more chromium and are not hardenable by heat treatment, such as the ASTM Type 405 or 430. The martensitic stainless steels generally contain 12% or more chromium and are hardenable by heat treatment, such as the ASTM Type 410 or 420.

stem: the mechanical linkage that translates external mechanical operation to opening and closing of the valve.

stem bushing: the part mounted on the yoke of the valve that mates with the stem thread (same as yoke sleeve).

stellites: cobalt alloys having good hardness and wear resistance characteristics.

strain ratcheting: a progressive distortion resulting from certain types of cyclic or steady-state plus cyclic loading in which plastic flow occurs, tending to relieve loading, followed by unloading strain, reversal, and reloading, causing further progressive flow.

stress corrosion: accelerated localized corrosive attack that causes cracking to occur in highly stressed areas of a metal.

stuffing box: the annulus between the valve stem and the valve that accommodates the packing stop ring, packing, lantern ring (if used), and the gland follower.

super hardening: a form of cold working applied only to the surface of a material when compressive deformation takes place in hard materials.

swivel disc: disc-stem connection that allows the stem to turn independently of the disc, preventing disc rotation while seating plus correcting any slight stem-seat misalignment.

throttle valve: a valve which obstructs the flow of fluid through it.

throttling linearity: for each increment of stem travel there results a corresponding percentage increment of flow increase.

transcrystalline cracking: see transgranular cracking.

transgranular cracking: cracking that occurs through the grains of a metal.

transported corrosion products: insoluble corrosion products deposited at certain locations other than those where the products were initially formed.

validity factors: method for representing the overall significance of corrosion rates given in handbook. This factor represents the weighted importance of consistency of results, number of samples, and duration of test.

valve trim: internal parts of valve wetted by the system flow other than the body and bonnet; normally includes seats, stems, discs, and other miscellaneous hardware such as guides and fasteners.

vena contracta: fluid contracts upon entering a reduced flow area and because of momentum, the minimum flow area occurs beyond the entrance to the reduced flow area.

wear (wear resistance): wear is a complex surface phenomenon resulting in mechanical attrition of moving surfaces in contact, as by welding and removal of particles through friction. Wear is a generic property influenced by other properties such as hardness, corrosion resistance, temperature effects, mechanical considerations, and the environment.

wear factor: a numerical method of ranking materials by wear resistance; the factor varies directly with wear.

yoke sleeve: see stem bushing.

zero leakage: a measurable quantitative rejection leakage value, expressed in atm cc/sec, above which the part is rejected and below which any smaller leakage would be insignificant in the operation of the leak detector components. The sensitivity of various leak detection instruments in determining zero leakage is as follows.*

sonic detectors	-10^{-1} atm cc/sec
bubble testing	
by soap solution	-10^{-4} atm cc/sec
by immersion	-10^{-4}, -10^{-5} atm cc/sec
halide torch	-10^{-5} atm cc/sec
argon leak detector	-10^{-9} atm cc/sec
hydrogen sensitive	
leak detector	-10^{-8} atm cc/sec
halogen leak detector	-10^{-6}, -10^{-9} atm cc/sec
helium leak detector	-10^{-10}, -10^{-14} atm cc/sec
leak detection by	
radioactivity	-10^{-12} atm cc/sec

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