Manual of Water Supply Practices

M49

Quarter-Turn Valves:

Head Loss, Torque, and Cavitation Analysis

Third Edition

American Water Works Association

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Production: Studio Text
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Preface

The purpose of this manual is to present a recommended method for calculating operating torque, head loss, and cavitation for quarter-turn valves typically used in water works service. It is a discussion of recommended practice, not an American Water Works Association (AWWA) standard. The text provides guidance on generally available methods for using quarter-turn valves as well as their cavitation, flow, and torque characteristics. Questions about specific situations or applicability of specific valves and values should be directed to the manufacturers or suppliers. Information in this manual is useful for technicians and engineers who want a basic understanding of the calculations associated with the use and specification of quarter-turn valves. The valve torque, flow, and cavitation coefficients given are typical but generic values covering a variety of products. Actual flow, cavitation, or torque coefficients for a particular manufacturer's valve should be used in calculations for a specific valve and application to obtain the highest calculation accuracy.

The history of this manual is related to that of American National Standards Institute ANSI/AWWA C504, Standard for Rubber-Seated Butterfly Valves. Until the 1994 edition, ANSI/AWWA C504 included Appendix A, which described a recommended method of calculating torques for butterfly valves. This appendix was deleted from the 1994 and subsequent editions of the standard for several reasons. The AWWA Standards Council directed that standards documents should not contain appendixes; appendix text should either be moved to the main body of the standard or be made into a separate, stand-alone document. Members of the committee for ANSI/AWWA C504 at the time were concerned that the existing text of Appendix A no longer represented the current state of knowledge concerning methods for calculating torques for butterfly valves. In 1993, a subcommittee was established to rewrite Appendix A as a separate manual incorporating the state-of-the-art theory for calculating torque and head loss values for butterfly valves. The second edition of the manual expanded the introduction and some equations, added torque sign conventions, added double-offset disc design variables and calculations, added equations for eccentricity torque, added metric units and equivalents, consolidated the nomenclature, and corrected some errors. This third edition manual broadens the application of these methods to include other quarter-turn valves such as ball, plug, and rotary cone valves.

Manual M49 refers to AWWA standards available for purchase from the AWWA Bookstore. Manufacturers graciously provided valve illustrations and other documentation. AWWA does not endorse any manufacturer's products, and the names of the manufacturers have been removed from the material provided.

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Acknowledgments

The AWWA Standards Council Committee for Butterfly Valves developed this manual with input from the Ball & Plug Valve Committees. The members listed below served on the Butterfly Valve Committee at the time the manual was approved. .

M. MacConnell, Chair, Metro Vancouver, Canada

User Members

S. Carpenter, San Diego County Water Authority, Escondido, Calif.
D.W. Coppes, Massachusetts Water Resources Authority, Chelsea, Mass.
S. Hattan, Tarrant Regional Water Dist., Ft. Worth, Texas
V.Q. Le, Los Angeles Water & Power, Los Angeles, Calif.
M. MacConnell, Metro Vancouver, Burnaby, BC, Canada
P.J. Ries, Denver Water, Denver, Colo.
S.Y. Tung, City of Houston, Houston, Texas

General Interest Members

A. Ali, ADA Consulting Ltd., Surrey, B.C.
M.D. Bennett, MWH, Cleveland, Ohio
B.E. Bosserman, Mission Viejo, Calif.
*J. W. Green,** Lockwood, Andrews & Newnam, Inc., Oakbrook Terrace, Ill.
J. Hebenstreit, UL LLC, Northbrook, Ill.
F.L. Hinker, Santa Rosa, N.M.
M.C. Johnson, Utah State University, Water Research Lab, Logan, Utah
T. Jordan, HDR Inc., Denver, Colo.
T.J. McCandless,† American Water Works Association, Denver, Colo.
*D. Middleton,** AECOM, Wakefield, Mass.
J.N. Nakashima, Lockwood, Andrews & Newnam, Inc., Oakbrook Terrace, Ill.
*W. Rahmeyer,** Utah State University, Water Research Lab, Logan, Utah
U. Sant, AECOM, Dallas, Texas
R.A. Ward, Tighe & Bond, Westfield, Mass.

Producer Members

A. Abouelleil, Henry Pratt Company, Aurora, Ill.
S. Allen, Bray Valves, Jonesboro, Ariz.
J.V. Ballun, Val-Matic Valve & Manufacturing Corp., Elmhurst, Ill.
*K.R. Graeff,** Rodney Hunt Company, Orange, Mass.
T.A. Hartman, Hartman Valve Corp., St. Louis, Mo.

---

* Alternate
† Liaison

H. Herold, VAG, Mannheim, Germany

*J.R. Holstrom,** Sub-committee chair, Val-Matic Valve & Manufacturing Corp., Elmhurst, Ill.

K. Johnson, M&H Valve Company, Anniston, Ala.

A.W. Libke, DeZURIK, Sartell, Minn.

*R. Tschida,** DeZURIK, Sartell, Minn.

J.H. Wilber, American AVK, Littleton, Colo.

*D. Woollums,** Mueller Group, Chattanooga, Tenn.

The AWWA Standards Council Subcommittee for Addition of Ball and Plug Valves assisted in the development of this manual. The members listed below served on the subcommittee at the time the manual was approved.

John R. Holstrom, Subcommittee Chair, Elmhurst, Ill.

A. Abouelleil, Henry Pratt Company, Aurora, Ill.

S.A. Bach,* American Water Works Association, Denver, Colo.

D.E. Burczynski, Kennedy Valve, Elmira, N.Y.

S. Hattan, Tarrant Regional Water District, Ft. Worth, Texas

H. Herold, VAG, Mannheim, Germany

F.L. Hinker, Santa Rosa, N.M.

J.R. Holstrom, Val-Matic Valve & Manufacturing Corp., Elmhurst, Ill.

K. Johnson, M&H Valve Company, Anniston, Ala.

M.C. Johnson, Utah State University, Water Research Lab., Logan, Utah

R. Kilborn, Town of Gilbert NWTP, Gilbert, Ariz.

A.W. Libke, DeZURIK-APCO, Sartell, Minn.

R. Looney, American AVK Company, Minden, Nev.

J.D. Martin, American R&D, Anniston, Ala.

S. Notch, Gilbert, Ariz.

T. O'Shea, DeZURIK-APCO, Willamette Division, Schaumburg, Ill.

J.E. Pearman, Ross Valve MFG. Company, Winchendon, Mass.

W.H. Peffley, Crawford, Murphy & Tilly, Springfield, Ill.

P. Savalia, Henry Pratt Company, Aurora, Ill.

G.E. Slaughter, Victaulic Company, Olathe, Kan.

M. Valentine,* American Water Works Association, Denver, Colo.

J.H. Wilber, American AVK, Littleton, Colo.

* Staff

Chapter 1

Introduction

Head loss, torque, and cavitation are important considerations in the selection and sizing of quarter-turn valves in water systems. Quarter-turn valve components must be able to withstand the forces and torques generated during use, and the actuator must drive and seat the valve. The head loss developed across any valve adds to the energy costs of a pumping system. Cavitation can damage a valve or adjacent piping if not controlled.

The topics in this introductory chapter include an explanation of basic quarter-turn valve design elements and their role in predicting operating head loss, torque, and cavitation. Prior editions limited this manual of standard practice to butterfly valves (BFV). This edition has been expanded to include information on other quarter-turn valves, including ball (BV), rotary cone (RCV), and eccentric plug valves (PV). Many of the included illustrations are targeted toward BFVs but are generally applicable to all the valves of this scope.

Head loss characteristics must be known to predict valve operating torque, and system designers also use these data to size a control valve, calculate pump head requirements, and evaluate the energy costs associated with the head loss across the valve in pumping applications. Valve torque is calculated to allow proper actuator sizing and to provide assurance that the valve components can withstand the internal forces produced by the fluid flow and pressure.

Cavitation is analyzed to avoid undesirable sound and vibration and to prevent damage to the valve and adjacent piping. Cavitation data are determined by flow testing. Values for the range of valve angles are helpful in predicting if cavitation will occur in a given application.

Head loss, torque, and cavitation vary with a valve's position (angle of opening). These characteristics also depend on the geometry of the valve body and closure member as well as the characteristics of the system in which the valve is installed. Flow testing procedures of a valve requires a smooth, undisturbed flow upstream and downstream of the valve, such as that produced by a run of straight, constant-diameter pipe. Although variation from this ideal condition has an effect on valve head loss and torque, these conditions are the benchmark and basis for analysis. Flow disturbances caused by piping configuration, such as elbows, reducers, or other valves within a distance less than eight times the diameter upstream of the valve, may require further review by applying the recommendations given in chapter 6.

Coefficients provided by the quarter-turn valve manufacturer may be used to calculate the head loss and torque as described in this manual of standard practice, provided that the data are determined on the basis of testing methods described in chapter 5. The typical coefficients provided in this manual are presented only for illustrative and approximation purposes. Information from the valve test data or the manufacturer is needed before calculations can be performed for a specific valve in a specific use with high accuracy. However, generalized or typical information will assist in determining the applicability or sensitivity of some characteristics for valve type selection and for most system design considerations.

The closure members of this manual of standard practice are typically referred to as the ball, disc (BFV), cone, or plug. This manual of standard practice may refer to a general closure member or to one specific design. International and European standards will also use the term *obturator* for the closure member.

SCOPE

The fluid flow and torque calculations are based on water or wastewater flow and do not specifically relate to other liquids or gases. The adjustments for application to other fluids can be found in other texts on fluid mechanics. This manual of standard practice covers round or circular BVs and BFVs within the scopes of AWWA and American National Standards Institute (ANSI) standards ANSI/AWWA C504-15, ANSI/AWWA C507-15, and ANSI/AWWA C516-14 with essentially full-ported designs in which the port diameter and closure member diameter are close to the nominal pipe size (NPS) or nominal diameter (in inches or millimeters). This includes BFVs in sizes 3 in. (75 mm) and larger and BVs in sizes 6 in. (150 mm) through 60 in. (1,500 mm).

This manual of standard practice also covers PVs that have round or oblong ports and are available with either full or reduced port areas within the scope of ANSI/AWWA C517-09. Reduced port areas are generally greater than 75 percent of full pipe area.

Rotary cone valves in sizes 6 in. thru 84 in. and pressure ratings of 125 cold working pressure (CWP) or 275 CWP in cast- or ductile-iron construction or ANSI Classes 150 and 300 in steel construction are often used in this industry and referenced in other AWWA manuals of standard practices, such as M44. This valve type does not have an AWWA standard devoted to design and construction. This type of valve is also included in this manual of standard practice.

Some manufacturers produce valves that are configured as three-way and/or four-way valves, which have three or four connection ports and require special considerations not included in this manual of standard practice. The valves covered are of the two-way (two end connections, on-off or throttling) configuration. For all of these valves, it is important to use the matching data for the valve design of interest.

NEW DEFINITIONS, MRST AND AST

For purposes of clarity and understanding, many of the AWWA quarter-turn valve standards are now referring to the operating torque requirements of the valves as two different terms. These are actuator sizing torque (AST) and minimum required shaft torque (MRST), and their definitions appear later in this chapter. These are not to be considered as single values but a series of values (or curves) that vary with valve position. In some cases, one or two (break and/or break and run) conservative or bounding values may be used throughout the entire valve stroke, but in many cases, values at 10°, 5°, or fewer degree increments of valve travel are necessary. The torque predictions of this manual of standard practice provide the most probable operating torque requirements for a valve when operated under the system conditions analyzed. This total operating torque is referred to as the MRST. Depending on the valve type, actuator standard, or manual of standard practice and the valve's application (on-off or modulating), the MRST is multiplied by an application factor (AF) to obtain an AST (AST = MRST × AF). This is also calculated at many valve positions to correctly size the actuator. See the valve or actuator standards for the application factors to be used.

The actuator sizing additional torque margin, allowances for in-service degradation, and/or safety factors for power (i.e., electric motor, cylinder, or vane) actuators are provided in other ANSI/AWWA standards and included in the AFs and other sizing requirements of the product standard.

DIAMETER ASSUMPTIONS

For the valve shaft diameter, valves meeting ANSI/AWWA C504-15 have the minimum shaft diameters given in the standard. ANSI/AWWA C516-14, ANSI/AWWA C507-15, and ANSI/AWWA C517-09 do not provide minimum shaft diameters. It is always best to obtain the shaft diameter by measurement or from the manufacturer's documentation.

Many sources are available for quarter-turn valve flow and torque coefficients. These include valve engineering handbooks; published research papers; and valve supplier manuals, catalogues, or bulletins. The manufacturer generally publishes flow coefficients (i.e., C_v, C_{vm}, or K) for most valves. Some manufacturers consider the torque coefficients (C_t) to be proprietary information and may not publish these data.

Much existing data were developed before published standardization methods, and investigators may have based their calculations on different valve diameter measurements. The major valve diameters include NPS, approach pipe inside diameter, valve port diameter, valve seat diameter, and valve closure member diameter (see Figure 1-1). Also, various publications use slight variations of these first-principles equations or use different units of measure. The user is cautioned to evaluate and convert such data to the proper format and units of measure. For instance, some BFV manufacturers provide a dynamic torque coefficient for use in the formula, $T_d = C_1 \times \Delta P$. When equated to the basic formula used herein, $T_d = C_t \times D^3 \times \Delta P$, it follows that $C_1 = C_t \times D^3$ or $C_t = C_1/D^3$.

If the data were developed on the basis of a BFV disc diameter and the prediction calculations used the nominal diameter, there will be a larger uncertainty in the results than if the disc diameter were used. This manual of standard practice gives direction on what diameter should be used for standardization, consistency, uncertainty, and/or conservatism purposes. However, for many good engineering reasons, much of the older data does not conform to these guidelines. In many instances, the exact approach pipe inside diameter, valve port diameter, and/or valve closure member diameter are not known at the time the calculation is performed. This forces the designer to assume a conservative diameter with greater uncertainty in the results.

For the valves within the scope of this manual of standard practice, the approach pipe inside diameter, valve port diameter, and valve closure member diameter are almost always equal to or less than the valve's nominal diameter when using US customary dimensions. Therefore, the use of the nominal pipe size (NPS) diameter as the diameter in torque prediction calculations will often provide a conservatively high torque value (as the diameter appears in the numerator of the equations). The nominal diameter of the valve may be used in these prediction calculations in lieu of the approach pipe inside diameter, valve port diameter, or valve disc diameter as specified with the understanding that the torque results have a higher uncertainty and are generally greater than a more precise evaluation. In all cases, if the diameter basis on which the data are based is known, the use of the same variable provides the highest-accuracy prediction.

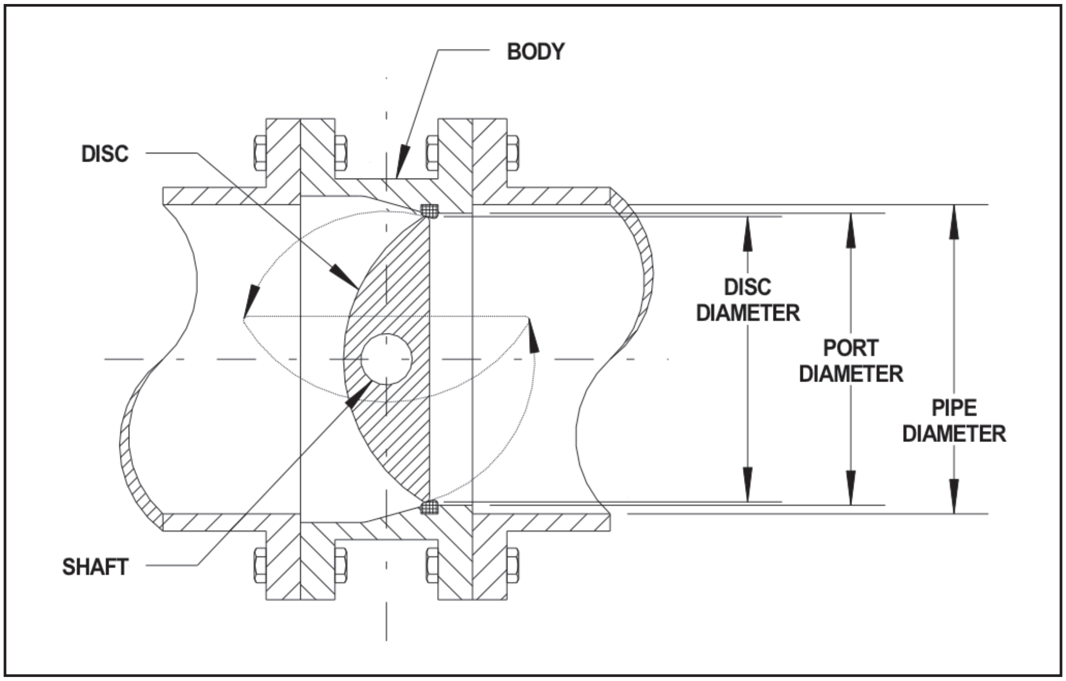


Figure 1-1 Valve disc, port, and pipe diameters

The flow coefficients, C_v and K, and testing and data collection methods that follow are those prescribed in the International Society of Automation (ISA) standard ANSI/ISA S75.02.01-2008, and are based on the test pipe inside diameter. This methodology does use two slight variations from the ANSI/ISA S75.02.01-2008 in that this practice subtracts the piping loss from the test data to obtain net (valve-only) coefficient values versus the gross (as measured, including pipe loss) values and the valve shaft axis orientation during the test. See chapter 5 for more detail.

QUARTER-TURN VALVE DESIGN

In general, valves may be classified as either linear operation or rotary operation. Linear-operating valves include slide gate, gate, globe, needle, and diaphragm valves. Rotary-operation valves include the BVs, BFVs, cone valves, and PVs in this manual of standard practice. As the full travel of many rotary-operation valves approximates a 90° rotation, they are often referred to as quarter-turn valves even though travel may be significantly more or less than 90° or a quarter turn. The quarter-turn valve is a versatile component for use with both shutoff and throttling in water systems. Quarter-turn valves are commonly supplied for the water industry in accordance with ANSI/AWWA C504-15, Standard for Rubber-Seated Butterfly Valves; ANSI/AWWA C507-15, Standard for Rubber-Seated Ball Valves 6 In. Through 60 In. (150 mm Through 1,500 mm); ANSI/AWWA C516-14, Standard for Large-Diameter Rubber-Seated Butterfly Valves Sizes 78 in. (2,000 mm) and Larger; or ANSI/AWWA C517-09, Standard for Resilient-Seated Cast-Iron Eccentric Plug Valves. As shown in Figures 1-2 through 1-5, these valves consist of a ball, cone, disc, or plug (closure member) supported in the body with a shaft, two stub shafts, or closure member trunnions and bearings. The quarter-turn operation is accomplished with a manual or power actuator connected to one shaft that penetrates the valve body and mounts to the exterior. Valves may have either metallic or elastomeric (rubber or plastic) seats.

Flow is controlled by positioning the closure member between 0° (0 percent, closed) to the full open (100 percent to approximately 90°) positions. The approximate effective throttling range for quarter-turn valves is 15° to 75° open (or 15 percent to 85 percent), but the range can vary based on application and valve design. Throttling at the lower angles (<15° or 15 percent) may cause erosion due to excessive local velocities or cavitation. Some valves are available with optional throttling or cavitation-reducing trim to extend the control range. See chapter 4 for discussions of cavitation. Throttling at the higher angles may provide limited control, because the valve has little effect on the system flow in many applications.

AWWA Ball Valve Design

AWWA BVs are characterized by the following design elements (see Figure 1-2):

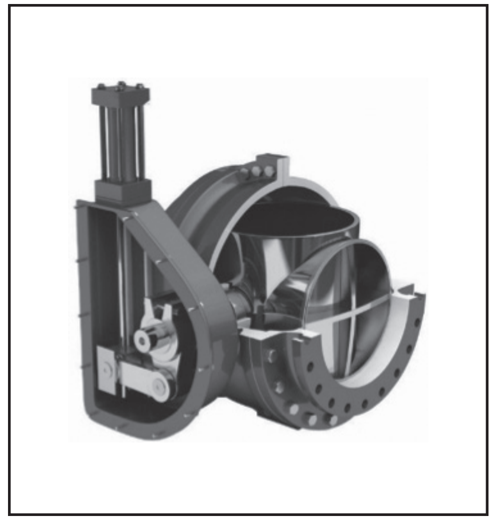
- They primarily consist of a bored spheroidal closure member (obturator/ball) that rotates roughly a quarter turn on a shaft or trunnion within the valve body.
- Ports are full nominal size (US customary) and unobstructed. (Note: Non-AWWA valves may have reduced ports.)
- Closure member (ball) may be shaft- or trunnion-mounted within the body. (Note: Non-AWWA valves may have a floating ball that is supported by the seats.)
- Closure member (ball), when symmetrically mounted, is position-seated and does not close tighter by increasing the shaft torque at the seated position.
- Closure member (ball), if eccentrically mounted, may be either position- or torque-seated. The seat seal will be tightened by increasing the shaft torque against the seat.
- Bodies may be one-, two-, three-, or four-piece construction.
- Seats may be metallic-to-metallic or metallic-to-elastomeric.
- BVs may be single- or double-seated.
- In double-seated valves, the downstream seat often provides the primary closure seal.
- They are often used in pump control service to control surges and may act as a power-operated check valve.
- In pump control service, a single-seated BV should be installed to seal tightest against system reverse flow, not pumped flow. (This generally places the seat end of the valve toward the pump.)
- They offer good flow control with a near equal percentage inherent valve characteristic.
- The full diameter circular port offers the lowest possible full open head loss.
- The unobstructed full open flow path does not produce cavitation or vibration.

AWWA Butterfly Valve Design

AWWA BFVs are characterized by the following design elements (see Figure 1-3):

- They primarily consist of a circular closure member (obturator/disc) that rotates about a quarter turn on a shaft within the valve body.
- Typically these valves have ports that are relatively close to full nominal size of pipe inside diameter.
- Seats may be metallic-to-elastomeric or metallic-to-metallic.

Two-Piece Body Design



Four-Piece Body Design

Courtesy of Val-Matic and DeZURIK

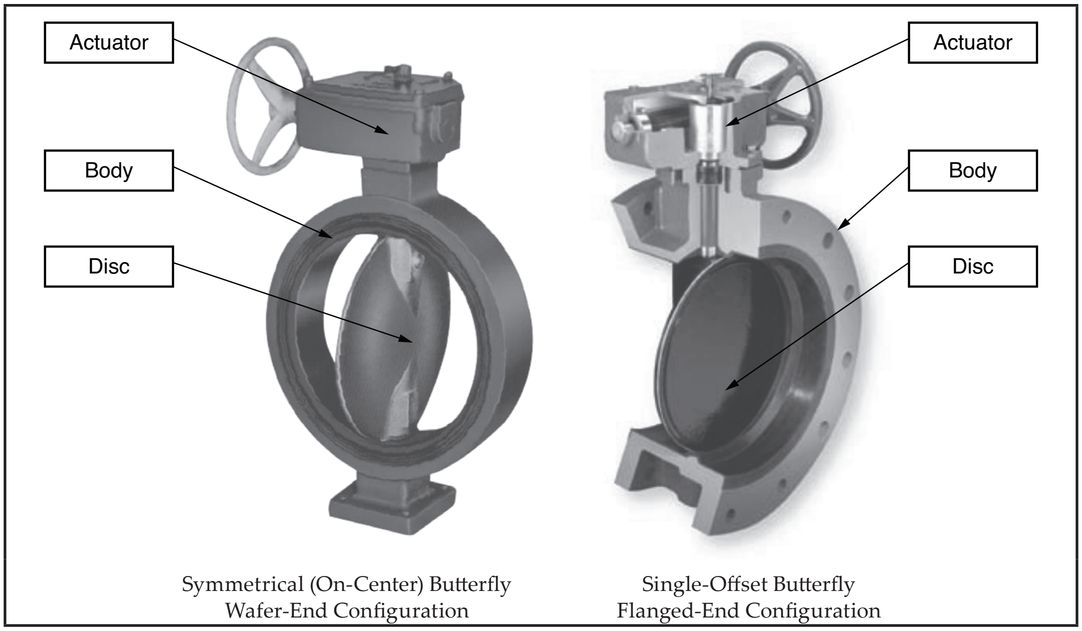
Figure 1-2 Typical ball valve construction

- The disc and seats may be symmetric (on center) design or single-offset, double-offset, or triple-offset designs.
- Symmetric and single-offset designs are position-seated and do not close tighter by increasing the seating shaft torque.
- Double- and triple-offset designs may be either position- or torque-seated. The seat seal will be tightened by increasing the shaft torque against the seat.
- They may be used in pump control service to control surges and may act as a power-operated check valve.
- They offer good flow control with a near equal percentage inherent valve characteristic.
- They offer very low full open head loss.
- Some designs may have a pressure seal direction preference.
- Some designs may have a flow and torque direction preference.

AWWA Plug Valve Design

AWWA PVs are characterized by the following design elements (see Figure 1-4):

- The PV primarily consists of an offset closure member (obturator/plug) that rotates about a quarter turn on a shaft within the valve body.
- Typically these valves have ports that are not circular and may have a full or slightly reduced port area.
- Seats are metallic-to-elastomeric.
- The plug and seat are an eccentric design (e.g., double-offset).
- Materials and construction are designed for both clean-water and wastewater service.
- PVs may be either position- or torque-seated, so the seat seal will be tightened by increasing the shaft torque against the seat.



Courtesy of Pratt and DeZURIK

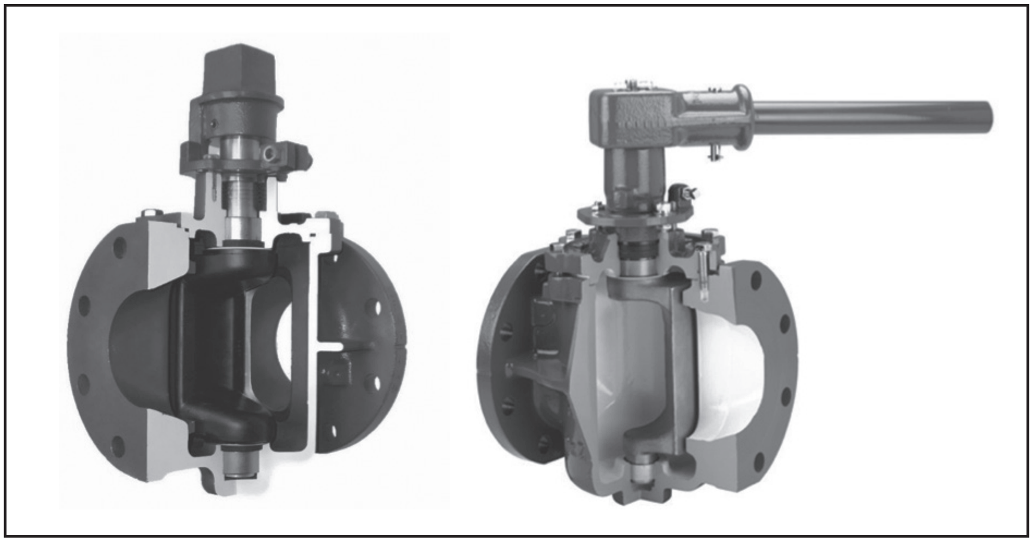
Figure 1-3 Typical butterfly valve construction

- They may be used in pump control service to control surges and may act as a power-operated check valve.
- In pump control service, a PV should be installed to seal tightest against system reverse flow, not pumped flow. (This generally places the seat end of the valve toward the pump.)
- They offer good flow control with a near equal percentage inherent valve characteristic.
- They offer moderate to low full open head loss.
- The normally preferred seal direction installation is the direct pressure orientation.

Rotary Cone Valve Design

Rotary cone valves used in the water and wastewater industry are not covered by an AWWA standard but are typically characterized by the following design elements (see Figure 1-5):

- They primarily consist of a bored and tapered conical closure member (obturator/cone) that rotates a quarter turn on a shaft or trunnion within the valve body and then seals by lowering the tapered cone into the metal seats with an axial movement.
- The actuator provides rotation and torque though the 90° operation and thrust or lift at the end of travel positions.
- Ports are full nominal size and unobstructed.
- Body is one piece with a closure bonnet or cover.
- Closure member (cone) is trunnion-mounted within the body.
- Closure member (cone) is typically symmetrically (concentrically) mounted and lift-seated but does close tighter by increasing the shaft thrust into the seat.
- Seat seals are metallic-to-metallic and generally Monel® for severe flow and corrosion-resistant service.



Courtesy of Val-Matic and DeZURIK

Figure 1-4 Typical plug valve construction

- The seats are precision ground to mate for drip-tight shutoff.
- Shaft axial movement of the cone into the seat provides the primary closure seal.
- They are often used in pump control service to control surges and may act as a power-operated check valve.
- They offer good flow control with an equal percentage inherent valve characteristic.
- The full diameter circular port offers the lowest possible full open head loss.
- Unobstructed flow path does not produce cavitation or vibration.

Butterfly Valve Offset Designs

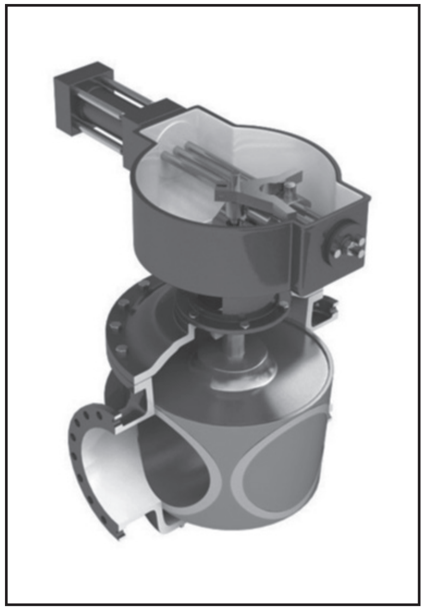
BFVs are available in several different offset designs. The discs, shafts, and seats may be aligned in various relative locations. The popular arrangements are the symmetric (on-center) design and the single-offset, double-offset, or triple-offset design.

In the symmetric (no-offset) design, the disc edge rotates and contacts perpendicularly with a theoretically cylindrical body seat surface (see Figure 1-6). This design is primarily position-seated and does not seat tighter with increased applied seating torque. In this configuration, the shaft intersects the disc's seating contact surface.

The single-offset design adds the seat offset (ε_1) and changes the perpendicular cylindrical contact surface into a theoretical conical body seat surface (see Figure 1-6). This design is also primarily position-seated and does not seat tighter with increased seating torque. In this configuration and the other offset designs, the shaft does not intersect the disc's seating contact surface.

The double-offset design adds the radial shaft offset (ε_2), which, in turn, offsets the centerline of the shaft perpendicular contact surface cone from the body perpendicular contact surface cone, resulting in an eccentric rotation and an eccentric wedge angle (see Figure 1-6). This eccentric action and wedge angle causes the seat load to increase with increased applied seating torque, and these valves may be torque-seated rather than just position-seated.

The triple-offset design rotates the body perpendicular contact surface cone of the double-offset design by the wedge angle offset (ε_3) (see Figure 1-6). This increases the



Courtesy of DeZURIK

Figure 1-5 Typical rotary cone valve construction

eccentric action and wedge angle. Therefore, the seat load increases with increased applied seating torque, and these valves may be torque-seated rather than just position-seated.

Offset valve designs are sensitive to differential pressure or flow direction orientation. Pressure on the shaft side of the disc tends to deflect the disc into the seat, and pressure from the seat side of the disc tends to push the disc away from the seat. Because of the eccentricity torque of the double- and triple-offset designs, the pressure difference in one direction assists seating but opposes unseating. In the opposite orientation, differential pressure opposes seating and assists unseating. Double- and triple-offset valves often have preferred installation orientations typically with the shaft upstream similar to the "direct" pressure installation of the eccentric PV. However, there are other torque-, flow-, and cavitation-related issues and concerns that might override and require a shaft downstream ("reverse") preferred installation orientation. Always check the flow direction and installation orientation markings on the valve and/or the manufacturer's drawings and O&M documentation before installing.

SYSTEM CONDITIONS

Analysis requires an understanding of system conditions that affect the head loss, torque, and cavitation calculations for quarter-turn valves, including those conditions on the following list:

1. **Fluid flow rate or velocity:** The maximum anticipated fluid flow rate or flow velocity through the nominal valve size should be determined with consideration of hydraulic design conditions and may include line break or other faulted condition flows when appropriate. The maximum anticipated flow velocity (or flow rate) is needed for the fluid dynamic torque calculations. This is generally at the full open position of the valve. If not at the full open position, then the valve's differential pressure or the valve's position when this maximum flow rate occurs must also be known and provided for analysis.

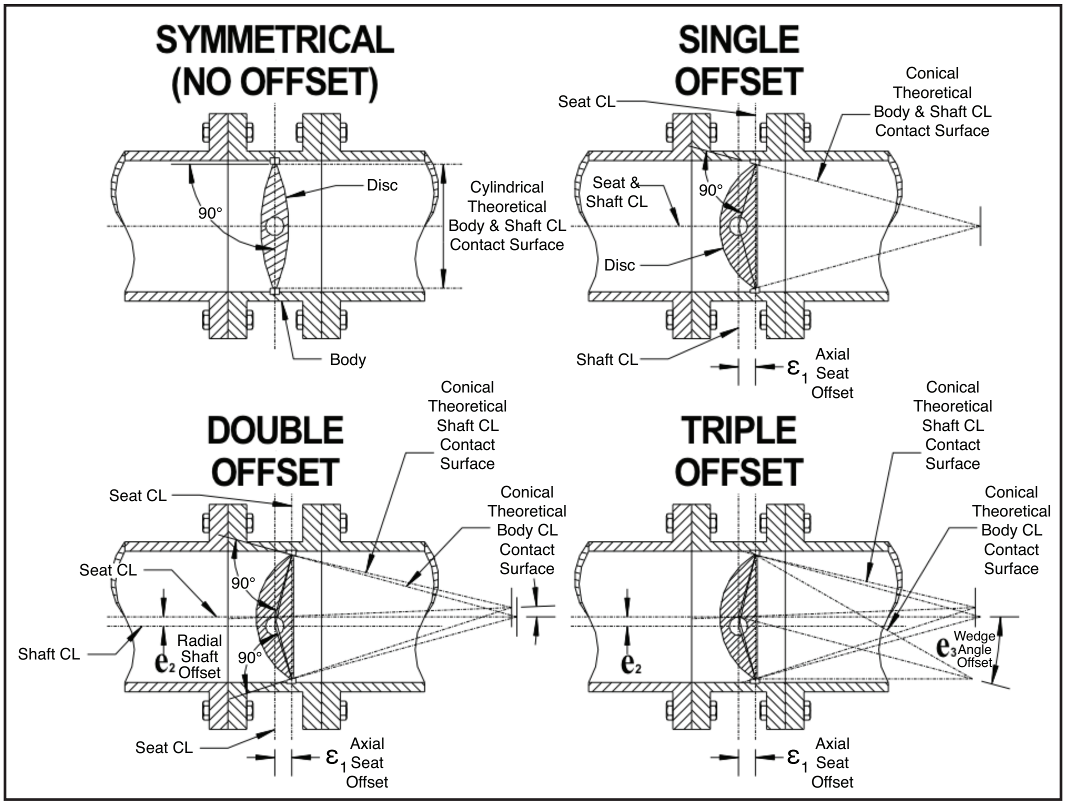


Figure 1-6 Butterfly valve offset designs

2. **Differential pressure:** The maximum (closed) differential pressure is needed for the torque calculations, sealing, and structural considerations. Cavitation calculations also require determination of pressure just upstream and downstream of the valve at the most severe throttling conditions.
3. **Piping installation:** Unless otherwise noted, the valve inlet and outlet are assumed to be straight piped conditions. Free discharge outlet and reservoir inlet installations (illustrated in Figure 1-7) represent unique applications that exceed the scope of this manual of standard practice. These installations affect both the head loss and torque characteristics of a quarter-turn valve. The valve manufacturer should be made aware of these conditions when applicable.
4. **Operating temperature:** Quarter-turn valves and actuators are designed to seat, unseat, control, and rigidly hold the valve closure members under a wide range of operating conditions. Temperature can affect seating torques and friction factors for valve bearings, so it should be considered. The operating temperature of the valves within this scope is 33°F to 125°F (0.6°C to 51.7°C). The valve manufacturer should be advised when operating temperatures are near the extremes or exceed the extremes of this range.
5. **Piping configuration:** Flow turbulence caused by upstream or downstream piping configurations may have a significant effect on valve performance. Nonsymmetrical flow streams or swirling action can magnify the operating torque and head loss of a quarter-turn valve and cause excessive vibration, reducing the valve's useful life. Installation guidelines are presented in chapter 6.

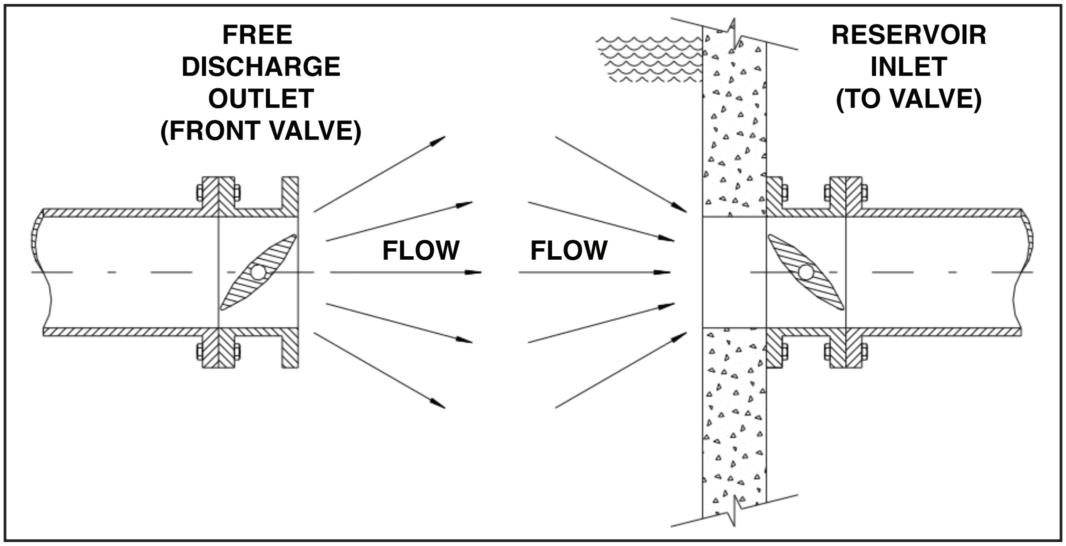


Figure 1-7 Free discharge and reservoir inlet installations of BFVs

DEFINITIONS

The following are definitions used in this manual of standard practice.

Actuator Sizing Torque, AST

The AST is typically calculated by the actuator manufacturer or the firm selecting and mounting the actuator to the valve. The AST is based on the MRST times the AF (AST = MRST × AF). The torque requirements at the seated position (seating or unseating) and the mid-stroke running positions (5° through 90°, opening or closing) should be evaluated. The AST has different values based on valve position. One or two conservative or bounding values may be used for simplicity, but this is not a single value and should normally be considered as a variable or curve of values.

Application Factor, AF

The AF is a multiplier that is used to determine the actuator and power source sizing. The AF is based on the type of actuator as well as the type of service (on-off service or modulating). The AWWA-recommended AF is provided in the valve standard.

Closure Member

This is the generic term for the valve's obturator. This may be a ball, disc, slide gate, gate, needle, plug, or other term for a specific valve design. Also see the obturator definition.

Direct Pressure

Applied pressure direction from which increasing pressure differential causes seat tightness and friction or seat load to increase as in a single-seated BV or eccentric PV. See the chapter 3 section "Flow Direction Through the Valve" and Figure 3-3.

Equal Percentage Inherent Valve Characteristic

A valve flow characteristic wherein a percentage change in valve position is accompanied by an equal percentage change in the inherent flow coefficient.

Minimum Required Shaft Torque, MRST

The MRST is normally provided by the valve manufacturer and is typically determined by the methods of this manual of standard practice. The torque requirements at the seated position (seating or unseating) and the mid-stroke running positions (5° through 90°, opening or closing) should be evaluated. The MRST has different values based on valve position. One or two conservative or bounding values may be used for simplicity, but this is not a single value and should be considered as a variable or curve of values. Note: For linear valves, such as slide gate, gate, and globe, MRST is often an acronym for *minimum required stem thrust* (not *shaft torque*).

Obturator

This is the generic term for the valve's closure member often used in international or European standards. This may be a ball, disc, slide gate, gate, needle, plug, or other term for a specific valve design.

Position Convention

The valve position may be designated in degrees open or percentage open. The fully closed position is referred to as the 0° or the 0% open position. The full open position is always referred to as 100% open and normally corresponds to approximately the 90° open position. Some quarter-turn valves may stroke more or less than 90° for full travel. This manual of standard practice uses degrees open as the position designation and assumes the 90° position and the 100% position are equivalent.

Position Seated

A position-seated quarter-turn valve does not seal tighter on the basis of how much torque is applied to the closure member at the seated position. These valves will often permit the closure member to pass completely through the seated position and start to reopen when passing the fully seated position. These valves are typically concentric in their rotation within and through the body seat. These may include symmetric and single-offset valves.

Reverse Pressure

Applied pressure direction from which the increasing pressure differential causes seat tightness and friction or load to decrease as in a single-seated BV or eccentric PV. See the chapter 3 section "Flow Direction Through the Valve" and Figure 3-3.

Seat Side Flow

This is the fluid flow direction from which the fluid passes the seat centerline before passing the shaft centerline in single-, double-, or triple-offset BFVs, single-seated BVs, and PVs. See the chapter 3 section "Flow Direction Through the Valve" and Figure 3-3.

Shaft-Mounted Closure Member

This closure member support design holds true when a shaft connects to and extends from the closure member directly though a bearing in the valve body. This design has two shaft extensions at diametrically opposed ends of the closure member, and one end penetrates the body for connection to an actuator. The shaft supports the closure member within the valve body and transmits torque for actuation.

Shaft Side Flow

This is the fluid flow direction from which the fluid passes the shaft centerline before passing the seat centerline in single-, double-, or triple-offset BFVs, single-seated BVs, and PVs. See the chapter 3 section "Flow Direction Through the Valve" and Figure 3-3.

Torque Seated

A torque-seated quarter-turn valve seals tighter on the basis of how much torque is applied to the closure member against the seats. These valves will generally not permit the closure member to pass completely through the fully seated position. These valves are typically eccentric in their rotation within and into the body seat. These may include double- and triple-offset BFVs, BVs, and PVs.

Trunnion-Mounted Closure Member

This closure member support design holds true when a shaft connects to and extends from the closure member directly though the valve body. The body bearings directly support the closure member at diametrically opposed closure member trunnions. This design has one shaft extension that penetrates the body for connection to an actuator and transmitting actuation torque. The shaft does not provide support of the closure member within the valve body.

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Chapter 2

Valve Head Loss and Equivalent Resistance System Model

A quarter-turn valve, like any restriction in a pipeline, is a source of head loss. As a quarter-turn valve closes, the head loss increases until the entire system head differential is established across the valve at the closed position. The head loss across a full open or throttling quarter-turn valve is important because head loss increases energy costs in pumping systems. In control applications, valve head loss is important to determine valve operating positions, control sensitivity, and cavitation potential.

The head loss or pressure drop across a quarter-turn valve can be calculated using many types of flow coefficients. Two commonly used coefficients are discussed in this chapter, and a simple methodology is presented for predicting quarter-turn valve head loss and flow rate during valve operation at mid-stroke positions.

DISCUSSION OF HEAD LOSS, CHOKING, AND CAVITATION

Most water works analyses can be performed without regard to choking limitations. Choking considerations add additional calculations and difficulty to the methodology but do not critically affect the results of most head loss, flow, or torque calculations within the scope of this manual. For more information on choking calculations also see ANSI/ISA S75.01.01-2012, ANSI/ISA S75.02.01-2008, and Hutchinson (1976).

The calculations of flow rate (or fluid velocity), valve head loss (or pressure drop), dynamic torque, and bearing friction torque given here do not include the effect of valve choking. Other coefficients (F_L, F_L^2, or K_M) are required to evaluate choking. It is not normally intended that the systems in which the valves of this scope are used should become fully choked at the maximum flow rates. If choking occurs, additional calculations are required to determine when classic head loss, flow, and torque equations are no longer appropriate and choked flow equations should be used. To add choking to the calculations requires the liquid pressure recovery factors (F_L^2) of the valve without attached fittings, and the system resistance is determined from the system upstream of the valve and the total system equivalent resistance. See the ANSI/ISA S75.02.01-2008 standard for more information on choking and liquid pressure recovery factors. Analyses including choked flow evaluations are not normally performed for the following reasons:

- These calculations add significant complexity to the methodology.
- In-depth operating system knowledge is not readily available to effectively perform the calculations.
- Torque calculations based on classic head loss and flow are generally acceptable for most applications.

When choked, the flow rate reaches a maximum limit (corresponding to the upstream line pressure at the valve), and the dynamic torque also does not continue to increase in proportion to increased valve head loss or pressure differential (because the flow rate is not increasing). Bearing friction torque does continue to increase with increasing valve differential pressure, but the method of determining the differential pressure across the valve changes to the maximum shutoff differential pressure minus the system loss at the choked flow rate. The cavitation analysis in this manual can also be used as an indicator of a choking concern as choking is actually caused by the development of heavy cavitation usually resulting in vapor pressure downstream of the valve.

Figure 2-1 shows typical results of how cavitation and choking occur in a quarter-turn valve application. From a calculation standpoint, the flow and differential pressure are treated as classical flow up to the F_L point and as a constant flow rate at all higher valve differential pressures. There is a small calculation error in the transition zone where actual test results are not linear in Figure 2-1. Just like the flow and torque coefficients, the F_L values vary by design and position. For critical, highly throttled pressure or flow control applications, it may be necessary to perform flow calculations that consider cavitation and choked flow to assure good design practices.

DEFINITIONS

Differential Pressure, ΔP

The maximum differential pressure (ΔP_{MAX}) of a quarter-turn valve is defined as the maximum difference between the upstream and downstream pressures when the valve is closed. During operation, the differential pressure varies on the basis of system conditions and valve position. The upstream pressure can be developed from a constant head source, such as the head from a reservoir or an elevated tank; a variable head source, such as head generated by a pump (see Figure 2-6); or a combination of both. For a conservative torque analysis, the downstream pressure may be assumed to be zero. The differential pressure is used to calculate the forces on the closure member (ball, cone, disc, or plug) and to estimate the flow characteristics of the piping system and calculate the flow rate and valve pressure drop at various valve openings. To determine differential pressure with a variable head created by a pump, the pump curve can be used to calculate the flow through the valve at all valve positions.

Flow Coefficients, General (C_v, C_f, C_d, C_{vm}, K, or K_v)

For any given line velocity or flow rate, a valve's head loss can be predicted by using standard flow equations and flow coefficients. Many flow equations are in use today, designed

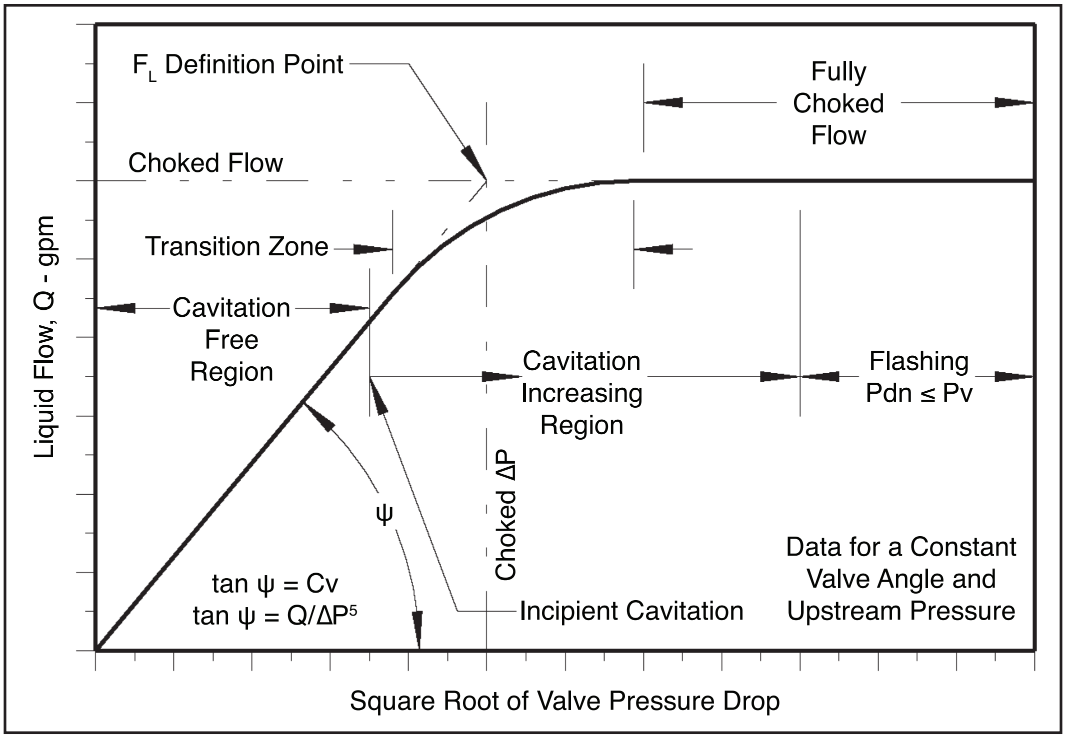

Figure 2-1 Typical quarter-turn valve flow, differential pressure, cavitation, and choking graphical explanation

to satisfy many types of specific flow systems and conditions. There are several variations of flow coefficients, such as K_v, C_v, C_f, and C_d. The two most common flow coefficients used with quarter-turn valves in water service are C_v and K. This manual primarily discusses the use of the resistance coefficient, K or K_v, and flow coefficient, C_v, from the basic fluid equations (Crane 2009). The metric equivalent to C_v is often referred to as K_v in other texts. In this manual, K_v will always refer to the flow resistance coefficient of the valve and not the metric equivalent to C_v. This manual refers to the metric equivalent to C_v as C_{vm} to avoid confusion.

For liquids, the flow coefficient, C_v, expresses the flow capacity in gallons per minute of 60°F (16°C) water with a pressure drop of 1 psi (6.89 kPa). For liquids, the flow resistance (or velocity head loss) coefficient, K, expresses the head loss as directly proportional to the velocity head. The subscript "v" is added to K to indicate when the coefficient is relative to the valve.

Flow coefficients are typically developed using a straight-run test pipe of the same nominal diameter as the valve. The valve may be connected to a pipeline with a slightly different inside diameter (ID). For example, unlined standard-weight 24-in. (600-mm) pipe has an ID of 23.25 in. (590.60 mm). Regardless of true or installed pipe ID, the valve calculations are generally based on nominal valve size—i.e., 24 in. (600 mm). Also, quarter-turn valve inlet diameters or port diameters are sometimes less than the nominal diameters; however, quarter-turn valve torque coefficients are still based on disc or nominal diameter (see chapter 5). The diameter of the butterfly valve disc is usually less than the pipe ID; disc diameter is often used in calculating hydraulic forces on the butterfly valve disc and shaft bearings but the use of nominal diameter in actuator sizing calculations is a conservative assumption.

Eccentric plug valves may have noncircular port areas, but the coefficients are still based on the nominal diameter of the valve. However, the grouping and extension of model test data will also be dependent on the port shape and port area.

Flow Coefficient, C_v

The C_v valve flow coefficient, often used for control valves, is defined as the flow of water at 60°F (16°C) in gallons per minute at a pressure drop of 1 psi across the valve. Many manufacturers publish C_v values for their valves in the fully open position and, often, the C_v values for intermediate positions and/or flow characteristic curves, which can be used to determine C_v values in throttling valve positions. These data can be readily used to calculate flow rate or pressure drop in water systems using the following equations:

(in US customary units)
$$Q = C_V \times \sqrt{\frac{\Delta P}{S_g}} \tag{2-1}$$

(in SI metric units)
$$Q = C_{VM} \times \sqrt{\frac{\Delta P}{S_g}} \tag{2-2}$$

$$C_{vm} = C_V \times 0.86 \tag{2-3}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
C_v	Valve flow coefficient. The flow of water through a valve at 60°F in US gallons per minute at a pressure drop of 1 psi Metric units note: In metric units, this variable is often identified as K_V. However, this is not used in this manual as it is easily confused with the resistance coefficient, K. When the resistance coefficient, K, is the resistance coefficient of the valve, it is subscripted with a "v" to indicate this reference, K_v. The metric flow coefficient, K_v, is defined as the flow of water with temperature ranging from 5–30°C through a valve in cubic meters per hour with a pressure drop of 1 bar (1 bar = 100 kPa). For purposes of this manual, the metric unit version of C_v will be identified by the variable symbol C_{vm}.	gpm/psi$^{1/2}$ (none)
C_{vm}	The metric equivalent to C_v (referred to as K_v in other texts)	none (m^3/hr/bar$^{1/2}$) (m^3/hr/(100 kPa)$^{1/2}$)
Q	Volumetric flow rate	gpm (m^3/hr)
S_g	Specific gravity of liquid relative to water at 60°F (16°C) (water = 1.0)	dimensionless
ΔP	Pressure drop (or loss) between any two reference points in a system	psid (kPa)

A common misconception relative to C_v is that it is sometimes referred to as the "flow capacity" of the valve. Although it is a measure of the valve's capacity to pass water at the specified 1-psi differential pressure, it is not the maximum possible flow rate for the valve or its design flow rate; that is, the flow capacity is not the same as the flow capability. A typical C_v for a 24-in. (600-mm) fully open butterfly valve is 24,400 gpm/psi$^{1/2}$. This statement does not mean that the flow capability of the valve is 24,400 gpm (1.58 m^3/sec), which would be equivalent to a flow velocity of 17.3 ft/s (5.3 m/s). Butterfly valves furnished per ANSI/AWWA C504-15 and ANSI/AWWA C516-14 are typically rated for a maximum velocity of 16 ft/s (4.9 m/s). That said, most butterfly valves meeting ANSI/AWWA C504-15 requirements are capable of withstanding and controlling full open flow velocities well above either 16 ft/s or 17.3 ft/s (4.9 m/s or 5.3 m/s). The C_v is not the same as the valve's design maximum flow rate or its maximum flow capability.

Additionally, it is difficult to compare the C_v of a valve with other pipe elements, such as elbows, tees, or runs of pipe, as these elements are usually provided as equivalent lengths or a resistance coefficient, K.

Flow Resistance Coefficient, K

The flow resistance coefficient, K, remains relatively constant for various types of valves or fittings through a broad range of sizes. K is defined as the number of flow velocity heads lost caused by a valve or fitting. For example, given that a typical 90° elbow has a K coefficient of 0.3, and a tee (branch run) has a K coefficient of 0.9, the system designer can place the 24-in. (600-mm) fully open butterfly valve K of 0.5 (versus a C_v of 24,400) as falling between the normal head losses of an elbow and the branch run of a tee.

The K values for pipe, valves, and fittings in series can be summed directly to find the total flow resistance K of the system. The K_{sys} of the system can then be used to estimate the flow rate and other parameters of the piping system, using the equivalent resistance model represented in Figure 2-6.

Before flow equations using the K_v valve resistance coefficient are presented, several qualifications are needed regarding the applicability of this methodology:

1. The flow equations and coefficients are based on test pipe size. The valve port, disc or closure member, and body diameter may vary from the nominal diameter as Figure 1-1 illustrates. The user of this method should adjust the coefficients to consider the effects of using a valve in a pipe of larger or smaller ID. This may be done using the beta ratio and relationships provided in the Crane (2009) and other fluid mechanics texts.
2. The flow equations are for water service at 60°F (16°C) assuming turbulent incompressible flow. Consult Crane's *Flow of Fluids* (2009) or similar texts for alternative equations that consider fluid density, compressible flow, laminar conditions, and choking flow.
3. Calculations are only as accurate as the test methodology and application used to develop the coefficients. Review laboratory test results whenever possible.
4. Two methodologies are commonly used for testing valves to determine head loss. ANSI/ISA S75.02.01-2008 calls for including the head loss of two times the pipe diameter upstream and six times the pipe diameter downstream of the valve. Other methods (and this manual) call for subtracting the pipe head loss so that only the valve head loss is reported. When valves with very low head loss are tested, the difference can be over 40 percent. The test procedures given in chapter 5 are similar to those in ANSI/ISA S75.02.01-2008 with only the valve's net head loss reported.

5. The accuracy of a calculated head loss is also affected by adjacent piping and fittings. Upstream reducers, elbows, or valves can cause high local velocities, which may significantly change the valve head loss. Similarly, unusual downstream conditions or free discharge applications produce varied results. As the valves of this scope are generally very low loss valves (e.g., the full open K_V is generally well below 1.0) the ID mismatch between the various types and materials of pipe and fittings, with and without interior linings and coatings, or a gasket that protrudes into the pipe can create significant additional head loss. In the case of the ball or rotary cone valves with which the port is full nominal diameter and the full open resistance is essentially the same as an equal length of the same diameter pipe, any ID change to or from the upstream and/or downstream pipe or fittings will generate an additional loss that can be greater than that of the valve itself.

These qualifications clearly suggest that the calculated head loss across a valve should be considered only an estimate, not an exact or calibrated quantity. Its purpose should be limited to energy comparison calculations or overall general system analysis.

Inherent Valve Characteristic

The control valve flow characteristic is the percentage of maximum flow though the valve plotted against travel position. In the case of quarter-turn valves, the travel position may be either percent of travel or opening angle. The inherent characteristic is that of the valve alone and is the percentage of full open C_v plotted against travel position as shown in Figure 2-4.

Installed Valve Characteristic

Once the valve is installed into a system, the flow resistance of the system interacts with that of the valve, and the installed characteristic is that of the valve and system resistance combined as shown in Figure 2-5.

Maximum System Flow Rate or Velocity, Q_{MAX} or V_{MAX}

The maximum system flow rate or velocity with the valve fully open is used to calculate valve torques in the range of open positions. If the valve will be operated during temporary high-flow conditions, such as fire flows or line-break flows, then the higher flow rate or velocity should be used. Although it is normally assumed that the maximum system flow rate or velocity is the same for both the opening and closing operations, it is often advantageous to consider extreme or emergency flow rates or velocities for only the operating (opening or closing) stroke of concern. This manual is generally based on the flow expressed as a fluid velocity in the pipe, not as a quantity, such as gallons per minute, liters per minute, cubic feet per second, or cubic meters per second. Conversions from quantity to pipe velocity are standard engineering practice and are not discussed in detail. This chapter provides the equivalent resistant system model that determines velocities (or flow rates) and pressures used for the torque and cavitation calculations.

HEAD LOSS CALCULATIONS

Given the K (or K_V for K of a valve) flow coefficient for a valve or fitting, head loss can be calculated with the following formula from Crane (2009):

$$\Delta H = \frac{K \times V^2}{2 \times g} \qquad (2\text{-}4)$$

If the flow coefficient is expressed as a C_v, it can be equated to K by

(in US customary units) $$K = \frac{891 \times D^4}{C_V^2} \qquad (2\text{-}5)$$

If the flow coefficient is expressed as a C_{vm}, it can be equated to K by

(in SI metric units) $$K = \frac{8.52 \times (D \times U_{C1})^4}{C_{vm}^2} \qquad (2\text{-}6)$$

Also, if the flow is expressed in gallons per minute, the fluid velocity can be found by

(in US customary units) $$V = \frac{0.4085 \times Q}{D_{PIPEID}{}^2} \qquad (2\text{-}7)$$

If the flow is expressed in meters per hour, the fluid velocity can be found by

(in SI metric units) $$V = \frac{353.7 \times Q}{D_{PIPEID}{}^2} \qquad (2\text{-}8)$$

where

Variable	Definition or Description	Units US Customary (SI metric)
C_v	Valve flow coefficient. The flow of water through a valve at 60°F in US gallons per minute at a pressure drop of 1 psi Metric units note: In metric units, this variable is often identified as K_V. However, this is not used in this manual as it is easily confused with the resistance coefficient, K. When the resistance coefficient, K, is the resistance coefficient of the valve, it is subscripted with a "v" to indicate this reference, K_v. The metric flow coefficient, K_v, is defined as the flow of water with temperature ranging 5–30°C through a valve in cubic meters per hour with a pressure drop of 1 bar (1 bar = 100 kPa). For purposes of this manual, the metric unit version of C_v will be identified by the variable symbol C_{vm}.	gpm/psi$^{1/2}$ None

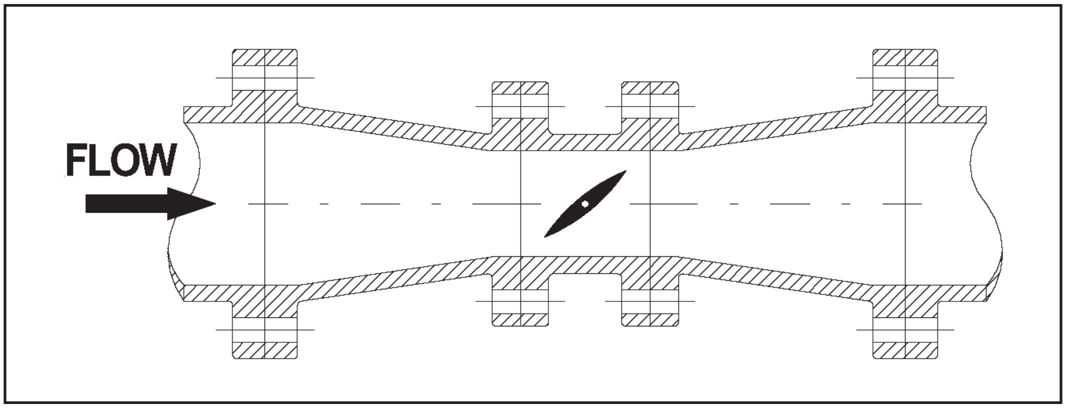

Figure 2-2 Smaller-than-line-sized butterfly valve installation

Variable	Definition or Description	Units US Customary (SI metric)
C_{vm}	The metric equivalent to C_v (referred to as K_v in other texts)	None (m^3/hr/bar$^{1/2}$) (m^3/hr/(100 kPa)$^{1/2}$)
D	Nominal valve diameter	in. (mm)
D_{PIPEID}	Pipe ID	in. (mm)
g	Gravitational constant Acceleration due to gravity, 32.2 ft/s² (9.81 m/s²)	ft/s² (m/s²)
K	Flow resistance coefficient of any component or fitting	dimensionless
Q	Volumetric flow rate	gpm (m^3/hr)
U_{C1}	Units conversion factor: US customary for torque in in.-lb: U_{C1} = 1 in./in. US customary for torque in ft-lb: U_{C1} = 1⁄12 (0.0833) ft/in. Metric for torque in N-m: $U_{C1} = 1 \times 10^{-3}$ (0.001) m/mm	in./in. (ft/in.) (m/mm)
V	Velocity of fluid flow	feet per second, ft/s (meters per second, m/s)
ΔH	Head loss between any two reference points in a system	feet of water (meters of water)

Several other types of flow equations are given in the references cited in the bibliography.

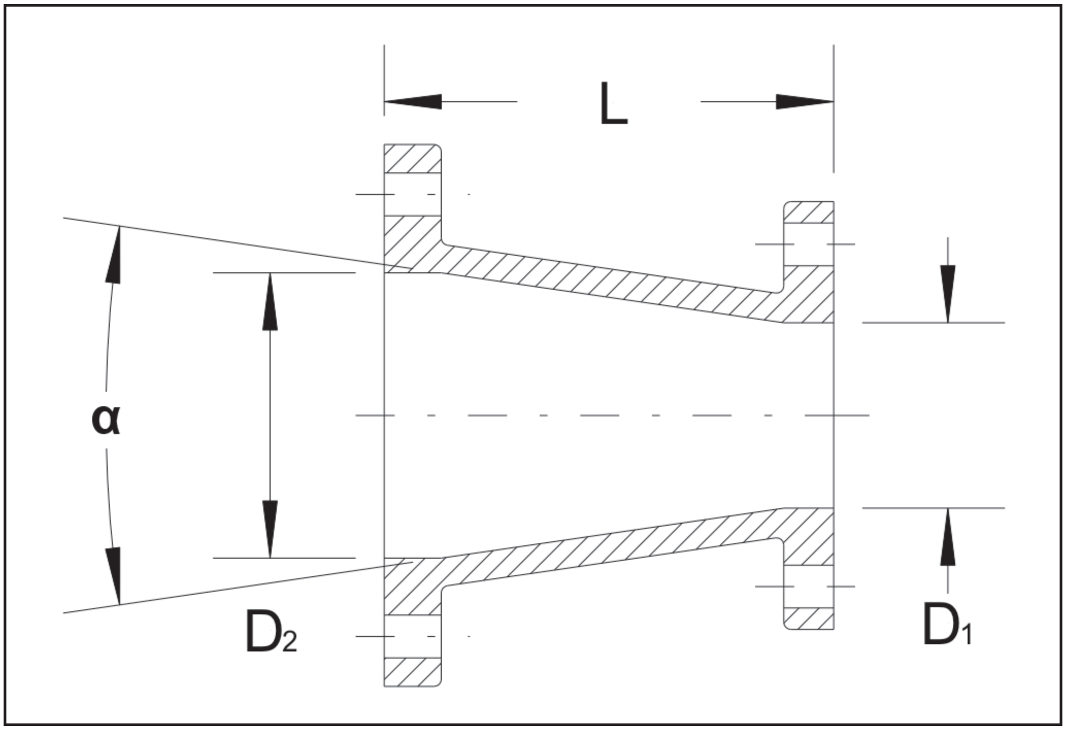

Figure 2-3 Reducer geometry

REDUCER INSTALLATIONS

It is common to use a smaller-than-line-size quarter-turn valve for pressure or flow control applications. The pipeline will therefore include pipe reducers on either side of the valve (see Figure 2-2). Reducer nomenclature and geometry is shown in Figure 2-3

Flow resistance coefficients (K) of elements in series are added directly but must be relative to a constant given diameter. When analyzing on the basis of the valve's diameter (the smaller diameter), the flow resistance of the reducer and expander sections should be evaluated as K_1 or based on the valve diameter (D or D_1). In this manner, these resistance values can be added to the valve flow resistance value (K_V) and treated as a unit in the analysis using the diameter of the valve as the flow rate–to–velocity conversion. If the analysis is to be based on the large pipeline diameter as the flow rate–to–velocity conversion, the reducer K_1, increaser K_1, and valve K_{Vd1} should be divided by β_r^4 and added or added together and the sum divided by β_r^4 to obtain the K_2 equivalent.

The resistance coefficient, K_1, for an upstream reducer can be calculated by the following formula from Crane (2009):

$$\alpha = 2 \times \mathrm{ARCTAN}\left(\frac{D_2 - D_1}{2 \times L}\right) \tag{2-9}$$

$$K_1 = 0.8 \times \left(\mathrm{SIN}\frac{\alpha}{2}\right) \times \left[1 - \left(\frac{D_1}{D_2}\right)^2\right] \tag{2-10}$$

The resistance coefficient, K_1, for a downstream reducer (increaser) can be calculated by the following formula:

$$K_1 = 2.6 \times \left(\mathrm{SIN}\frac{\alpha}{2} \right) \times \left[1 - \left(\frac{D_1}{D_2} \right)^2 \right]^2 \tag{2-11}$$

To be based on the pipeline diameter, K_2, use the following equations:

$$\beta_r = \frac{D_1}{D_2} \tag{2-12}$$

$$K_2 = \frac{K_1}{\beta_r^4} \tag{2-13}$$

$$K_{Vd2} = \frac{K_{Vd1}}{\beta_r^4} \tag{2-14}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
D_1	Reducer reduced pipeline (small end) diameter	in. (mm)
D_2	Reducer large pipeline (large end) diameter	in. (mm)
K_1	Reducer resistance coefficient based on the reduced (small end) diameter, D_1	dimensionless
K_2	Reducer resistance coefficient based on the large end diameter, D_2	dimensionless
K_{Vd1}	Valve resistance coefficient based on the reference diameter, D_1 (typically the nominal diameter, D)	dimensionless
K_{Vd2}	Valve resistance coefficient based on the diameter, D_2	dimensionless
L	Tapered length of reducer	in. (mm)
α	Reducer (increaser) included angle, degrees; for angles ≤45° (π4 radians]	degrees (radians)
β_r	Beta ratio for reducer flow resistance calculation: Reducer (increaser) diameter ratio	dimensionless

Reducer Assembly Flow Coefficient Example

Given the following input valve and reducer data, Tables 2-1 and 2-2 calculate the flow coefficients for the reducer, valve, and increaser as an assembly:

Table 2-1 Reducer and Increaser Calculations

Input	Input	Input	Calc.	Calc.	Calc.	Calc.	Calc.	Calc.
Valve Size D, D_1 inches	Pipe Size D_2 inches	Fitting Length inches	Included Angle α Radians Eq 2-9	Beta Ratio (D_1/D_2) β_r Eq 2-12	Reducer K_1 Valve Diameter Basis Eq 2-10	Increaser K_1 Valve Diameter Basis Eq 2-11	Reducer K_2 Pipe Diameter Basis Eq 2-13	Increaser K_2 Pipe Diameter Basis Eq 2-13
24	36	36	0.330	0.667	0.07	0.13	0.37	0.67

Table 2-2 Valve Reducer, and Increaser Assembly Adjusted Flow Coefficients

Input	Input	Input	Calc.	Calc.	Calc.
Valve Angle θ (deg.)	$K_{V\theta}$ [1]	Valve $C_{V\theta}$ gpm/psi$^{1/2}$ [1]	Calculated Reducer Assembly Flow Resistance "$K_{RA1\theta}$" Valve Diameter Basis	Calculated Reducer Assembly Flow Resistance "$K_{RA2\theta}$" Pipe Diameter Basis	Calculated Reducer Assembly Flow Coefficient "$C_{Vassy\theta}$" gpm/psi$^{1/2}$ Eq 2-5
90	0.30	31,391	0.50	2.56	24,195
80	0.40	27,185	0.60	3.06	22,105
70	1.10	16,393	1.30	6.61	15,051
60	3.10	9,765	3.30	16.73	9,458
50	8.30	5,968	8.50	43.06	5,896
40	24.80	3,453	25.00	126.59	3,438
30	83.30	1,884	83.50	422.74	1,882
20	333.30	942	333.50	1,688.37	941
10	3,000.00	314	3,000.20	15,188.54	314

[1] The valve data are for the example only and have no relationship to an actual valve.

INHERENT AND INSTALLED CONTROL VALVE FLOW CHARACTERISTICS

From the definitions at the beginning of the chapter, the control valve flow characteristic is the percentage of maximum flow through the valve plotted against travel position. In the case of quarter-turn valves, the travel position may be either percentage of travel or opening angle. The inherent characteristic is that of the valve alone and is the percentage of full open C_v plotted against travel position as shown in Figure 2-4. Once the valve is installed into a system, the flow resistance of the system interacts with that of the valve, and the installed characteristic is that of the valve and system resistance combined as shown in Figure 2-5.

The graph lines of Figure 2-5 show how the valve and system interact as the system length and resistance increase. The system length is shown as the number of diameters of pipe (L/D). As shown in this figure, the installed characteristic curve moves up and to the left as the system resistance increases. If the valve is installed in a system by itself and there is little or no piping attached, the valve has complete control of the flow and the differential pressure across the valve is essentially a constant at all angles. As the system resistance increases, the valve's capability to control the flow is reduced to lower valve angles, and the differential pressure across the valve is reduced as the valve opens.

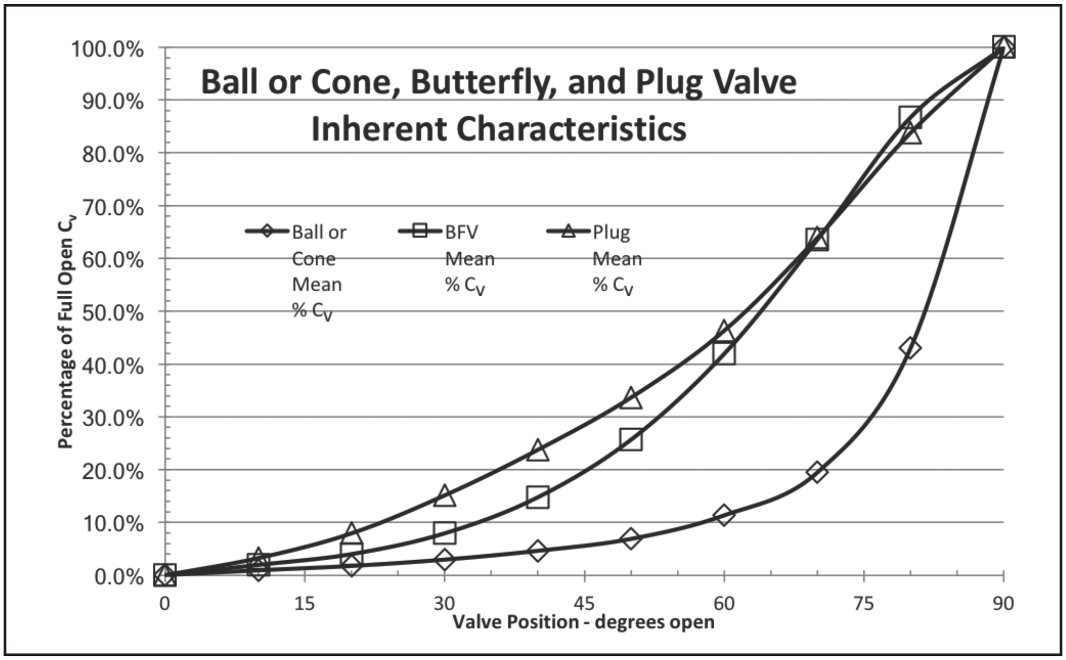

Figure 2-4 Typical inherent valve characteristics

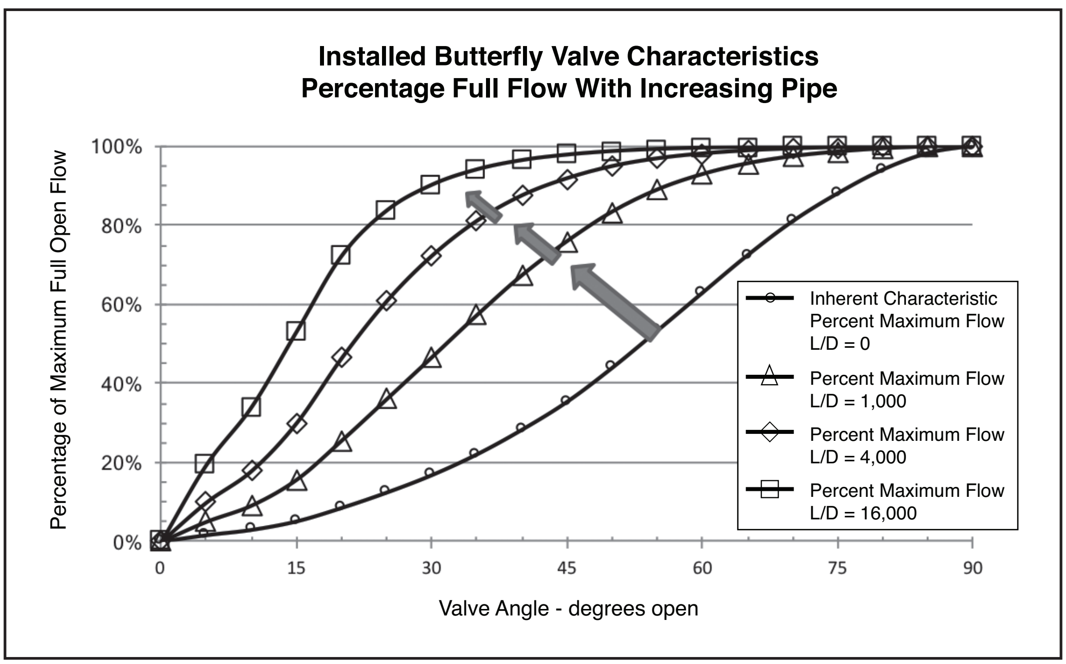

Figure 2-5 Installed valve characteristics

EQUIVALENT RESISTANCE SYSTEM MODEL

As can be seen from the previous installed characteristics discussion, the system resistance affects the valve's differential pressure and flow rate at a given valve position. To see how the valve controls the flow and pressure drop and to determine operating torque, the installed characteristic of the system must be calculated. This model is based on knowing the maximum potential energy of the system (pressure) and the maximum kinetic energy

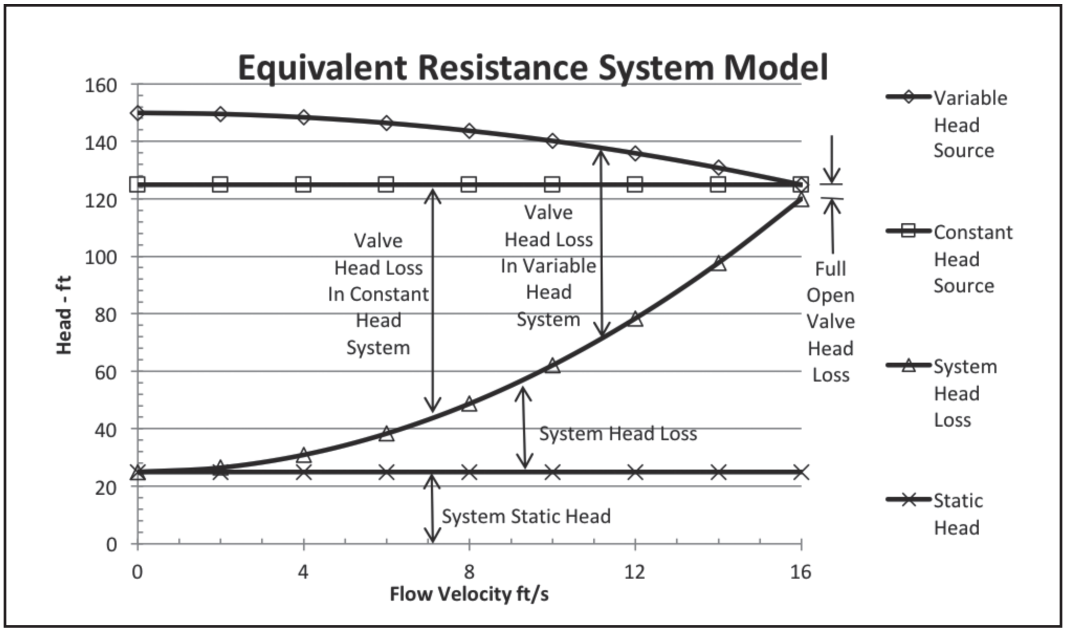

Figure 2-6 Equivalent resistance system model

of the system (flow). The maximum potential energy of the system is the differential head or pressure across the valve when it is closed and the flow rate or velocity is zero. The maximum kinetic energy the system can develop is the maximum flow rate or line velocity when the valve is fully open. With these two inputs, an equivalent system resistance (K_{sys}) can be determined. This model is graphically shown in Figure 2-6.

This model assumes that during the valve's opening or closing operation the remainder of the system is not changing and that the only resistance change is the valve being analyzed. Most systems have several operating functions and conditions, such as start-up, shutdown, multiple normal operating conditions or scenarios, and emergency functions. Typically, the highest differential head or pressure and the greatest velocity or flow rate are used even if they do not coincide functionally. This is a conservative assumption for actuator sizing. Alternately, each operating case can be analyzed separately.

The valve's location within the system is not required for purposes of actuator sizing and valve head loss calculations. However, for choking and cavitation analysis, it is necessary to determine the static pressure just upstream of the valve. For accurate analysis of choking and cavitation conditions, the valve's upstream and/or downstream pressures are required as well as the differential pressure. For choking and cavitation analysis, the equivalent system resistance must be split into upstream and downstream components, and the closed static upstream (or downstream) pressure is required as well as the differential pressure. It can be conservatively assumed that the valve is located at or near the end of the system and that most or all of the system losses occur upstream of the valve as the pressure downstream of the valve approaches 0 psig (0 kPa). This is a worst-case analysis, but results are often unfavorable or unacceptable as cavitation is almost always predicted.

For single-point control valve sizing calculations, the upstream pressure, the controlled flow rate, the downstream pressure or differential pressure, and the water temperature are required to predict throttled valve position, choking, and the operating cavitation index. Given three or more sets of controlled flow conditions (pressures, flow, and temperature) for the extremes of the process, an operating "window" can be calculated, and performance can be evaluated within the window.

Constant Head Source Methodology

A typical constant head source application in a water system is the flow of water from an elevated reservoir to a residential water tap. Although the water level in the elevated reservoir changes throughout the day, at any given time, the supply head (which is the elevation of the water level in the reservoir) is constant regardless of the water flow rate. The sum of all other losses, except the valve being analyzed, in a flow system equals ΔH_{sys}. The calculation of operating torque for a single quarter-turn valve in a system assumes that no other variable loss coefficients, such as other valves, are changing in the system during the valve travel. Therefore, the velocity head loss coefficient of all components other than the valve in question can be considered constant and equal to K_{sys}. When calculating the head loss (or differential pressure), flow rate (or line velocity), and operating torque of a quarter-turn valve, the system will be evaluated as two basic components: (1) the valve being analyzed and (2) the remainder of the system piping. In a system with a constant head source, ΔH_{sys} is considered constant for all flow conditions. When a valve in the system is closed, the maximum differential pressure across the valve (ΔH_{MAX}) is equal to the maximum differential across the system (ΔH_{sys}).

The system is modeled on the basis of the maximum shutoff (closed) differential head or differential pressure and the maximum flow rate or velocity. (Note: These inputs are required by AWWA C504-15, AWWA C507-15, AWWA C516-14, and AWWA C517-09.)

The following steps and equations are used to develop the head or pressure losses and flow rates at various valve positions using the equivalent resistance system model.

1. Calculate K_{sys} using K_v for a fully open quarter-turn valve:

$$K_{sys} = \frac{2 \times g \times \Delta H_{MAX}}{V_{MAX}^2} - K_{V90} \tag{2-15}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
g	Gravitational constant Acceleration due to gravity, 32.2 ft/s² (9.81 m/s²)	ft/s² (m/s²)
K_{sys}	System flow resistance coefficient (excluding the valve)	dimensionless
K_{V90}	Flow resistance coefficient of valve at full open (≈90°, ≈π/2 radians). Note: Use of K_{V90} assumes the valve travels 90° to full open.	dimensionless
V_{MAX}	Maximum full open velocity, ft/s (Note: Based on nominal valve diameter if converted from a quantity flow rate.)	feet per second, ft/s (meters per second, m/s)
ΔH_{MAX}	Head loss across the closed valve or total system with valve closed	feet of water (meters of water)
ΔH_{SYS}	Head loss across the system	feet of water (meters of water)

2. The flow velocity through the valve at the valve angle (θ) may be calculated using

$$V_\theta = \sqrt{\frac{2 \times g \times \Delta H_{MAX}}{\left(K_{sys} + K_{V\theta}\right)}} \tag{2-16}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
$K_{V\theta}$	Flow resistance coefficient of the valve at valve angle θ	dimensionless
V_θ	Approach velocity of fluid flow at valve angle θ	feet per second, ft/s (meters per second, m/s)

3. Calculate $\Delta H_{V\theta}$ at the valve angle (θ):

$$\Delta H_{V\theta} = \frac{K_{V\theta} \times V_\theta^2}{2 \times g} \tag{2-17}$$

or

$$\Delta H_{V\theta} = \frac{\Delta H_{MAX} \times K_{V\theta}}{\left(K_{V\theta} + K_{SYS}\right)} \tag{2-18}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
$\Delta H_{V\theta}$	Head loss across the valve at angle θ	feet of water (meters of water)

4. Calculate ΔP_θ at the valve angle (θ):

$$\Delta P_{V\theta} = 0.4335 \times \Delta H_{V\theta} \tag{2-19}$$

or

$$\Delta P_{V\theta} = \frac{\Delta P_{MAX} \times \Delta K_{V\theta}}{\left(K_{V\theta} + K_{SYS}\right)} \tag{2-20}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
$\Delta P_{V\theta}$	Pressure drop (or loss) across the valve at valve angle θ	psid (kPa)

5. Repeat steps 2, 3, and 4 for other valve angles (θ):

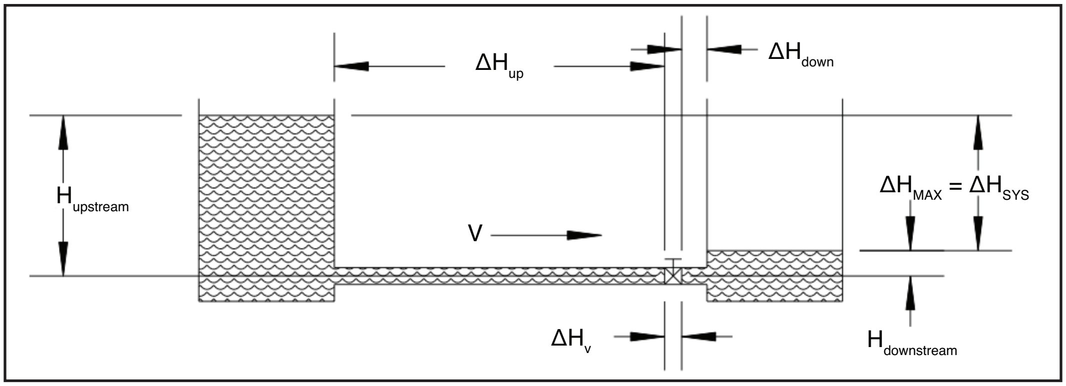

Figure 2-7 Example equivalent resistance system model diagram.

Table 2-3 Head Loss Calculation Data for Constant Head Source Example

Input	Input	Calc.	Calc.	Calc.	Calc.	Calc.	Calc.	Calc.
Valve Angle θ (degrees)	$K_{V\theta}$ [1]	$\Delta H_{v\theta}$ [2]	V_θ [2]	ΔH_{SYS} (100%) [4]	ΔH_{SYS} Upstream of Valve (75%) [3], [5]	Velocity Head at Valve $V^2/2G$ [3], [5]	Static Head H Upstream of Valve [3], [5]	Static Pressure P Upstream of Valve [3], [5]
90	0.30	0.9	14.2	99.1	74.3	3.1	122.6	53.1
80	0.40	1.2	14.2	98.8	74.1	3.1	122.8	53.2
70	1.10	3.4	14.0	96.6	72.5	3.1	124.5	54.0
60	3.10	8.9	13.6	91.1	68.3	2.9	128.8	55.8
50	8.30	20.8	12.7	79.2	59.4	2.5	138.1	59.9
40	24.80	44.0	10.7	56.0	42.0	1.8	156.2	67.7
30	83.30	72.5	7.5	27.5	20.6	0.9	178.5	77.4
20	333.30	91.3	4.2	8.7	6.5	0.3	193.2	83.8
10	3,000.00	99.0	1.5	1.0	0.8	0.0	199.2	86.3
0	Inf.	100.0	0.0	0.0	0.0	0.0	200.0	86.7

[1] The valve data are for the example only and have no relationship to an actual valve.

[2] Used for head loss (pressure loss), velocity (flow), and torque calculations.

[3] Used for choking and cavitation calculations; not required for head loss (pressure loss), velocity (flow), and torque calculations.

[4] Informational only.

[5] These columns are calculated using standard hydraulic principles left to the reader (see Crane, 2009).

Constant Head Source Methodology Example

Given the following valve and system data,

- 24-in. AWWA class 150B butterfly valve with 24-in.-diameter single-offset disc, seat side flow
- Maximum head differential (ΔH_{MAX}) is 100 feet of water (ΔP_{MAX} = 43.4 psid, Eq 2-19)

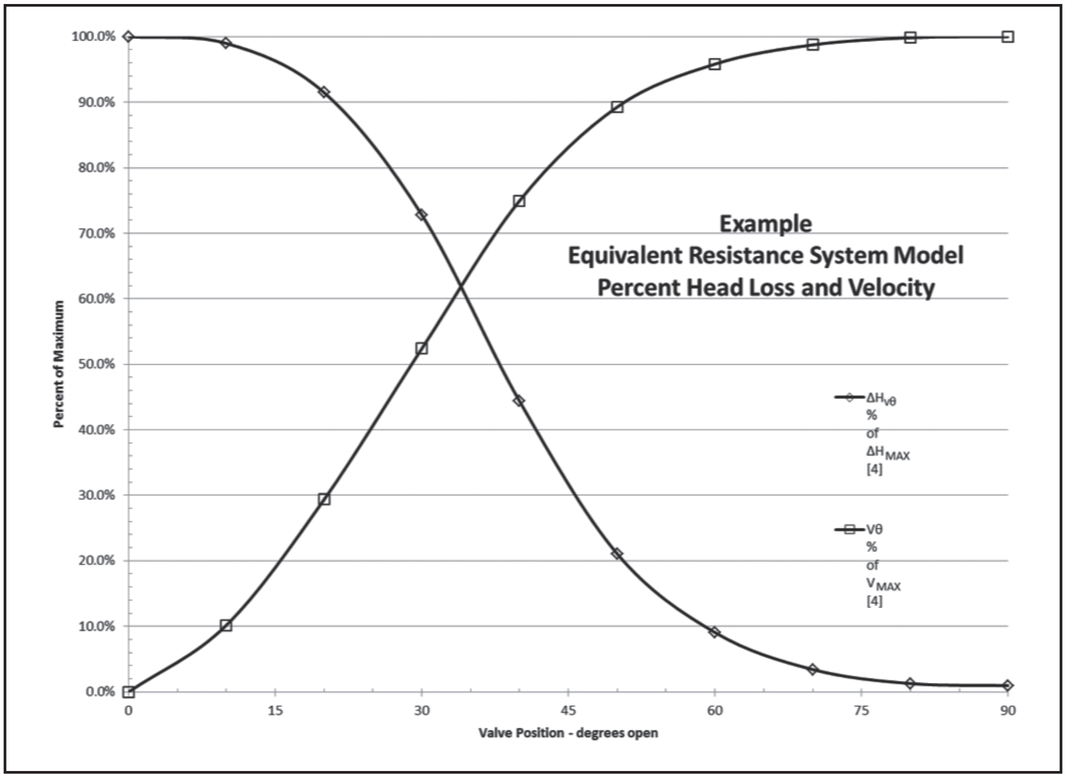

Figure 2-8 Graphical relationship between head loss and velocity

- Closed upstream head ($H_{upstream}$) is 200 feet of water ($P_{upstream}$ = 86.7 psi, Eq 2-19)
- System downstream static head ($H_{dnstream}$) is $H_{upstream} - \Delta H_{MAX}$ = 200 – 100 = 100 feet of water ($P_{dnstream}$ = 43.4 psi, Eq 2-19)
- Maximum system flow rate, Q_{MAX} = 20,000 gpm (V_{MAX} = 14.2 ft/s, Eq 2-7)
- Water temperature is 60°F (with a vapor pressure of 0.58 feet absolute = 0.25 psia = –33.3 feet gauge, –14.4 psig)
- $K_{v\theta}$ listed in Table 2-3
- $K_{sys} = \dfrac{2\times 32.17\times 100}{16^2} - 0.3 = 31.61$ from Eq 2-15
- Percentage of system upstream of valve = 75% ($K_{sys\text{-}up}$ = 23.70)
- See Figure 2-7 for system model as well as Table 2-3 and Figure 2-8 for calculation results and a graph of the results

VARIABLE HEAD SOURCE METHODOLOGY

Similar to the constant head source methodology, a typical variable head source application is pumping water from a lake up to an elevated reservoir. In this application, the supply head of the pump is a function of the flow rate through the pump as shown on the pump curve. The sum of the head losses in a flow system at any given time equals ΔH_{sys}. It is seldom necessary to use this method as assuming a constant head source is adequate for most uses. If the pump curve is in the shape of a second-order inverted parabola (similar to that shown in Figure 2-8), the results from a constant head assumption and the variable head methodologies are identical. It is only necessary to use this variable head method when the pump curve is oddly shaped and cannot be approximated by a second-order polynomial. This is because the variable shutoff head (or pressure) is higher than the constant-head shutoff head (or pressure) for the same run-out or maximum fluid flow (or velocity). Therefore, using the constant head method, the calculated system resistance (K_{sys}) will artificially increase. This increase is essentially equal and opposite to the reduction in the source head. As the valve responds to the area between the variable shutoff head (or pressure) curve and the system resistance curve, the resulting differential calculations match.

1. Calculate K_{sys} using Eq 2-15 with one exception as noted below.
   The ΔH_{sys} at V_{MAX} are read from the pump curve. The value for ΔH_{sys} is the difference between the pump supply head and the system static head.
2. Calculate the flow velocity of the system at the desired valve angle (θ) using

$$V_\theta = \sqrt{\frac{2 \times g \times \Delta H_{MAX}}{\left(K_{sys} + K_{V\theta}\right)}} \tag{2-21}$$

   In this equation, $K_{v\theta}$ is the K of the valve at the desired valve angle. Because $\Delta H_{sys\theta}$ is variable and dependent on $V_{V\theta}$, this equation must be solved using an iterative process.

   a. Assume a value for $V_{V\theta}$ less than that of V_θ at the next higher valve angle.
   b. Calculate the corresponding $\Delta H_{sys\theta}$:

$$\Delta H_{sys\theta} = \frac{\left(K_{SYS} + K_{V\theta}\right) \times V_\theta^2}{2 \times g} \tag{2-22}$$

   c. Examine the pump curve and determine if $\Delta H_{sys\theta}$ matches the curve.
   d. If the $\Delta H_{sys\theta}$ is not a good match with the calculated value, then assume a higher or lower V_θ and repeat Steps 2a through 2c until $\Delta H_{sys\theta}$ matches the pump and system curve.
3. Calculate $\Delta H_{v\theta}$ at the desired valve angle (θ):

$$\Delta H_{V\theta} = \frac{K_{V\theta} \times V_\theta^2}{2 \times g} \tag{2-23}$$

4. Calculate $\Delta P_{V\theta}$ at the desired valve angle (θ):

$$\Delta P_{V\theta} = 0.4335 \times \Delta H_{V\theta} \tag{2-24}$$

5. Repeat Steps 2 through 4 for other valve angles.

ENERGY CALCULATIONS

One reason for using quarter-turn valves is that they offer low head loss in a compact space. In a pumped system, when flow passes through a valve or fitting, the resulting head loss requires additional energy from the pumps. Head loss therefore translates directly into electricity consumption by the pump motors.

An equation used for calculating yearly energy cost is as follows:

Table 2-4 Annual Pumping Cost Example Calculation

Valve Type	Full Open K_V	ΔH at 15,000 gpm (10.6 fps)	A_{COST}
Ball or Rotary Cone 1	0.04	0.07	$ 97.25
Ball or Rotary Cone 2	0.05	0.09	$ 121.56
Ball or Rotary Cone 3	0.06	0.10	$ 145.87
Butterfly or Plug 1	0.40	0.70	$ 972.50
Butterfly or Plug 2	0.50	0.87	$ 1,215.62
Butterfly or Plug 3	0.60	1.05	$ 1,458.75
Globe 1	3.20	5.59	$ 7,779.98
Globe 2	4.00	6.99	$ 9,724.97
Globe 3	4.80	8.38	$ 11,669.97

(in US customary units)
$$A_{COST} = \frac{1.65 \times Q \times \Delta H \times S_g \times C \times U}{E} \tag{2-25}$$

(in SI metric units)
$$A_{COST} = \frac{23.8 \times Q \times \Delta H \times S_g \times C \times U}{E} \tag{2-26}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
A_{COST}	Annual energy cost	$/year
C	Cost of electricity	$/kWh
E	Efficiency of pump and motor set (80%; 0.80, typical)	Decimal/ fraction (%/100)
Q	Volumetric flow rate	gpm (m^3/hr)
S_g	Specific gravity of liquid relative to water at 60°F (16°C) (water = 1.0)	dimensionless
U	Pump usage percentage, 100% (1.0) equals 24 hours per day	Decimal/ fraction (%/100)
ΔH	Head loss between any two reference points in a system	feet of water (meters of water)

Energy Calculation—Example

For example, a 24-in. (600-mm) ball or rotary cone valve (K_v = 0.05, ±0.01[20%]), a butterfly or plug valve (K_v = 0.50, ±0.1[20%]), and a globe valve (K_v = 4.00, ±0.8[20%]) are installed in a 15,000 gpm (10.6 ft/s, 3.2 m/s, 1 m^3/s) pump line. The efficiency of the pump and motor

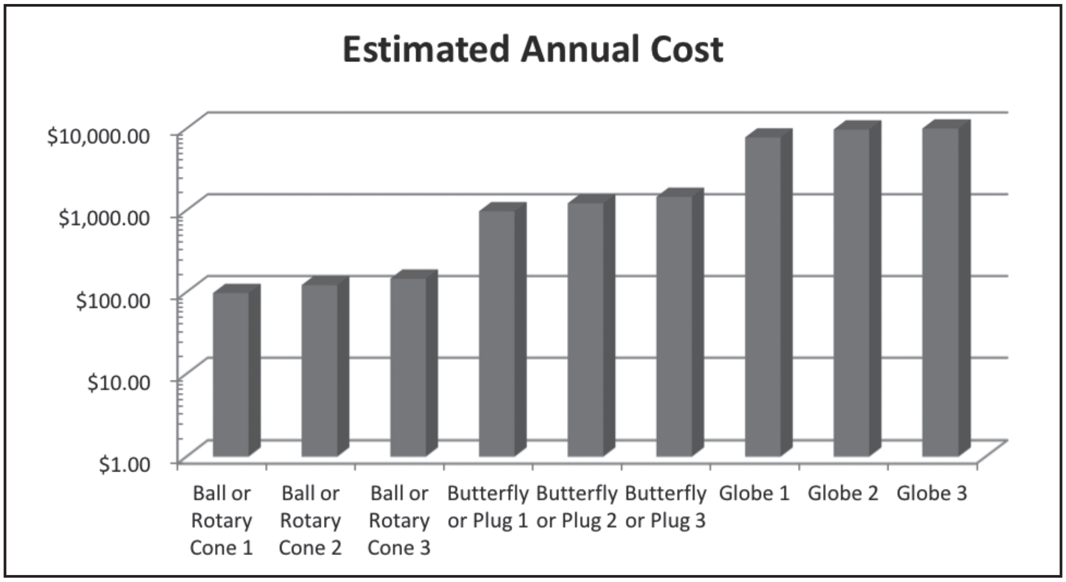

Figure 2-9 Comparative operating costs by valve type

set is 80% (0.8, decimal/fraction), and the cost of electricity is $0.09 per kWh. The cost of the valves' head loss is computed using Eq 2-22. A 24-in. (600-mm) valve in the example can consume from $97 to almost $12,000 in annual electricity, assuming water (S_g = 1.0) is pumped through the valve 50% (0.5, decimal/fraction) of the time. Table 2-4 provides other examples.

$$A_{COST} = \frac{1.65 \times Q \times \Delta H \times S_g \times C \times U}{E} \tag{2-27}$$

This example is theoretically correct given that usage, flow rate, and the pump efficiency are assumed constant. In operation, the lower loss valves allow greater pumped flow at the same system operating system pressure; the pump runs farther out on the pump curve and may even require the pump to draw a different (slightly lower or greater) current during operation. The savings is truly seen in the usage (running time reduction) and increased pump efficiency to deliver the same quantity of water (or inventory). The savings will be measured in the pump usage reduction over time and reduced number of pumps operating at the same time to produce the system demand.

As presented in Figure 2-9, it is noted that the greatest energy cost change is between valve types and is not as dramatic between the relative resistances of similar valve models, classes, and types. This is not to say that globe valves in the example should not be used in pump control service as they offer different engineering performance variables, including a wider flow control range, cavitation trim variations, different inherent flow characterizations, and transient control solutions, but to point out that there is an energy cost associated with that performance difference. When looking at similar design valves, the savings between individual product models, pressure classes, or suppliers is generally insignificant. For these low-loss quarter-turn valves, the difference between various suppliers is generally less than that of a single elbow or even the difference between short- and long-radius elbow.

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Chapter 3

Valve Torque

In a quarter-turn valve, valve torque is the turning effort needed to rotate the valve's closure member (ball, cone, disc, or plug) or hold it in position. Torque varies with system conditions, valve design, and closure member position. The methodology given in this manual is a step-by-step procedure for predicting valve-operating torque and represents the current method used by many quarter-turn valve manufacturers in the water industry.

Rotary cone valves are unique in their seating and unseating as they change from torque and rotation to lift, thrust, and axial movement. This involves the development of a series of thrust and screw torque-to-thrust conversion equations that are not currently included in this manual. Therefore, seating and unseating of the rotary cone valve is not covered until the linear-valve manual of standard practice is available.

DISCUSSION OF TORQUE CALCULATIONS

The torque calculations are broken into 10 separate torque components, and each is derived from a first-principles approach. The 10 separate torque components are classified into two categories: (1) passive or friction-based and (2) active or dynamically generated. These 10 components are listed in Table 3-1.

Each of these separate torque components is evaluated mathematically from a first-principles approach, and their equations are presented, except for the buoyancy torque (item 7) and the thrust bearing torque (item 5). These two are generally considered as negligible for this scope of the valves. The components of hub seal friction torque (item 3), weight and center of gravity torque (item 6), lateral offset or eccentricity torque (item 8), and hydrostatic unbalance torque (item 10) may not be applicable depending on the valve design and installation variables. Also, packing friction and hub seal friction may be considered as a single torque. Seating (and/or unseating) friction torque (item 1), packing friction torque (item 2), bearing friction torque (item 4), and dynamic or fluid dynamic torque (item 9) should always be included in operating torque calculations for ball, butterfly, and plug valves. The lift, thrust, and torque-to-thrust conversions for the seating and unseating of rotary cone valves are not evaluated, but all other mid-stroke torque requirements should be evaluated.

Table 3-1 Torque Component Category

Item No.	Torque Component	Torque Category
1	Seating (and/or unseating) friction torque	Passive or friction-based
2	Packing friction torque	
3	Hub seal friction torque	
4	Bearing friction torque	
5	Thrust bearing friction torque	
6	Weight and center of gravity torque	Active or dynamically generated
7	Buoyancy torque	
8	Lateral offset or eccentricity torque	
9	Dynamic or fluid dynamic torque	
10	Hydrostatic unbalance torque	

The passive torque components are friction-related and, in general, are either constant for a given valve or directly dependent on the differential pressure. These components always oppose actuator motion and are generally considered to be essentially the same magnitude in either direction of operation (opening or closing) except for seating and unseating. Seating and unseating torque may be evaluated separately or considered the same when differences are small.

The active or dynamic torque components are generated in the valve by the effects of the internal fluid media (water) or gravity acting on the valve. These components may oppose or assist the actuator's operation. Because dynamic torque generally tends to close the valve, the actuator may act as a brake to control the speed of the closing stroke but must also overcome this torque in the opening stroke.

The separate actuating torque methodology provided here is generally used for valves of larger sizes. Actuator sizing in valves 12 in. (300 mm) and smaller is driven primarily by the passive/friction-based torque requirements as the active torque components are a small fraction of the total required operating torque. The transition point at which the dynamically generated torque components become the major part of the total required torque depends on many factors of the valve design. However, it can be generally stated that this transition occurs in the 14-in. (350-mm) to 30-in. (900-mm) range for this scope. Actuator sizing for valves larger than 30 in. (900 mm) is often significantly based on the dynamic flow conditions.

On the basis of this and the fact that the smaller-sized valves are easily tested and grouped into a smaller range of required actuator torque over the full span of the design pressure and flow rate, this complex calculation methodology may be replaced by a simple calculation based on size and pressure using curve-fitting techniques of test data. In the smaller sizes, the manufacturer may provide curve-fit equations, graphical, or tabulated information.

This separate effects methodology becomes increasingly important in the larger valve sizes, say ≈18 in. (450 mm) and larger, and at very high fluid line velocities (greater than 16 ft/s or 4.9 m/s). It is economically or physically infeasible to test many large-diameter valves, and using separate effects calculations, model test data, and grouping of the test data is necessary. The modeling techniques of dimensionless coefficients, hydraulic similitude, grouping, interpolation, and extrapolation are not discussed in this manual.

The friction-based torque components are either constant or related to the valve diameter to the second power (D^2). Because the dynamic torque component is a function of the valve diameter to the third power (D^3), it becomes the major torque affecting the actuator sizing of the larger valves. This is why the maximum operating flow rate or fluid line velocity is needed for actuator sizing of larger valve sizes.

The hydrostatic unbalance torque is also of great importance (if it exists) to larger valve sizes as it is a function of the diameter to the fourth power (D^4) although it can be ignored as insignificant in valves ≈36 in. (900 mm) and smaller. It is seldom seen under actual operating conditions, but its influence can be very significant in valve sizes larger than 36 in. (900 mm) when it is present.

This separate effects methodology is then best applicable for determining the required actuator torque for the larger sizes of valves when the torque components are determined individually (rather than by curve-fitting techniques of the total torque) as the combination of both the operating shutoff differential pressure and maximum operating flow rate (or line velocity) has a significant effect on results.

The methods of calculating the required actuator torque, system flow, pressure drop, and cavitation indices described herein are applicable to most quarter-turn valves. The test-developed coefficients must be specific for the valve type and design, but the first-principles methodology is basically unchanged.

The flow and torque coefficient data (K_v or C_v and C_t) used must be from the same valve design and same test procedure. Care should be taken not to use the flow coefficients from one valve design and apply them to another design or mix flow and torque coefficients. In other words, the flow and torque coefficients should be a matching set of data.

DEFINITIONS

The methodology that follows is based on several terms and concepts that are defined in this section.

Torque Coefficients, C_t and $C_{t\theta}$

Torque coefficients are developed on the installation of a valve in a straight run of pipe without upstream or downstream flow disturbances, such as nearby elbows, tees, or increasers. The effects of these pipe-fittings are beyond the scope of this manual, and such conditions should be brought to the attention of the valve manufacturer. Specific installation guidelines are given in chapter 6. Manufacturers may consider torque coefficients to be proprietary information.

Bearing Torque, T_b and $T_{b\theta}$

Bearing torque calculations are dependent on the valve closure member and shaft diameters and the coefficient of friction between the shaft or trunnion and the bearing materials. Minimum shaft diameters are listed in ANSI/AWWA C504-15. A methodology for sizing minimum shaft diameter is given in ANSI/AWWA C516-14 for larger valves and ANSI/AWWA C507 for ball valves. Consult the valve manufacturer's drawings and other documentation for actual diameters or take measurements.

Bearing Coefficient of Friction, C_f

The shaft bearing material supports the shaft and disc in the valve body, allowing rotation. The static coefficient of friction for the bearing and shaft or trunnion material couple is needed to calculate valve bearing friction torque. Consult the valve or bearing manufacturer or other mechanical engineering references and handbooks for typical friction coefficients.

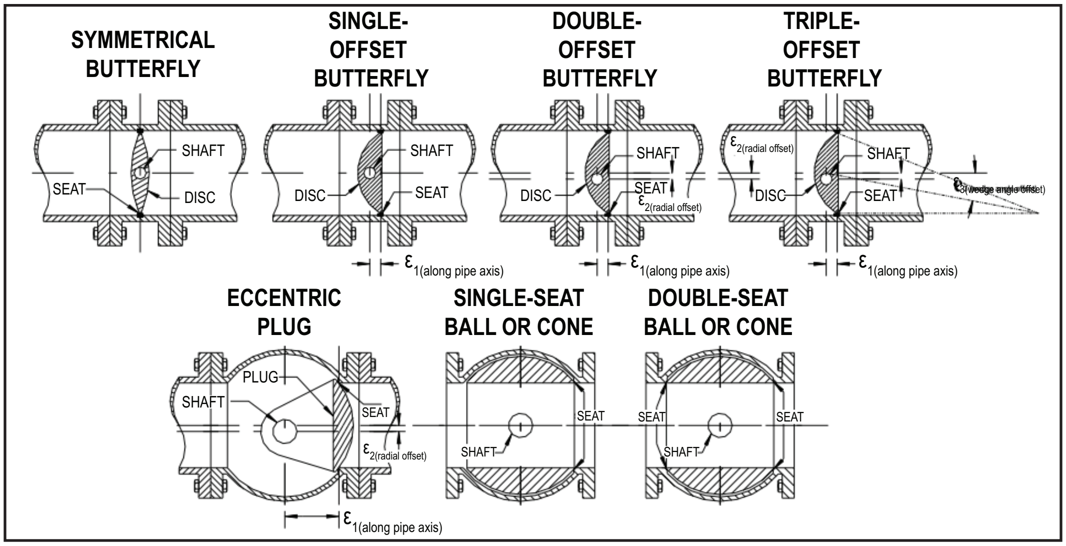

Figure 3-1 Basic closure member design geometries

Seating and Unseating Torque, T_s and T_{us}

Seat designs are as numerous as valve manufacturers, and each has its own torque characteristics. This manual presents first-principles methodology for predicting seating and unseating torque through use of seating and unseating coefficients that can be derived from tests for any type of seat. These coefficients may vary with seat material, temperature, and valve pressure rating. Seating torque and unseating torque may be considered to be the same. However, some designs, such as double- or triple-offset designs, may have separate values or coefficients for seating and unseating as well as for each flow direction. The total unseating torque is sometimes referred to as the "break torque."

Closure Member Geometry

Closure member geometry is important to a calculation of valve torque, flowing conditions, and cavitation during valve travel (see Figure 3-1). As an example, a symmetrical butterfly valve disc normally has a tendency to close caused by the flow passing across the disc. This is similar to the lift of an airplane wing. Most quarter-turn valves normally have a tendency to close at most positions but may have a tendency to open at some positions. This manual does not use the third butterfly valve offset of the triple-offset design in calculations as it is inconsequential to torque and flow equations and becomes part of the dynamic torque coefficients acquired from testing. The coefficients used in the equations should be specific to and based on the type of valve being evaluated. The offset dimensions ε_1 and ε_2 are shown in Figure 3-1.

Shaft Orientation

When a quarter-turn valve is installed in a horizontal pipe, the shaft orientation is important for calculating torque. When the valve shaft is horizontal in a horizontal pipe with one side empty, the water pressure above and below the shaft is unbalanced and tends to rotate the valve closure member (see Figure 3-2). This orientation affects the calculations of hydrostatic torque and center of gravity torque. For the center of gravity torque, the position of the center of gravity when fully open, above or below the shaft axis, determines if the center of gravity torque assists the opening or closing operation.

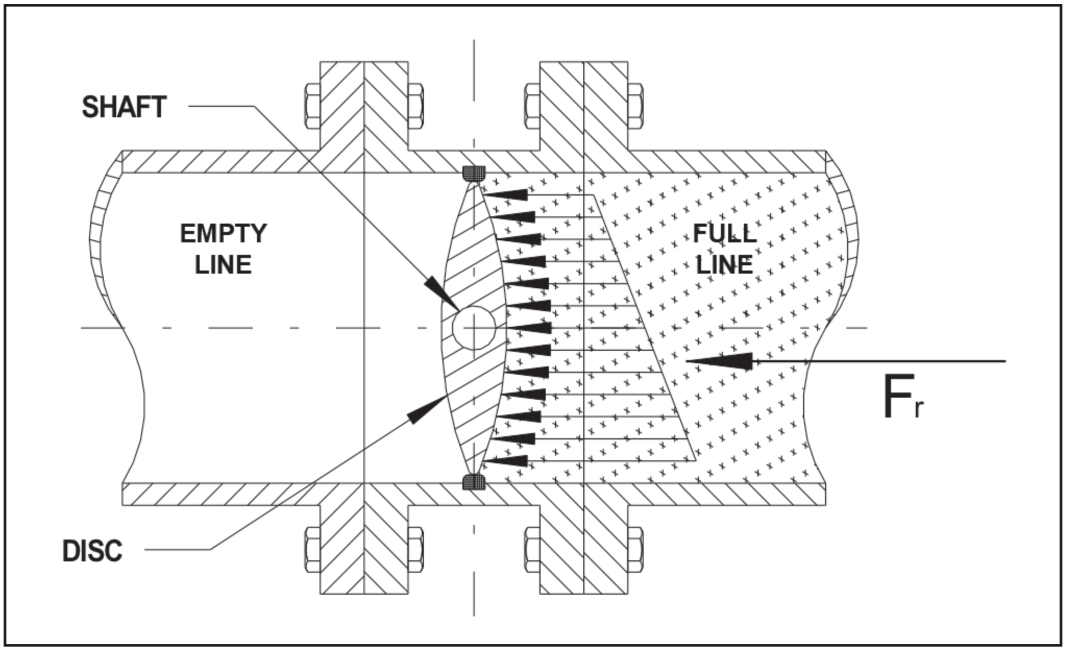

Figure 3-2 Horizontal valve shaft in a horizontal pipe

Flow Direction Through the Valve

Because many quarter-turn valves have offset closure members or other asymmetrical features, the orientation of the valve in the line with respect to flow is important. The valve may have a higher torque requirement with flow toward the shaft side of the closure member or with flow toward the other side. The manufacturer's intended valve orientation must be assured during installation (see Figure 3-3). For purposes of this manual, the flow direction is referred to as seat-side flow or reverse pressure when the seat is upstream of the valve shaft and as shaft-side flow or direct pressure when the seat is downstream of the valve shaft.

Many quarter-turn valves do not have a flow direction preference, but actuator sizing and seat tightness may be dependent on flow direction. Some single-seated ball and plug valves may have a preferred flow direction. Single-seated ball and plug valves used in pump start, stop, and check operation may have installation orientation requirements when the seat is located like a check valve (between the pump and the valve) to close tight against reverse flow rather than pump flow. In this application, the valve sees reverse pressure when operating open or closed against a running pump and direct pressure when isolating a stopped pump.

Flow in both directions should be considered in the analysis if the installation orientation is not known or if flow reversal is possible.

Torque Sign Conventions

Valve-generated active torque components (T_d, T_h, T_{cg}, and T_{ecc}) are considered as positive values when they tend to close the valve and negative when they tend to open the valve (see Figure 3-4). The signs for friction-based (passive) torque components (T_b, T_s, and T_p) are always considered as positive values because they always oppose actuator motion. Therefore, the total required actuator torque in the opening direction is the summation of all torque components, and the active torque components are subtracted in the closing direction. The most conservative approach for actuator sizing is to sum the absolute values of all torque components, but this may require a substantially oversized actuator.

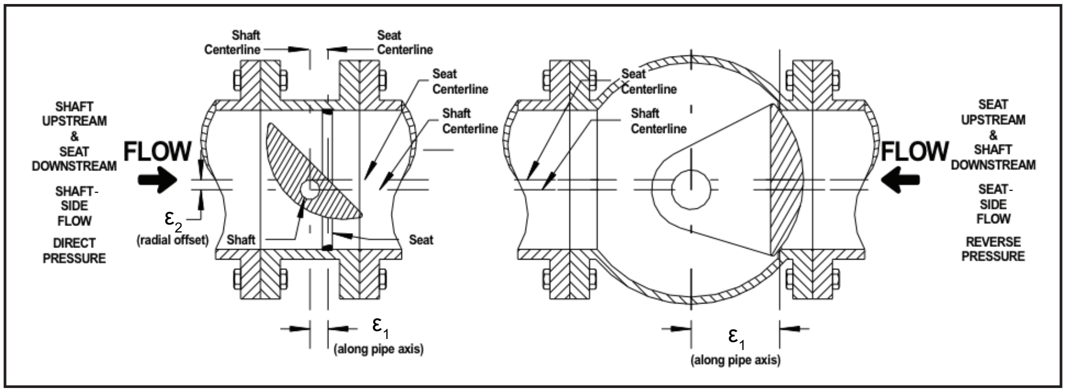

Figure 3-3 Seat-side, reverse, and shaft-side direct flow direction orientations

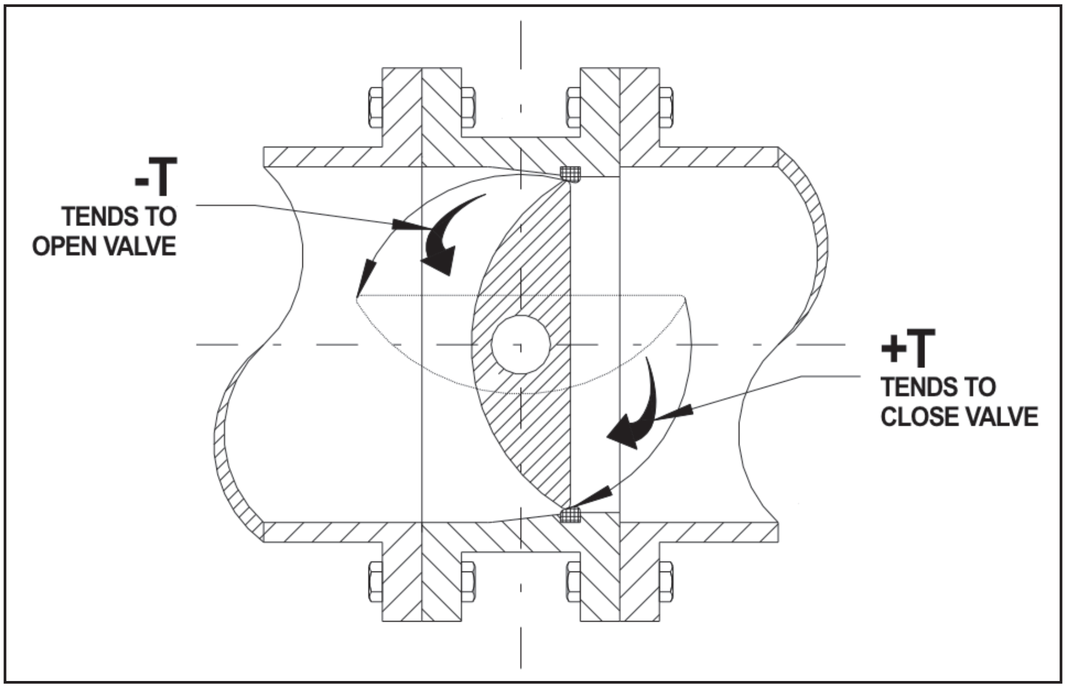

Figure 3-4 Active torque sign convention; positive values tend to close the valve

COMBINING TORQUE COMPONENTS

Quarter-turn actuating valve torque consists of separate elements that contribute to the total valve actuator torque requirement. The summations of the separate effect torque calculations given here are used to design the valve's shaft and connections as well as select the required actuator size.

It is the intention of this manual to determine the most probable actuator operating torque requirements for the condition analyzed. Data collected and used should be the normal or mean data for the valve and service application. As this method is the summation of many separately generated torque requirements, each torque component will have a possible uncertainty associated with the results. The uncertainty of any torque component may be either positive or negative at any time and is normally summed as the square root sum of the squares for the combined total uncertainty. In this manner, the use of application and/or safety factors will not be compounded.

When these calculations and summations are used to select the actuator, they are referred to as the minimum required shaft torque (MRST). Actuator sizing is based on the actuator sizing torque (AST), which includes an application factor (AF) as given in the valve or actuator standard. The AF is chosen on the basis of the valve type, actuator type, and service (on-off or modulating). The MRST is multiplied by an AF to obtain an AST (AST = MRST × AF). This is also calculated at many valve positions to correctly size the actuator. See the valve standards for the application factors to be used.

Depending on the position of the closure member, the type of valve installation, and other factors as described herein, the general equations used in computing total operating torque requirements or MRST are as follows:

Total seating torque:

$$T_{ts} = T_{b0^\circ} - T_{cg0^\circ} - T_h + T_s + T_p - T_{ecc} \tag{3-1}$$

Total unseating (break) torque:

$$T_{tus} = T_{b0^\circ} + T_{cg0^\circ} + T_h + T_{us} + T_p + T_{ecc} \tag{3-2}$$

Total opening (run) torque:

$$T_{to\theta} = T_{b\theta} + T_{cg\theta} + T_{d\theta} + T_p \tag{3-3}$$

Total closing (run) torque:

$$T_{tc\theta} = T_{b\theta} - T_{cg\theta} - T_{d\theta} + T_p \tag{3-4}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
T_{b0°	Bearing torque at valve angle 0° (always positive)	in.-lb (or ft-lb) (N-m)
$T_{b\theta}$	Bearing torque at valve angle θ (always positive)	in.-lb (or ft-lb) (N-m)
$T_{cg\theta}$	Center of gravity torque at valve angle θ (positive value tends to close the valve; negative value tends to open the valve)	in.-lb (or ft-lb) (N-m)
T_{cg0°	Center of gravity torque at valve angle 0° (positive value tends to close the valve; negative value tends to open the valve)	in.-lb (or ft-lb) (N-m)
$T_{d\theta}$	Dynamic torque at valve angle θ (positive value tends to close the valve; negative value tends to open the valve)	in.-lb (or ft-lb) (N-m)
T_{ecc}	Eccentricity torque (positive value tends to close the valve; negative value tends to open the valve) Note: Only considered at the seated position during opening or at closing	in.-lb (or ft-lb) (N-m)

Variable	Definition or Description	Units US Customary (SI metric)
T_h	Hydrostatic torque (positive value tends to close the valve; negative value tends to open the valve) Note: Only considered at the seated position during opening or at closing	in.-lb (or ft-lb) (N-m)
T_p	Packing and hub torque (always positive)	in.-lb (or ft-lb) (N-m)
T_s	Seating torque (always positive)	in.-lb (or ft-lb) (N-m)
$T_{tc\theta}$	Total closing torque at valve angle θ (positive value opposes actuator motion; negative value assists actuator motion)	in.-lb (or ft-lb) (N-m)
$T_{to\theta}$	Total opening torque at valve angle θ (positive value opposes actuator motion; negative value assists actuator motion)	in.-lb (or ft-lb) (N-m)
T_{ts}	Total seating torque (positive value opposes actuator motion; negative value assists actuator motion)	in.-lb (or ft-lb) (N-m)
T_{us}	Unseating torque (always positive)	in.-lb (or ft-lb) (N-m)
T_{tus}	Total unseating torque (positive value opposes actuator motion; negative value assists actuator motion)	in.-lb (or ft-lb) (N-m)
θ	Subscript indicating valve opening position angle, fully closed equals 0° (0 radians); fully open normally equals 90° (π/2 rad) Note: Some designs may not travel the full 90° (π/2 rad) to the full open position.	degrees (radians)

Historical Note: *Appendix A of AWWA C504, 1987 edition multiplied the bearing friction torque by a factor of 1.2. This conservatism was removed in the first edition of this document to make these calculations more characteristic of actual valve operating conditions and so that safety factors and/or allowances for degradation are not duplicated when selecting an actuator. This factor also tended to compensate for the effects of the closure member and shaft weight that were not addressed but are now considered separately in this methodology.*

The total operating torque (T_t) represents the torque or turning effort needed to rotate the closure member. The total torque is usually computed at the closed position (0°, break torque) and at 10° or smaller increments of valve position (run torque). The total torque must be calculated independently for both the opening and closing directions because some torque component signs vary with direction of rotation (See Figures 3-4 and 3-5). Hence, the computed total opening torque at a given angle will be different from the total closing torque at that same angle. Magnitude and direction of torque are essential for selecting actuators that have variable output torque characteristics (such as spring-return cylinder actuators). Actuator sizing recommendations are given in chapter 6.

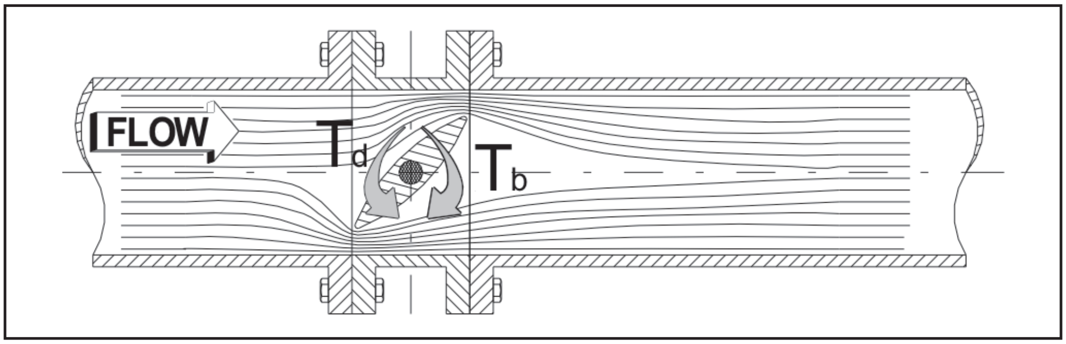

Figure 3-5 Dynamic torque (T_d) and bearing torque (T_b) during valve closure

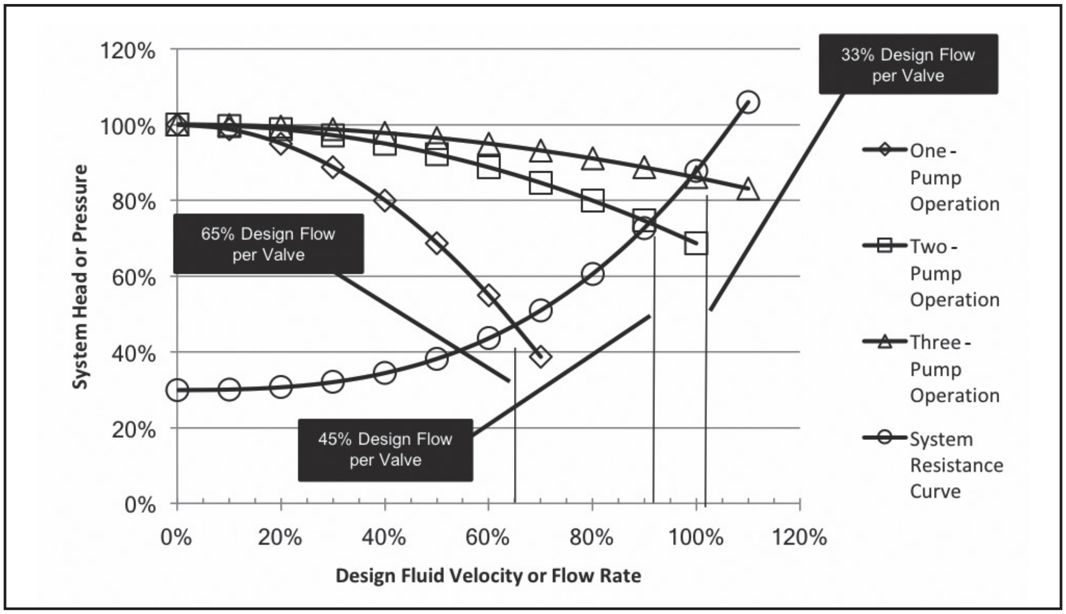

Figure 3-6 One-, two-, and three-pump system operating curves

Once the magnitude and direction of the component torques are clearly understood, other types of analyses can be performed. For example, a valve and actuator may be sized for the normal maximum system flow rate, but the same assembly must be capable of closing during a line-break flow condition (it will never need to open under line-break flow). The torque calculation at the higher flow rate in the closing direction can be used only to check the actuator size and valve torque capability. Because many valves tend to close as a result of flow, the valve and actuator may be perfectly capable of withstanding a high line-break flow rate with the actuator sized for the normal opening operating flow conditions.

Other special torque calculations include various pump combinations running in a multiple-pump application or reverse flow conditions (if applicable as valve dynamic torque often depends on flow direction). For example, when only one pump is running in a multiple-pump application (see Figures 3-6 and 3-7), the pump will run farther down on its pump curve and develop a flow rate higher than it would when the other pumps are operating. This higher flow rate will cause higher valve torques and can stall a power actuator if sized for the flow rate on the basis of all pumps running.

The individual torque components are discussed in detail in the following sections.

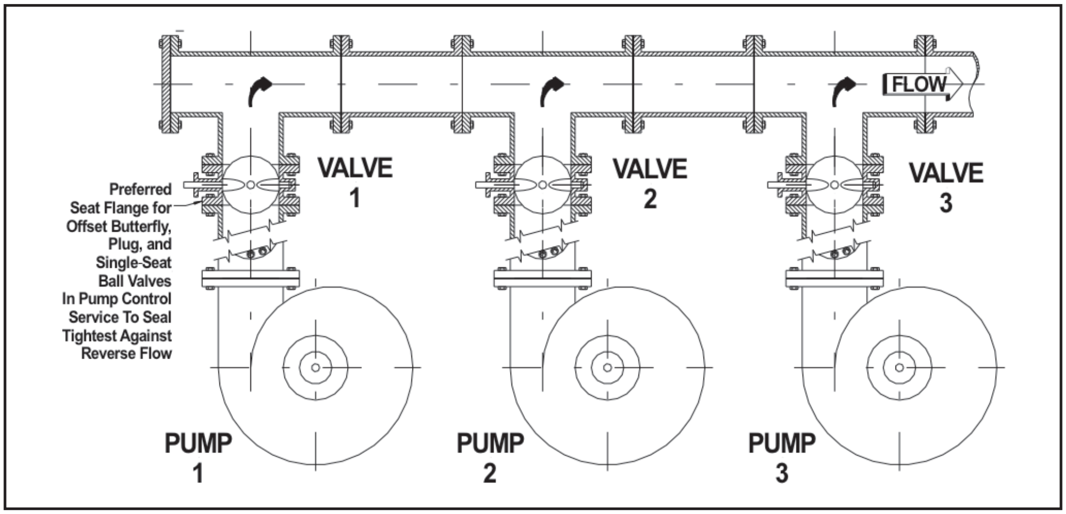

Figure 3-7 Multiple-pump installation; example shown with butterfly valves

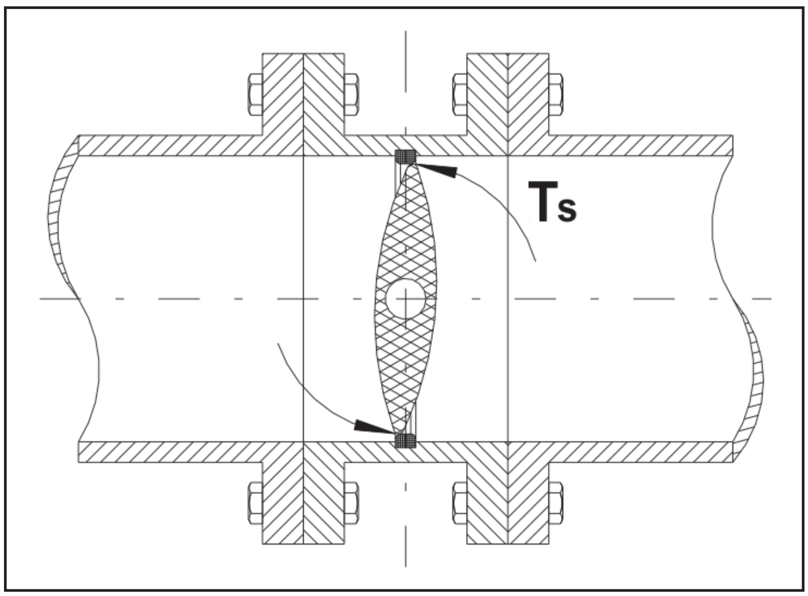

Figure 3-8 Butterfly valve seating torque

SEATING AND UNSEATING TORQUE

In symmetric and single-offset valves, the seating torque (T_s) is caused by the friction and interference between the valve's seat mating surfaces (rubber or metal) as shown in Figure 3-8. In double- and triple-offset designs, the seating torque is also based on the seat load necessary to provide the desired level of seat tightness. Seating torque is a function of many factors, including seat type, material, valve size, fluid temperature, and pressure drop across the closure member. The total effect is determined by tests. Given that all other factors are identical, seating torque is normally proportional to the square of the closure member diameter. This formula is derived from first-principles equations and integrated around the perimeter of the seat. This basic derivation is included in Appendix B.

Rotary cone valves are a unique case of a quarter-turn valve in that they are designed to seat and unseat by lifting or dropping the plug in a linear stroke direction parallel with the axis of the valve's shaft. A special link/lever mechanism provides for lifting, rotating, and dropping the plug in one smooth linear motion of an input crosshead within the actuator, which provides both torque and lift or thrust. The crosshead may be pushed or pulled by an acme screw stem threaded through it or by a hydraulic cylinder (or hydraulic/pneumatic tandem cylinder). Because of this specialized motion of its closure member, a cone valve's shaft torque can be evaluated only in the mid-travel (rotating) portion of its motion. There is no torsional load on the portion of the valve's shaft during the lifting or dropping portion of its motion at the ends of travel. The complete motion of the closure member can be evaluated when the axial resistance to lifting or dropping and the torsional resistance to rotation have each been resolved individually. As this lifting and dropping of the cone involves the thrust calculations of linear valves and the conversion of torque to the thrust of a screw stem, the seating and unseating portion of rotary cone valves is not currently covered in this manual.

Many seat designs have essentially constant and pressure-independent friction coefficients, and other seat designs are pressure-influenced or pressure-dependent, and friction (load) may increase or decrease with applied pressure. Therefore, two separate seating coefficients are given in the formula. The C_{sc} coefficient is the constant or pressure-independent seating coefficient, and the C_{sp} coefficient is the pressure-dependent seating coefficient. For a given valve design, either of these may be zero. For example, throttling valves that do not have seats do not have a seating torque, and both coefficients are zero. Valves with seats generally have a positive C_{sc} coefficient, but the C_{sp} coefficient may be zero or either a positive or negative value.

Ball and butterfly valves use Eq 3-5. Ball valves will use D or D_{port} for D_d. Ball valve test data for determining C_{sc} and C_{sp} can be based on solving the same equations using port and/or nominal diameter.

(ball, butterfly)

$$T_s = U_{C2} \times (C_{sc} + C_{sp} \times \Delta P_{max}) \times D_d^2 \qquad (3\text{-}5)$$

Some manufacturers of double or triple eccentric butterfly valves use the following seat loading methodology basis for determining the seating torque:

(double- and triple-offset butterfly)

$$T_s = U_{C2} \times D_d^2 \times W_{seat} \times K \times \Delta P_{max} \times \mu_{seat} \qquad (3\text{-}6)$$

where

Variable	Definition or Description	Units US Customary (SI metric)
C_{sc}	Constant or pressure-independent coefficient of seating torque [1]	lb/in. (N/m)
C_{sp}	Pressure-dependent coefficient of seating torque [1]	lb/in./psi (N/m/kPa)
D_d	Disc diameter (for ball valves use D or D_{port} as D_d)	in. (mm)
T_s	Seating torque (always positive)	in.-lb (or ft-lb) (N-m)

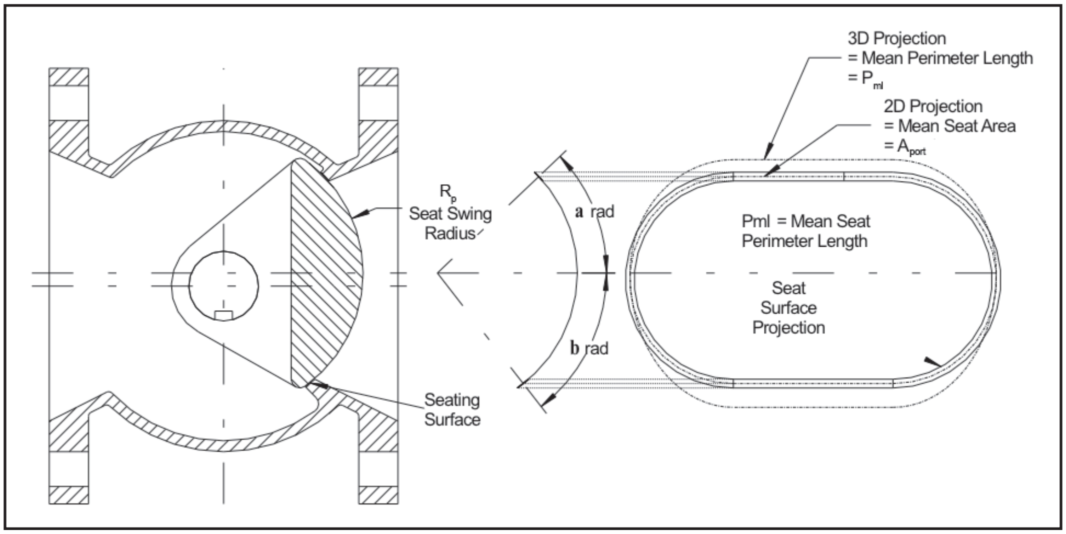

Figure 3-9 Plug valve seat perimeter and port area

Variable	Definition or Description	Units US Customary (SI metric)
U_{C2}	Units conversion factor: US customary for torque in inch pounds: U_{C2} = 1 in./in. US customary for torque in foot pounds: U_{C2} = 1⁄12 (0.0833) ft/in. Metric for torque in N-m: $U_{C2} = 1 \times 10^{-6}$ (0.000001) m²/mm²	in./in. (ft/in.) (m²/mm²)
ΔP_{MAX}	Maximum pressure drop (or loss) across the closed valve or total system with valve closed	psid (kPa-d)
W_{seat}	Seat width	in. (mm)
K	Seating compression coefficient, typically equal to two	none
μ_{seat}	Seat friction coefficient	none

[1]These variables are dependent on direct or reverse pressure direction application. For torque-seated (double- or triple-offset) valves, the coefficients may be based on the torque required to reach the required pressure and leakage rate.

For plug valves, the following equation is used:

(plug)
$$T_s = U_{C2} \times \left(C_{sc} + C_{sp} \times \Delta P_{max}\right) \times P_{ml} \times R_p \tag{3-7}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
P_{ml}	Mean seat perimeter length	in. (mm)
R_p	Swing radius of plug valve rubber covered plug face, see Figure 3-9	in. (mm)

Seating or unseating torque (T_s or T_{us}) is always positive because it opposes any obturator movement. The effects of seat cleanliness, aging, and degradation are not usually included in the seating coefficient. The test used to determine the seating coefficient (discussed in chapter 5) is based on a good-condition, well-maintained valve. Manufacturers may apply a safety factor or in-service degradation factor to the seating coefficient or the calculated seating torque to account for long-term service conditions. It should be noted that wear and aging may decrease or increase the seating torque depending on valve design, installation orientation, and service operating conditions; therefore, it is inappropriate to apply a degradation factor without good operating experience.

The pressure-dependent coefficient (C_{sp}) represents the change in torque in seat designs that are pressure-assisted or otherwise variable based on the operating differential pressure. The pressure-dependent coefficient (C_{sp}) may be positive or negative depending on installation orientation. Seat designs that are not affected by pressure differential may have a C_{sp} value of zero.

If the torque required to seat and the torque required to unseat are substantially different, separate seating coefficients and unseating coefficients can be developed in a similar manner using the following equations:

(ball, butterfly) $$T_{us} = U_{C2} \times \left(C_{usc} + C_{usp} \times \Delta P_{max}\right) \times D_d^2 \quad (3\text{-}8)$$

where

Variable	Definition or Description	Units US Customary (SI metric)
C_{usc}	Constant or pressure-independent coefficient of seating torque [1]	lb/in. (N/m)
C_{usp}	Pressure-dependent coefficient of seating torque [1]	lb/in./psi (N/m/kPa)
D_d	Disc diameter (for ball valves use D or D_{port} as D_d)	in. (mm)
T_s	Seating torque (always positive)	in.-lb (or ft-lb) (N-m)
U_{C2}	Units conversion factor: US customary for torque in in.- lb: U_{C2} = 1 in./in. US customary for torque in foot- lb: U_{C2} = 1⁄12 (0.0833) ft/in. Metric for torque in N-m: $U_{C2} = 1 \times 10^{-6}$ (0.000001) m^2/mm^2	in./in. (in./ft) (m^2/mm^2)
ΔP_{MAX}	Maximum pressure drop (or loss) across the closed valve or total system with valve closed	psid (kPa-d)

[1]These variables are dependent on direct or reverse pressure direction application. For torque seated (double- or triple-offset) valves, the coefficients may be based on the torque required to reach the required pressure and leakage rate.

For plug valves:

(plug) $$T_{us} = U_{C2} \times \left(C_{usc} + C_{usp} \times \Delta P_{max}\right) \times P_{ml} \times R_p \quad (3\text{-}9)$$

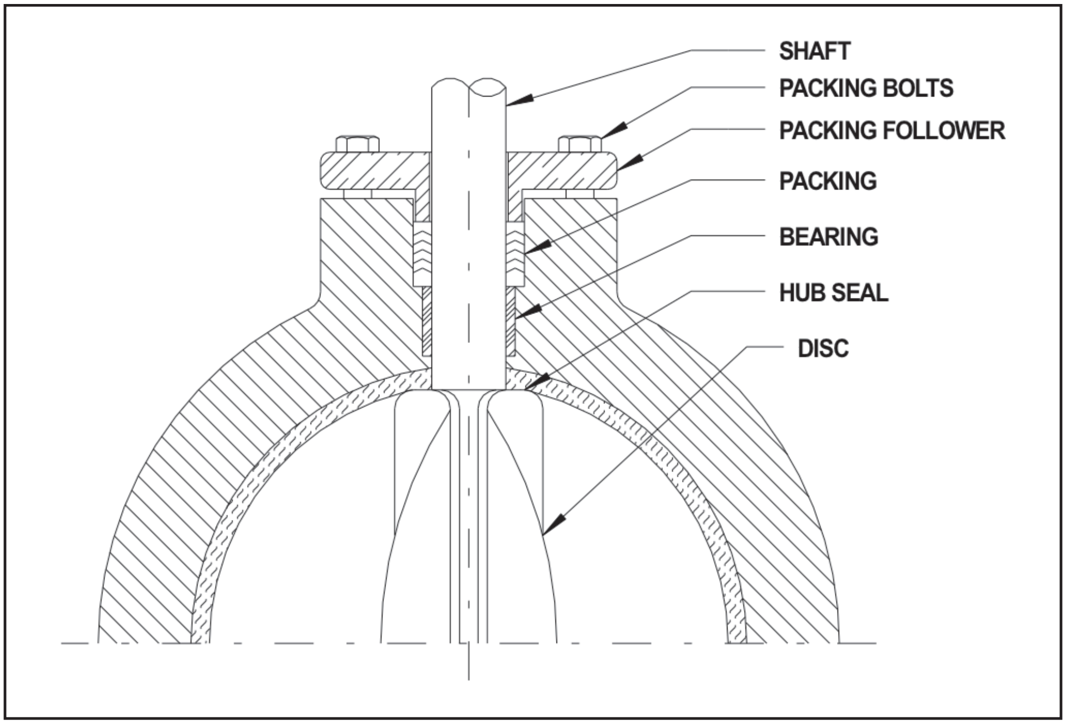

Figure 3-10 Butterfly valve packing and hub seal torque (T_p)

where

Variable	Definition or Description	Units US Customary (SI metric)
P_{ml}	Mean seat perimeter length	in. (mm)
R_p	Swing radius of plug valve rubber plug face	in. (mm)

PACKING AND HUB TORQUE

Packing torque (T_p) is caused by friction between the shaft seal (packing) and the valve shaft. The hub seal torque is caused by friction between the closure member and shaft and the body hub seal where the shaft penetrates the pressure boundary (hub) (see Figure 3-10). These are often considered as a single packing torque value. This value is frequently determined by testing and may be given by valve size, shaft diameter, a constant times the shaft diameter, or other formulation.

Packing and hub seal torque (T_p) is always positive because it opposes any closure member movement. This value is usually a small component of total torque and is often ignored on larger valves. When the shaft seal is of the O-ring type or V-packing type, this component of torque is not significant and may be ignored. However, over-tightening of the shaft packing bolts or studs can cause a significant packing torque increase. Consult the valve manufacturer for packing adjustment instructions and recommendations. In some cases, this torque may be considered as a component of seating torque or of other frictional components of torque and may be assumed to be zero.

As packing is friction based and there are many types of packing, loading methods, and adjustment procedures, there is broad variation in friction data. Many packing manufacturers provide packing friction ranges and calculation procedures based on packing type, material, method of loading, and operating pressure. Some typical packing torque calculations that may be used are the following:

$$T_p = U_{C1} \times C_{pck\ f} \times d_s^{\ 2} \tag{3-10}$$

$$T_p = U_{C1} \times C_{pck\ tq} \times d_s \tag{3-11}$$

or for chevron-type packing,

$$T_p = U_{C1} \times \frac{3 \times P_c \times \pi \times v \times H_p \times \mu_p}{4} \times d_s^{\ 2} \tag{3-12}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
$C_{pck\ f}$	Packing force coefficient	lb/in. (N/m)
$C_{pck\ tq}$	Packing torque coefficient	lb (N)
d_s	Shaft diameter	in. (m)
H_p	Packing height	in. (m)
P_c	Pressure class or maximum design pressure (the greater of)	psi (k/Pa)
T_p	Packing and hub seal torque (always positive)	
U_{C1}	Units conversion factor: US customary for torque in in.-lb: U_{C1} = 1 in./in. US customary for torque in ft-lb: U_{C1} = 1⁄12 (0.0833) in./ft Metric for torque in N-m: $U_{C1} = 1 \times 10^{-3}$ (0.001) m/mm	in./in. (in./ft) (m/mm)
μ_P	Packing coefficient of friction (typically 0.1 to 0.3)	dimensionless
ν	Packing radial stress to axial stress transfer ratio (typically assumed to be ≈0.5)	dimensionless

BEARING TORQUE

The bearing torque (T_b) in a quarter-turn valve is a function of the coefficient of friction between the bearing and the shaft or trunnion, the shaft or trunnion diameter, the closure member diameter (or area), combined closure member and shaft(s) weight, vertical orientation of the shaft axis, and the pressure drop across the closure member at each angle of rotation (see Figure 3-11).

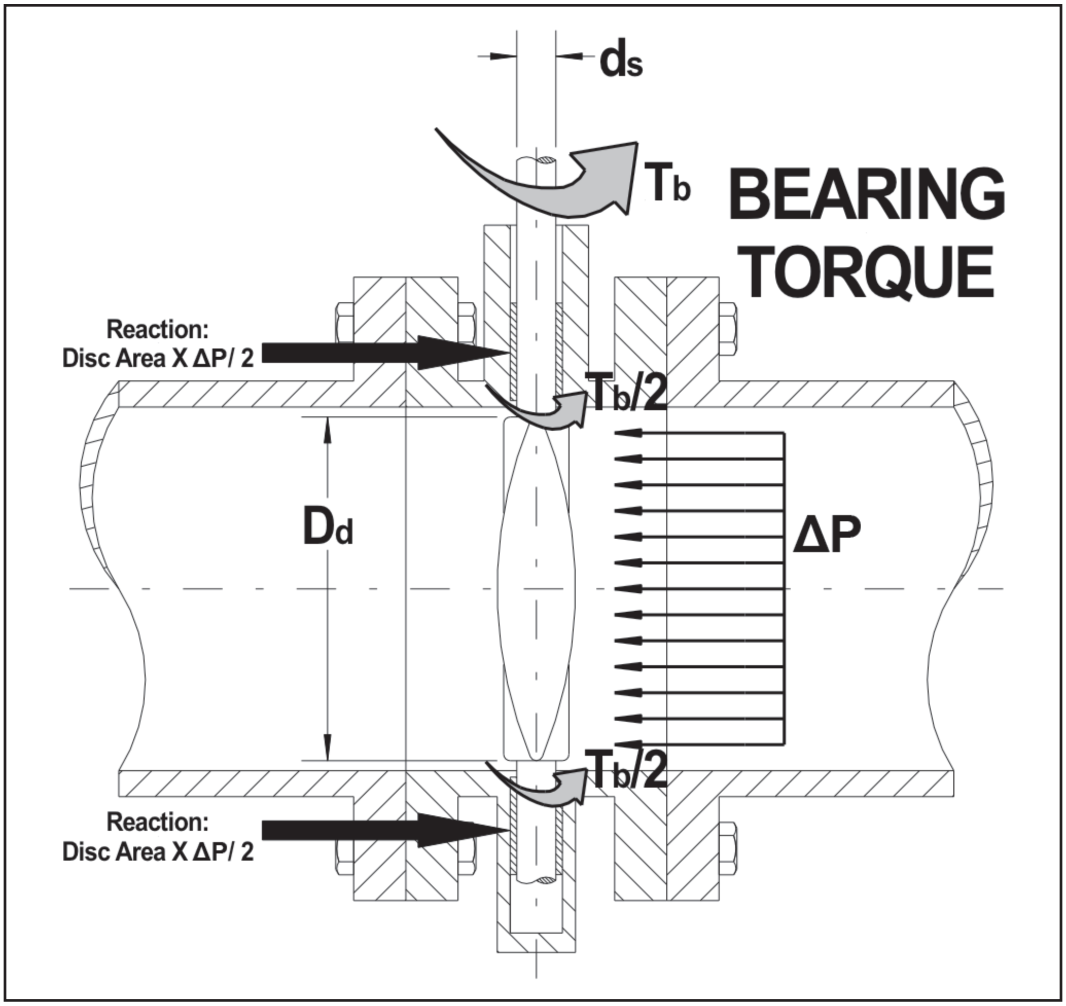

Figure 3-11 Bearing friction torque (T_b)

Ball, Butterfly, and Rotary Cone Valves

For bearing torque due to differential pressure only,

$$T_{b\theta} = U_{C2} \times \frac{\pi \times D_d^2 \times \Delta P_\theta \times d_s \times C_f}{8} \tag{3-13}$$

For bearing torque due to differential pressure plus closure member and shaft weight as a conservative direct summation:

$$T_{b\theta} = U_{C2} \times \frac{(\pi \times D_d^2 \times \Delta P_\theta + W_{d\&s}) \times d_s \times C_f}{8} \tag{3-14}$$

or

$$P_{ew} = \frac{4 \times (W_{d\&s})}{\pi \times D_d^2} \tag{3-15}$$

and

$$T_{b\theta} = U_{C2} \times \frac{\pi \times D_d^2 \times (\Delta P_\theta + P_{EW}) \times d_s \times C_f}{8} \tag{3-16}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
C_f	Coefficient of friction between the shaft or trunnion and bushing, dimensionless (this value may be obtained from a flow test, engineering handbooks, the bearing manufacturer, or the valve manufacturer)	dimensionless
D_d	Disc diameter for butterfly valves or nominal diameter (D) or port diameter (D_{port}) for ball valves	in. (mm)
d_s	Shaft diameter. For trunnion-mounted valves, use trunnion diameter (d_{tr}).	in. (mm)
d_{tr}	Trunnion diameter for trunnion-mounted obturators	in. (mm)
P_{EW}	Pressure equivalent to closure member and shaft weight	psi (kPa)
$T_{b\theta}$	Bearing torque at valve angle θ (always positive)	in.-lb (or ft-lb) (N-m)
$W_{d\&s}$	Weight of the obturator (ball, disc, or plug) and shaft(s) assembly (banjo). For applications involving additional seismic loading, $W_{d\&s}$ may be multiplied by G or (G ± 1), where G is the additional gravitational acceleration multiplier	lb (kg)
ΔP_θ	Pressure drop (or loss) while at closure member angle θ	psid (kPa-d)

Plug Valves

For bearing torque due to differential pressure only,

$$T_{b\theta} = U_{C2} \times \frac{A_{port} \times \Delta P_\theta \times d_{sm} \times C_f}{2} \tag{3-17}$$

For bearing torque due to differential pressure plus closure member and shaft weight as a conservative direct summation:

$$T_{b\theta} = U_{C2} \times \frac{\left(A_{port} \times \Delta P_\theta + W_{d\&s}\right) \times d_{sm} \times C_f}{2} \tag{3-18}$$

or

$$P_{ew} = \frac{(W_{d\&s})}{A_{port}} \tag{3-19}$$

and

$$T_{b\theta} = U_{C2} \times \frac{A_{port} \times (\Delta P_{\theta} + P_{EW}) \times d_{sm} \times C_f}{2} \tag{3-20}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
C_f	Coefficient of friction between the shaft or trunnion and bushing, dimensionless (this value may be obtained from a flow test, engineering handbooks, the bearing manufacturer, or the valve manufacturer)	dimensionless
A_{port}	Valve mean port area based on the center or mean seating point around the perimeter of the seat	in.^{2} (mm^2)
d_{sm}	Mean shaft diameter. Top or actuator shaft and the bottom or blind shaft may be different diameters.	in. (mm)
P_{EW}	Pressure equivalent to obturator and shaft weight [2]	psi (kPa)
$T_{b\theta}$	Bearing torque at valve angle θ (always positive)	in.-lb (or ft-lb) (N-m)
$W_{d\&s}$	Weight of the obturator (ball, disc, or plug) and shaft(s) assembly (banjo). For applications involving additional seismic loading, $W_{d\&s}$ may be multiplied by G or (G ±1), where G is the additional gravitational acceleration multiplier [2]	lb (kg)
ΔP_{θ}	Pressure drop (or loss) while at plug angle θ	psid (kPa-d)

[2] Eqs 3-18, 3-19, and 3-20 conservatively add the weight of the closure member and shafts directly into the bearing friction calculation. This is a worst-case approach as the bearing force load caused by the differential pressure and the force caused by the closure member and shaft(s) weight should be added as vector sums with respect to the installed orientations. The addition of this weight component to this torque is normally insignificant at higher differential pressures and is significant only at low differential pressures or when high seismic loads are required. This generally adds between 0.5 and 4.0 psi (3.45 and 27.6 kPa) to the calculated differential pressure for valves within the scope of this manual. See the section in this chapter titled "Other Components of Torque" for a more complete methodology for critical applications. For applications involving additional seismic loading, $W_{d\&s}$ may be multiplied by G (horizontal) or G ±1 (vertical), where G is the additional seismic acceleration.

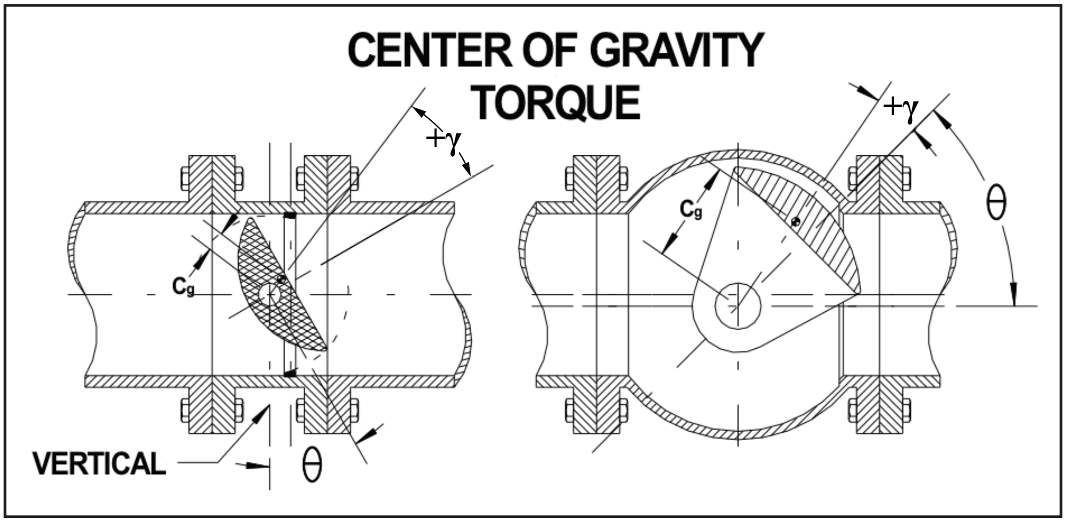

Figure 3-12 Center of gravity torque (T_{cg})

Bearing torque is always positive because it opposes any closure member movement. The value is highest at the near-closed position because of the high differential pressure when the valve is nearly closed. The bearing torque reduces to almost zero as the valve reaches the fully open position.

CENTER OF GRAVITY TORQUE

Center of gravity torque (T_{cg}) is caused by the offset center of gravity of the closure member. This torque occurs when the valve shaft is located in or near the horizontal plane, and it is a function of the closure member position and weight as well as the distance from the axis of rotation to the center of gravity (see Figure 3-12). With a horizontal valve shaft and a horizontal pipeline, this torque is greatest when the center of gravity location is situated on the pipeline axis. This torque varies considerably on the basis of closure member design and the center of gravity location. This torque may be assumed as insignificant or may be included as a worst-case constant maximum value throughout the valve travel by setting COS(θ + γ) equal to one (1) throughout the valve travel. See the section on "Other Components of Torque" and Figures 3-23 and 3-24 for more complete installation details for use in critical applications where the pipe axis and shaft axis are known to be in other orientations. Ball and rotary cone valves have zero (0) or a very small C_g, and this torque is generally not applicable to these valves.

For horizontal shaft and pipe axis, the basic equation is

$$T_{cg\theta} = U_{C1} \times S_C \times W_d \times C_g \times \mathrm{COS}\,(\theta + \gamma) \tag{3-21}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
C_g	Valve closure member center of gravity distance from shaft centerline	in. (mm)

Variable	Definition or Description	Units US Customary (SI metric)
S_c	Sign convention variable: For torque active components, +1 when the torque tends to close the valve, or –1 when the torque tends to open the valve. For center of gravity torque, the sign convention variable is +1 when the center of gravity is above the horizontal when the closure member is in the full open position or –1 when the center of gravity is below the horizontal when closure member is full open. For torque transmitted to the actuator, a positive value when valve shaft torque opposes actuator motion or a negative value when valve shaft torque assists actuator motion (actuator is acting as a brake)	dimensionless
$T_{cg\theta}$	Center of gravity torque at valve angle θ (positive value tends to close the valve; negative value tends to open the valve)	in.-lb (or ft-lb) (N-m)
W_d	Weight of valve closure member	lb (kg)
γ	Center of gravity offset angle (non-symmetric closure member designs)	degrees (radians)
θ	Valve opening position angle, closed = 0° (0 radians); full open ≈90° (π/2 radians)	degrees (radians)

HYDROSTATIC TORQUE

Hydrostatic torque (T_h) is caused by the static elevation head of the water acting against one side of the closure member when the other side of the closure member is empty (refer to Figures 3-13 and 3-25). This torque component occurs when pipe axis and shaft axis are at or near horizontal (pipe and shaft angles greater than zero). The sign convention variable may be assigned to indicate when assisting or opposing actuator motion, or it might be considered to oppose both operating directions as a worst-case assumption. This torque component only occurs when there is no flow, the valve is seated, and one side of the line is empty.

When the water pressure forces on the closure member are integrated across the circular surface, it can be replaced with a single force vector, $F_{resultant}$, as shown in Figure 3-13. Although not used in the derivation of the formula, this vector provides a clearer visualization of the effect of the fluid pressure load. The derivation of this torque is not included but is based on a circular area. Although plug valves may have a noncircular port area, this derivation is suitable for use with all valves within this standard practice.

When valve shaft and pipe axis are horizontal,

$$T_h = S_c \times U_{C1} \times \frac{\rho \times \pi}{5.333} \times \left(\frac{D_d}{12}\right)^4 \times \left(1 + \frac{8 \times \varepsilon_2}{D_d}\right) \qquad \text{US cust. units} \qquad (3\text{-}22)$$

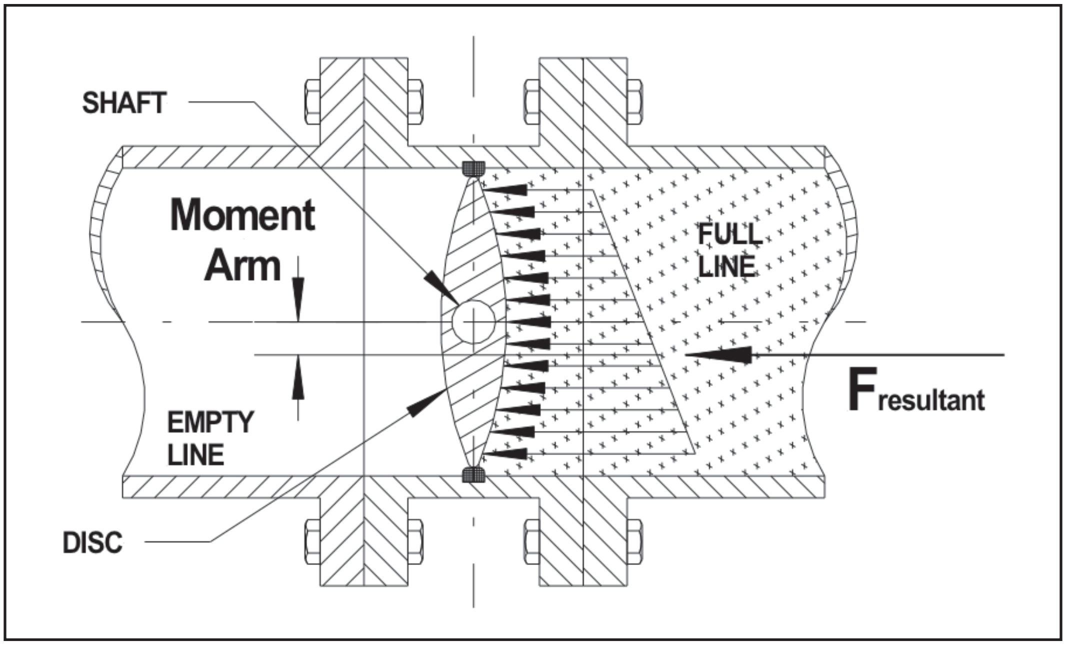

Figure 3-13 Hydrostatic torque (T_h)

or

$$T_h = S_c \times \frac{\rho \times \pi \times g}{64} \times (U_{C1} \times D_d)^4 \times \left(1 + \frac{8 \times \varepsilon_2}{D_d}\right) \qquad \text{SI (metric) units} \qquad (3\text{-}23)$$

or

For cold water where ρ = 62.4 lb/ft³,

$$T_h = S_c \times U_{C1} \times 0.0017726 \times (D_d)^4 \times \left(1 + \frac{8 \times \varepsilon_2}{D_d}\right) \qquad \text{US cust. units} \qquad (3\text{-}24)$$

For cold water where ρ = 1,000 kg/m³,

$$T_h = S_c \times 481.322 \times (U_{C1} \times D_d)^4 \times \left(1 + \frac{8 \times \varepsilon_2}{D_d}\right) \qquad \text{SI (metric) units} \qquad (3\text{-}25)$$

where

Variable	Definition or Description	Units US Customary (SI metric)
D_d	Closure member diameter, for noncircular (plug valve) ports use $D_d = (4 \times A_{port}/\pi)^{1/2}$	in. (mm)
g	Gravitational constant Acceleration due to gravity, 32.2 ft/s² (9.81 m/s²)	ft/s² (m/s²)

Variable	Definition or Description	Units US Customary (SI metric)
S_c	Sign convention variable: For torque active components, +1 when the torque tends to close the valve or –1 when the torque tends to open the valve. For center of gravity torque, the sign convention variable is +1 when the center of gravity is above the horizontal when the closure member is in the full open position or –1 when the center of gravity is below the horizontal when closure member is full open. For torque transmitted to the actuator: a positive value when valve shaft torque opposes actuator motion or negative value when valve shaft torque assists actuator motion (actuator is acting as a brake).	dimensionless
T_h	Hydrostatic torque (positive value tends to close the valve; negative value tends to open the valve) Note: Only considered at the seated position during opening or at closing	in.-lb (or ft-lb) (N-m)
U_{C1}	Units conversion factor: US customary for torque in in.-lb: U_{C1} = 1 in./in. US customary for torque in ft-lb: U_{C1} = 1/12 (0.0833) ft/in. Metric for torque in N-m: $U_{C1} = 1 \times 10^{-3}$ (0.001) m/mm	in./in. (ft./in.) (m/mm)
ε_2	Closure member lateral offset Note: ε_2 equals 0 for symmetric or single-offset closure member designs. Sign convention note: For hydrostatic torque, ε_2 is positive when oriented above the valve centerline and negative when oriented below the valve centerline. See Figure 2-3.	in. (mm)
ϱ	Fluid density Note: Standard water density is considered to be 62.4 lb/ft³	lb/ft³ (kg/m³)

DYNAMIC TORQUE

Dynamic torque (T_d) is a flow-induced torque determined to be a function of valve geometry, flow rate, and valve position (Figure 3-14).

$$T_{d\theta} = U_{C2} \times C_{t\theta} \times D_d^3 \times \Delta P_\theta \tag{3-26}$$

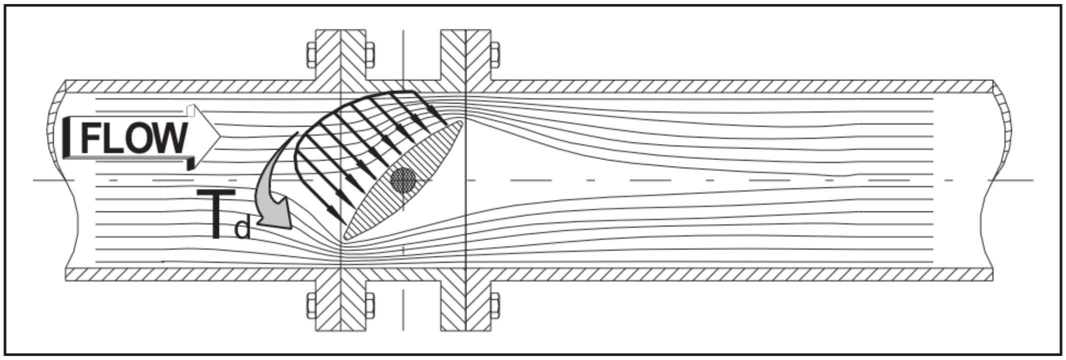

Figure 3-14 Dynamic torque (T_d) for a symmetrical disc butterfly valve

where

Variable	Definition or Description	Units US Customary (SI metric)
$C_{t\theta}$	Coefficient of dynamic torque at valve angle θ (positive value tends to close valve)	dimensionless
D_d	Disc diameter [3]	in. (mm)
$T_{d\theta}$	Dynamic torque at valve angle θ (positive value tends to close the valve; negative value tends to open the valve)	in.-lb (or ft-lb) (N-m)
U_{C2}	Units conversion factor: US customary for torque in in.-lb: U_{C2} = 1 in./in. US customary for torque in ft-lb: U_{C2} = $\frac{1}{12}$ (0.0833) ft/in. Metric for torque in N-m: $U_{C2} = 1 \times 10^{-6}$ (0.000001) m²/mm²	in./in. (ft./in.) (m²/mm²)
ΔP_θ	Pressure drop (or loss) while at closure member angle θ	psid (kPa)

[3]The nominal valve diameter (D) will be used for ball, rotary cone, and plug valves. The nominal valve diameter (D) is often used for butterfly valves when the D_d is unknown, and this tends to increase the uncertainty but produces conservatively larger torque values.

The maximum coefficient of dynamic torque (which reflects a constant pressure drop at all travel positions) occurs at approximately the 65° to 80° open position. In contrast, the maximum total dynamic torque (the summation of all operating torque requirements) normally occurs at an intermediate travel position between 0° (closed) and 50° (open) where the differential pressure is high (that is, pressure drop varies with valve position). Pressure drop and dynamic torque are dependent on the characteristics of the piping system and cannot be assumed without a system analysis.

Dynamic torque coefficients for a symmetrical closure member are normally independent of flow direction (Figure 3-15). They are functions of closure member geometry, valve travel, and pressure drop.

Dynamic torque coefficients for a single-offset butterfly valve disc, as shown in Figure 3-15, depend on flow direction through the valve as well as disc geometry, valve travel, and pressure drop across the butterfly valve disc. The dynamic torque coefficient at the open position may be negative (giving the closure member a tendency to open) when

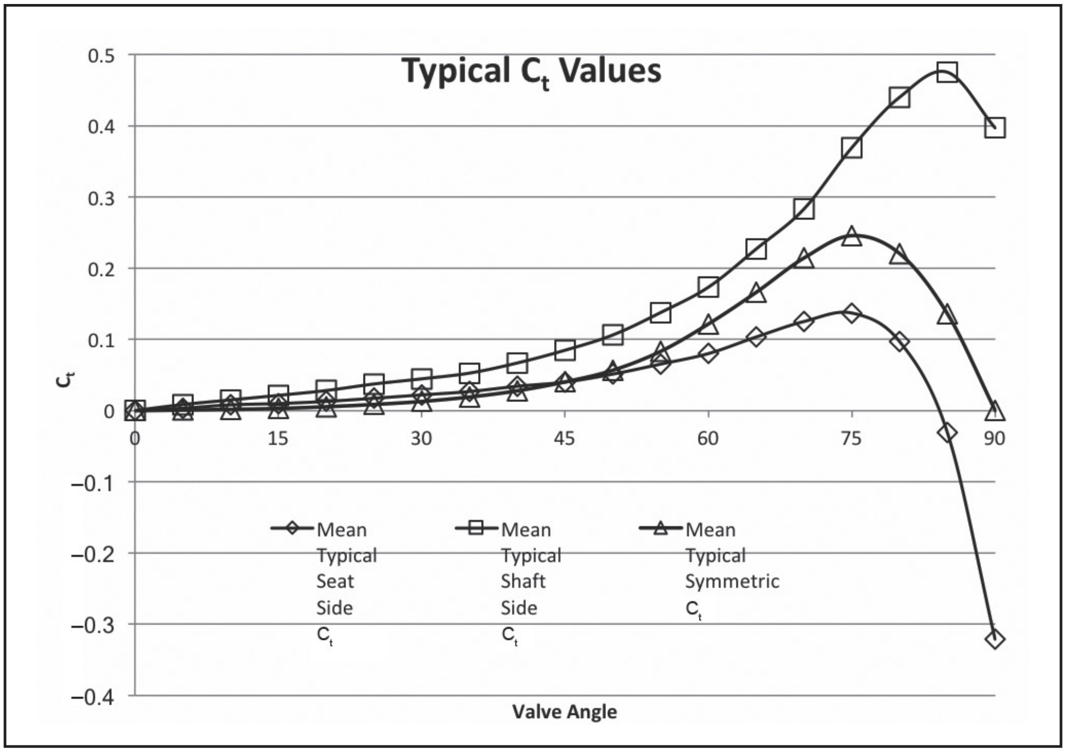

Figure 3-15 Dynamic torque coefficient (C_t) graph for butterfly valves with symmetrical and offset closure members

the valve is installed with the seat upstream. The butterfly valve single-offset disc torque coefficient can also change sign near the 85° position. If the valve is positioned at an angle at which the torque direction is unstable (when C_t crosses zero at angles other than 90°) for extended periods, fatigue damage caused by torque reversals and vibration may occur, and prolonged modulation in this valve position should be avoided.

Although the dynamic torque coefficients reach maximum at about 70° to 80° (78% to 90%) open (as shown in Figure 3-15), the total valve opening torque reaches maximum at a much lower angle (about 35° or 39%) as shown in Figures 3-17 and 3-18). Figures 3-19 and 3-20 demonstrate how the total actuator torque changes during the closing stroke. In Figure 3-16, dynamic torque is highest at ≈45° (50%) open because the pressure drop (ΔP) is an order of magnitude higher there than at 80° (90%) open, and dynamic torque is the product of both the dynamic torque coefficient and the pressure drop (see Eq 3-26).

Because dynamic torque is proportional to the closure member diameter cubed, it often becomes the largest torque on valves greater than about 20 in. (500 mm) for velocities near 16 ft/s (5.2 m/s) and above. On smaller valves, typically 6 in. (150 mm) and less, dynamic torque can be ignored, and the actuator may be sized for seating, bearing, and packing torque unless the maximum velocity exceeds 16 ft/s (5.2 m/s).

Because the dynamic torque is dependent on the pressure drop during the valve travel, the dynamic torque coefficient and the flow coefficients must be from the same valve data set and not mixed with coefficients from other valves.

SHAFT OFFSET OR ECCENTRICITY TORQUE

Shaft offset or eccentricity torque for a quarter-turn valve with a double- or triple-offset shaft is shown in Figure 3-21. This design is subject to an additional torque related to the

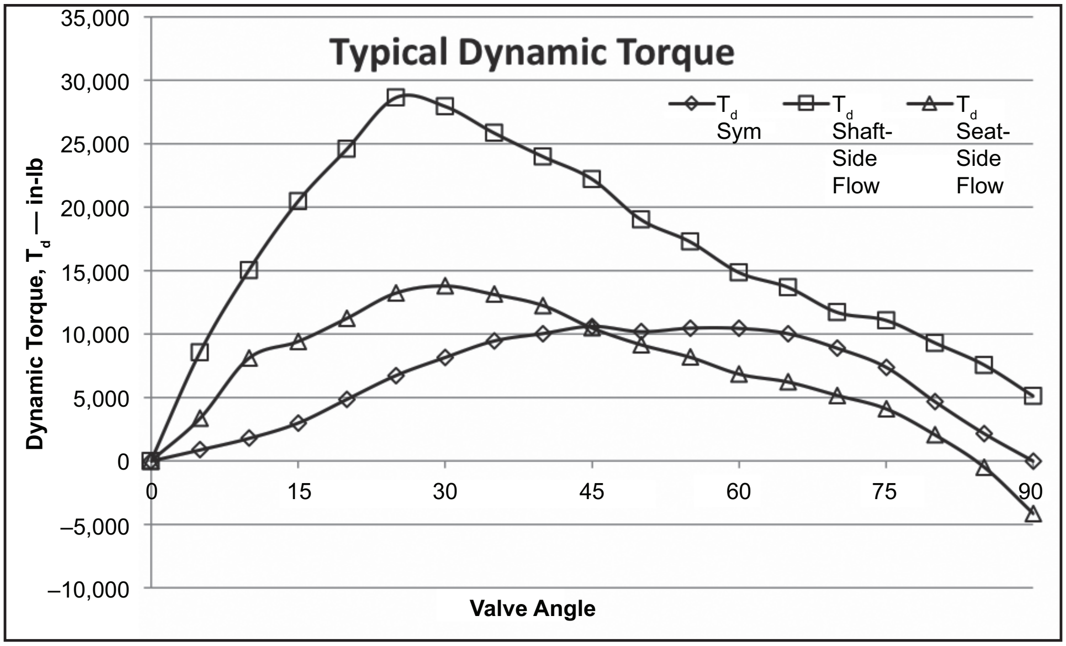

Figure 3-16 Dynamic torque (T_d) for a butterfly valve with symmetric and offset discs

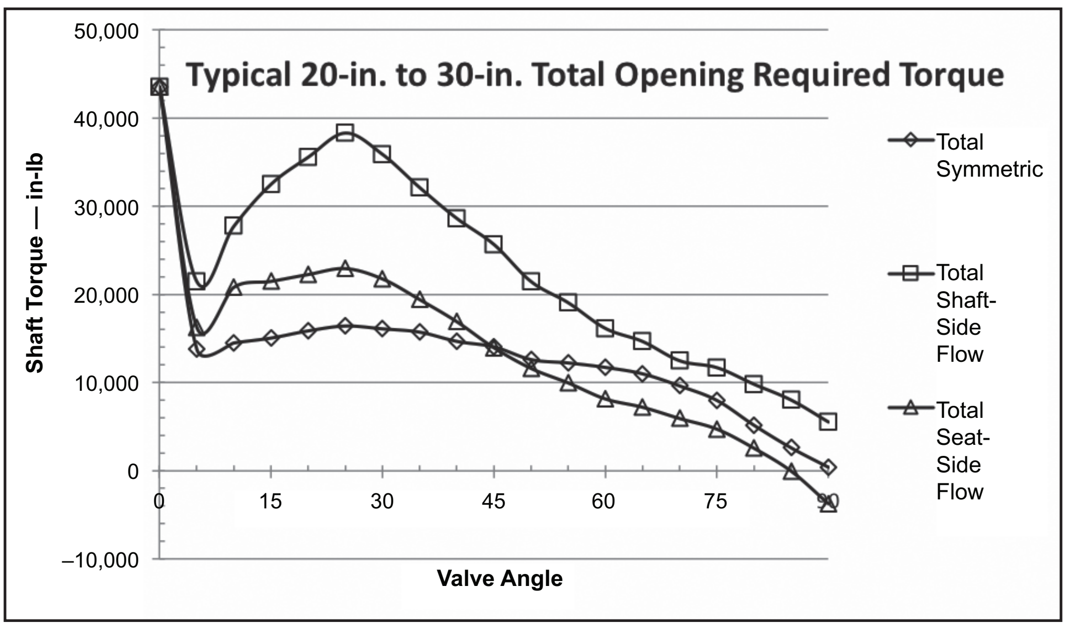

Figure 3-17 Total opening torque (T_{to}) for a 20-in. to 30-in. butterfly valve with symmetric and offset discs (in both flow directions)

lateral offset or eccentricity and the hydrostatic differential force on the closure member. This torque only applies at the full closed or seated position as it becomes a part of the dynamic torque coefficients and calculations at other closure member positions. This torque is often treated as a positive value but may be given a sign if the seating and unseating torque components are to be considered individually.

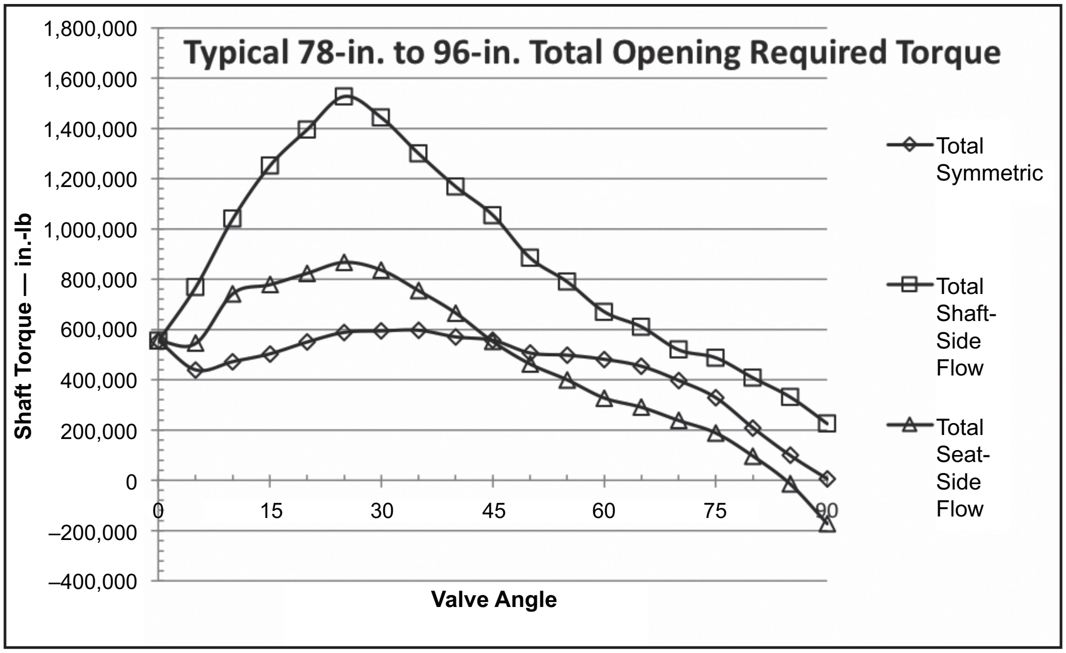

Figure 3-18 Total opening torque (T_{to}) for a 78-in. to 96-in. butterfly valve with symmetric and offset discs (in both flow directions)

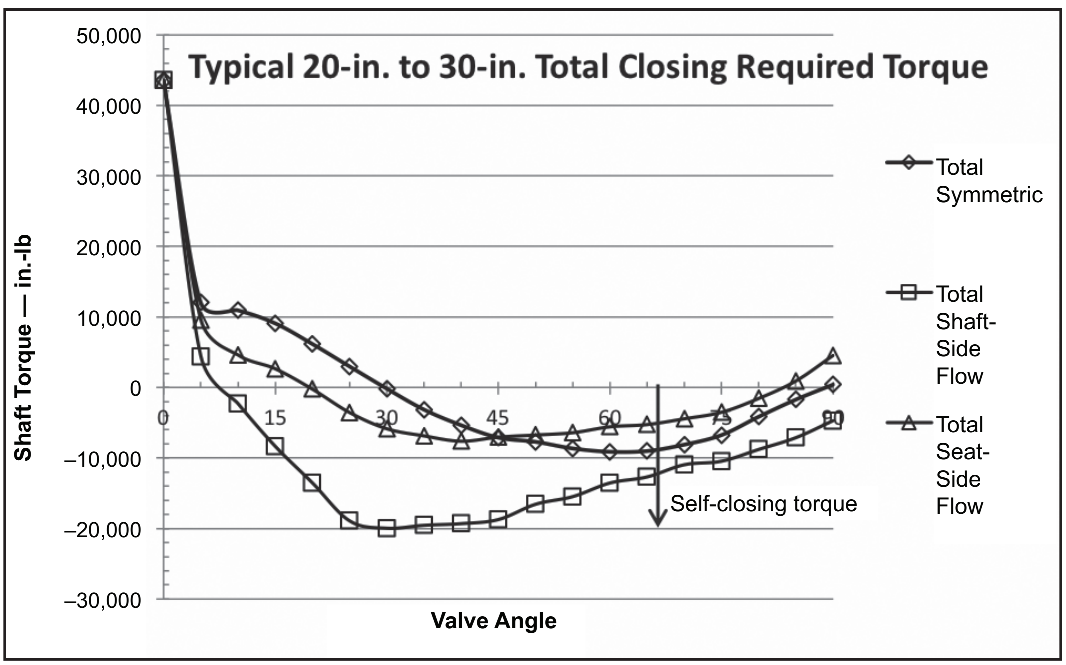

Figure 3-19 Total closing torque (T_{tc}) for a 20-in. to 30-in. butterfly valve with symmetric and offset discs

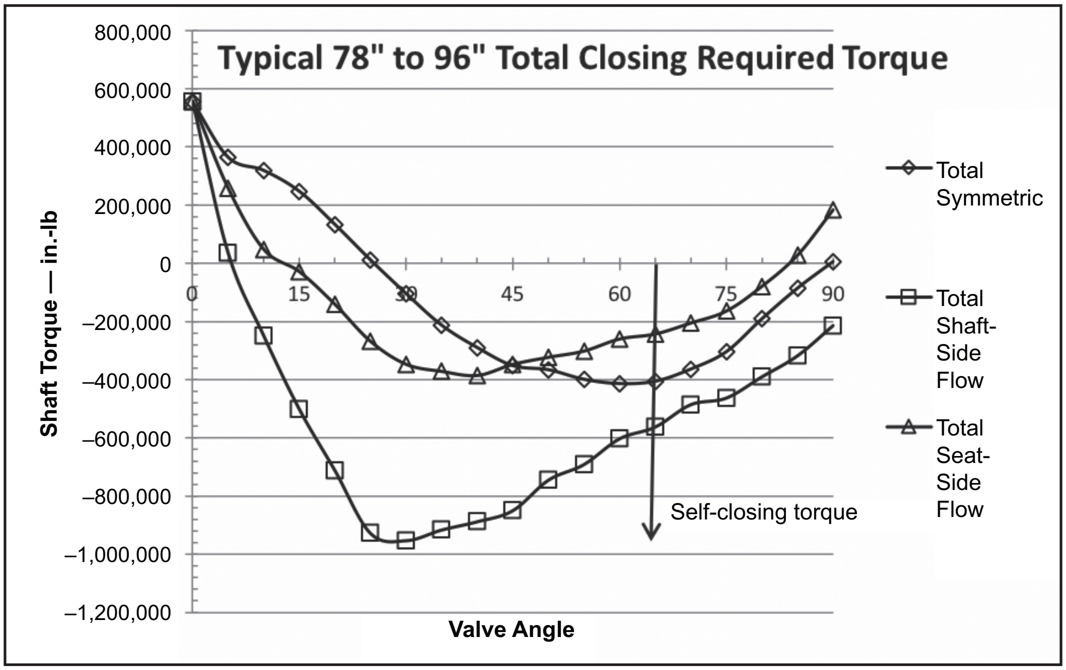

Figure 3-20 Total closing torque (T_{tc}) for a 78-in. to 96-in. butterfly valve with symmetric and offset discs

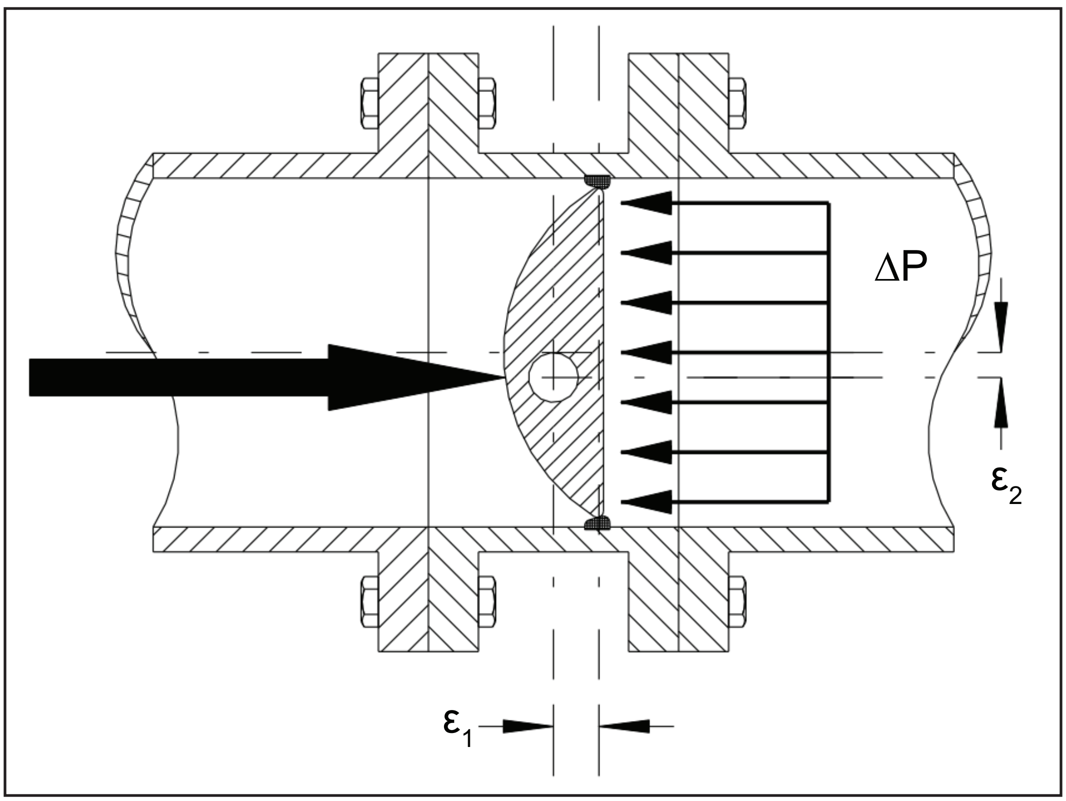

Figure 3-21 Shaft offset or eccentricity torque (T_{ecc})

(ball, butterfly, or rotary cone)
$$T_{ecc} = S_C \times U_{C2} \times \frac{\pi \times D_d^2 \times \varepsilon_2 \times \Delta P_V}{4} \tag{3-27}$$

or

(plug)
$$T_{ecc} = S_C \times U_{C2} \times A_{port} \times \varepsilon_2 \times \Delta P_V \tag{3-28}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
A_{port}	Port area of plug valve	in. (mm)
D_d	Disc diameter; use nominal diameter (D) or D_{port} for ball, rotary cone, and plug valves	in. (mm)
S_c	Sign convention variable: For torque active components, +1 when the torque tends to close the valve or –1 when the torque tends to open the valve. For center of gravity torque, the sign convention variable is +1 when the center of gravity is above the horizontal when the closure member is in the full open position or –1 when the center of gravity is below the horizontal when closure member is full open. For torque transmitted to the actuator, a positive value when valve shaft torque opposes actuator motion or negative value when valve shaft torque assists actuator motion (actuator is acting as a brake).	dimensionless
T_{ecc}	Eccentricity torque (positive value tends to close the valve; negative value tends to open the valve) Note: Only considered at the seated position during opening or at closing.	in.-lb (or ft-lb) (N-m)
U_{C2}	Units conversion factor: US customary for torque in in. lb: U_{C2} = 1 in./in. US customary for torque in ft lb: U_{C2} = 1⁄12 (0.0833) ft/in. Metric for torque in N-m: $U_{C2} = 1 \times 10^{-6}$ (0.000001) m^2/mm^2	in./in. (ft/in.) (m^2/mm^2)
ΔP_V	Pressure drop (or loss) across the valve, general form	psid (kPa-d)
ε_2	Closure member lateral offset Note: ε_2 equals 0 for symmetric or single-offset closure member designs. Sign convention note: For hydrostatic torque, ε_2 is positive when oriented above the valve centerline and negative when oriented below the valve centerline See Figure 2-3	in. (mm)

OTHER COMPONENTS OF TORQUE

The elements of torque described in the preceding sections apply to most quarter-turn valve applications. In certain designs, installations, and sizes, calculation of other torques or more exacting analysis may be needed. Detailed explanations of these methods are normally beyond the scope of this manual. Some components and methods are described here for clarification and convenience when needed. It is not generally possible to do this type of analysis during the design phase of a project as the exact details and orientations are not known or available.

Bearing Torque Caused by Closure Member and Shaft Assembly Weight, T_{bw}

The following is a more precise method for determining the bearing friction torque caused by the closure member and shaft weight. This is not normally used for simplicity but has been included for critical applications, which often involve the consideration of large seismic accelerations. This result may be added to the bearing friction torque caused by differential pressure only, Eq 3-13 or Eq 3-17.

The bearing torque from the closure member and shaft weight (T_{bw}) in a quarter-turn valve is a function of the coefficient of friction between the bearing and the shaft or trunnion, the shaft or trunnion diameter, the closure member and shaft (banjo) weight, and the orientation angle of pipe and shaft with respect to the vertical axis (see Figure 3-22).

$$T_{bw} = U_{C2} \times \frac{W_{d\&s} \times \sqrt{COS^2(\Omega) + \left[SIN(\Omega) \times SIN(\phi)\right]^2} \times C_f \times d_s}{2} \tag{3-29}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
A_{port}	Port area of plug valve	in.^{2} (mm)^2
C_f	Coefficient of friction between the shaft or trunnion and bushing, dimensionless. (This value may be obtained from a flow test, engineering handbooks, the bearing manufacturer, or the valve manufacturer.)	dimensionless
D_d	Disc diameter; use nominal diameter (D) or D_{port} for ball, rotary cone, and plug valves	in. (mm)
d_s	Shaft diameter	in. (mm)
T_{bw}	Bearing torque from closure member weight relative to installation orientation (always positive)	in.-lb (or ft-lb) (N-m)
$T_{b\theta}$	Bearing torque at valve angle θ (always positive)	in.-lb (or ft-lb) (N-m)
U_{C2}	Units conversion factor: US customary for torque in in.-lb: U_{C2} = 1 in./in. US customary for torque in-ft lb: U_{C2} = 1⁄12 (0.0833) ft/in. Metric for torque in N-m: $U_{C2} = 1 \times 10^{-6}$ (0.000001) m²/mm²	in./in. (ft/in.) (m/mm)

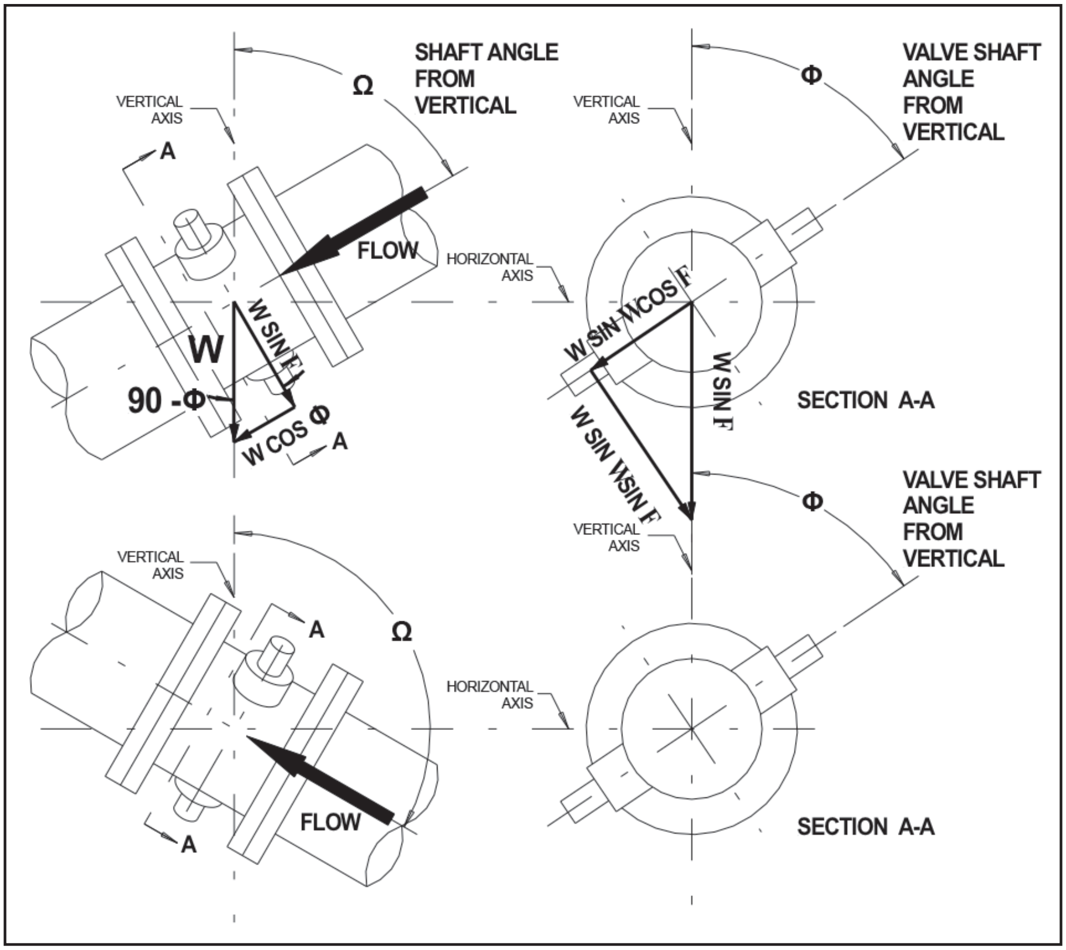

Figure 3-22 Bearing torque caused by closure member and shaft(s) weight orientation angles

Variable	Definition or Description	Units US Customary (SI metric)
$W_{d\&s}$	Weight of the closure member and shaft(s) assembly (banjo). For applications involving additional seismic loading, $W_{d\&s}$ may be multiplied by G or (G ± 1), where G is the additional gravitational acceleration multiplier.	lb (kg)
ΔP_θ	Pressure drop (or loss) while at closure member angle θ	psid (kPa-d)
φ	Valve installed shaft angle from vertical axis, 0° to 90° (or 0 to π/2 radians)	degrees (radians)
Ω	Pipe angle from vertical axis for hydrostatic and bearing torque, 0° to 90° (or 0 to π/2 radians) for flow downhill; 90° to 180° (or π/2 to π radians) for flow uphill	degrees (radians)

The vector summation with the total bearing friction torque from both the loads caused by differential pressure and closure member and shaft(s) weight is as follows:

$$T_{btotal\theta} = (T_{b\theta} + T_{bw}) \tag{3-30}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
$T_{btotal\theta}$	Total bearing torque at valve angle θ with addition of closure member and shaft(s) weight relative to installation orientation (always positive)	in.-lb (or ft-lb) (N-m)
T_{bw}	Bearing torque from closure member and shaft(s) weight relative to installation orientation (always positive)	in.-lb (or ft-lb) (N-m)
$T_{b\theta}$	Bearing torque at valve angle θ (always positive)	in.-lb (or ft-lb) (N-m)

Center of Gravity Torque for Installed Orientation, $T_{cg\theta}$

For installations in which the pipe axis and shaft axis are known to be in other orientations, the following equations may be used. This more involved method is generally used for the more critical applications and often involves the consideration of large seismic accelerations (Figures 3-23 and 3-24).

$$T_{cg\theta} = U_{C1} \times S_c \times W_d \times C_g \times \text{SIN}\,(\beta - \theta - \gamma) \times \text{SIN}\,(\varphi) \tag{3-31}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
C_g	Valve closure member center of gravity distance from shaft centerline	in. (mm)
S_c	Sign convention variable: For torque active components, +1 when the torque tends to close the valve or –1 when the torque tends to open the valve. For center of gravity torque the sign convention variable is +1 when the center of gravity is above the horizontal when the closure member is in the full open position or –1 when the center of gravity is below the horizontal when closure member is full open. For torque transmitted to the actuator, a positive value when valve shaft torque opposes actuator motion or negative value when valve shaft torque assists actuator motion (actuator is acting as a brake).	dimensionless

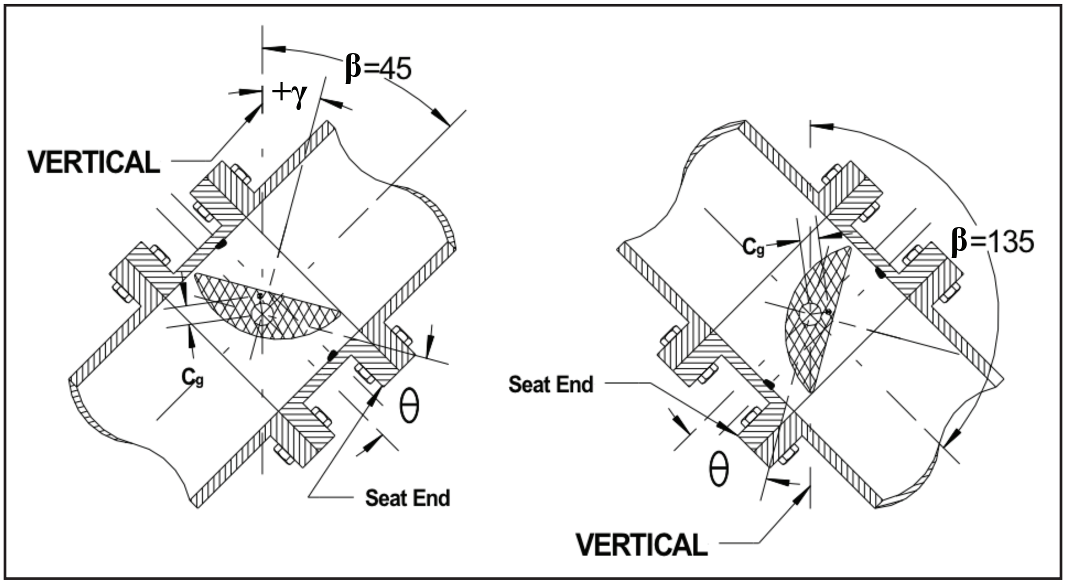

Figure 3-23 Center of gravity torque pipe angle definition

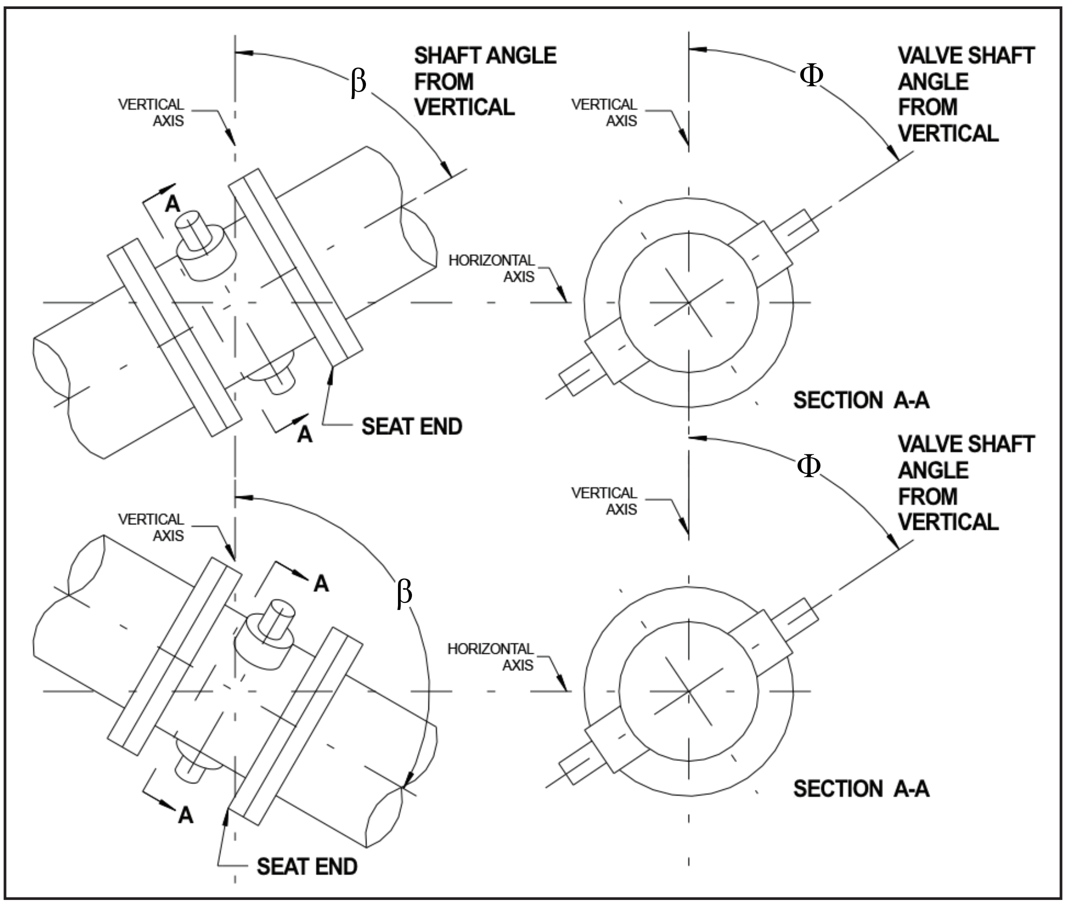

Figure 3-24 Valve shaft and pipe orientation from vertical axis for center of gravity torque

Variable	Definition or Description	Units US Customary (SI metric)
$T_{cg\theta}$	Center of gravity torque at valve angle θ (positive value tends to close the valve; negative value tends to open the valve)	in.-lb (or ft-lb) (N-m)
U_{C1}	Units conversion factor: US customary for torque in in.-lb: U_{C1} = 1 in./in. US customary for torque in ft-lb: U_{C1} = 1⁄12 (0.0833) ft/in. Metric for torque in N-m: $U_{C1} = 1 \times 10^{-3}$ (0.001) m/mm	in./in. (ft/in.) (m/mm)
W_d	Weight of valve closure member. For applications involving additional seismic loading, $W_{d\&s}$ may be multiplied by G or (G ± 1), where G is the additional gravitational acceleration multiplier.	lb (kg)
β	Pipe angle from vertical axis for center of gravity relative to seat location, 0° to 90° (or 0 to π/2 radians) when seated position is above horizontal; >90° to 180° (or π/2 to π radians) when seated position is below horizontal.	degrees (radians)
γ	Center of gravity offset angle (nonsymmetric closure member designs). May also include an adjustment for valve designs where the seating angle is not perpendicular to the pipe axis.	degrees (radians)
θ	Valve opening position angle, closed = 0° (0 radians); full open ≈ 90° (π/2 radians)	degrees (radians)
ϕ	Valve installed shaft angle from vertical axis, 0° to 90° (or 0 to π/2 radians)	degrees (radians)

Hydrostatic Torque for Installed Orientation, T_h

When the valve shaft is not vertical and the pipe axis is not horizontal but known, the following equations may be used. This more exact method is only necessary in critical analysis and requires a known installation orientation not generally available during the early design phases of a project (Figure 3-25).

$$T_h = S_c \times U_{C1} \times \frac{\rho \times \pi}{5.333} \times \left(\frac{D_d}{12}\right)^4 \times \left[\mathrm{SIN}(\phi) + \frac{8 \times \varepsilon_2}{D_d}\right] \times \mathrm{SIN}(\Omega) \quad \text{in US cust. units} \qquad (3\text{-}32)$$

For cold water where ϱ = 62.4 lb/ft³,

$$T_h = S_c \times U_{C1} \times 0.0017726 \times (D_d)^4 \times \left[\mathrm{SIN}(\phi) + \frac{8 \times \varepsilon_2}{D_d}\right] \times \mathrm{SIN}(\Omega) \quad \text{in US cust. units} \qquad (3\text{-}33)$$

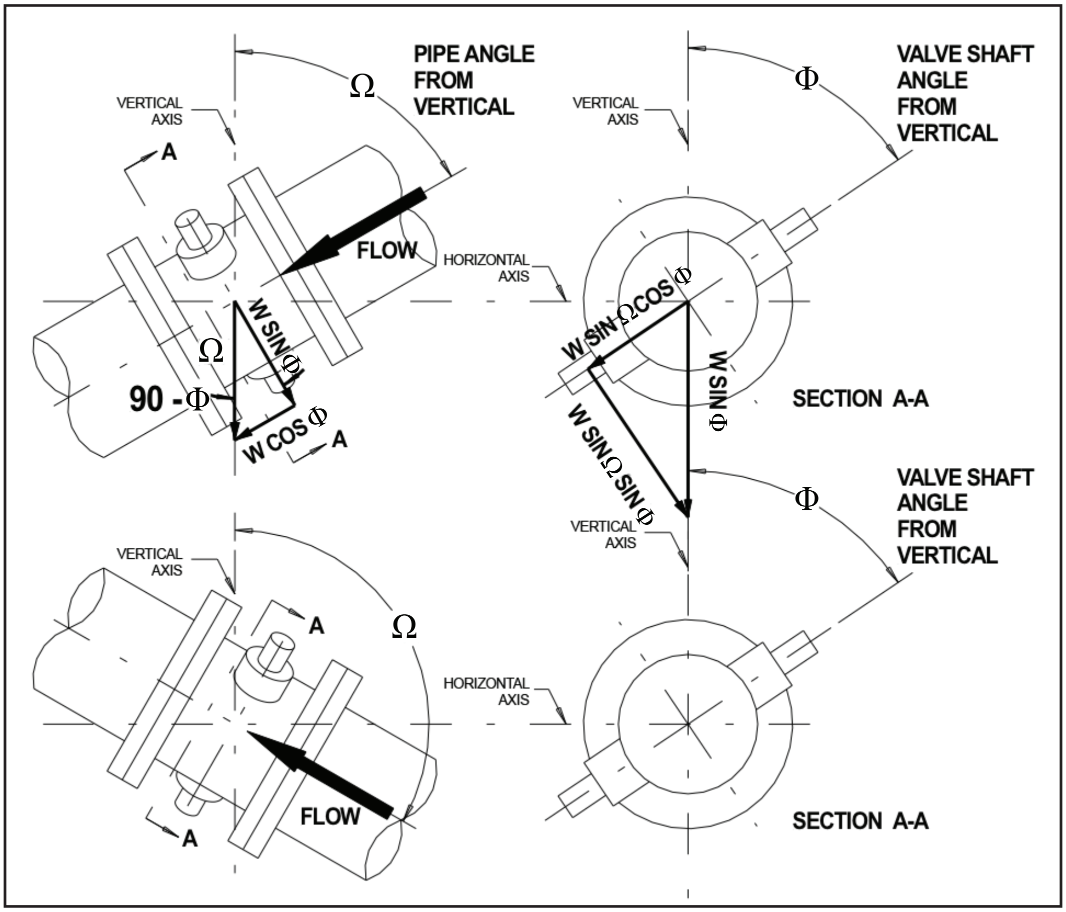

Figure 3-25 Valve shaft and pipe orientation from vertical axis for hydrostatic and bearing torque

For cold water where ϱ = 1,000 kg/m³,

$$T_h = S_c \times 481.322 \times (U_{C1} \times D_d)^4 \times \left[SIN(\phi) + \frac{8 \times \varepsilon_2}{D_d} \right] \times SIN(\Omega) \quad \text{in SI (metric) units} \qquad (3\text{-}34)$$

where

Variable	Definition or Description	Units US Customary (SI metric)
D_d	Disc diameter; use nominal diameter (D) or D_{port} for ball, rotary cone, and plug valves	in. (mm)
g	Gravitational constant Acceleration due to gravity, 32.2 ft/s² (9.81 m/s²)	ft/s² (m/s²)

Variable	Definition or Description	Units US Customary (SI metric)
S_c	Sign convention variable: For torque active components, +1 when the torque tends to close the valve or –1 when the torque tends to open the valve. For center of gravity torque, the sign convention variable is +1 when the center of gravity is above the horizontal when the closure member is in the full open position or –1 when the center of gravity is below the horizontal when closure member is full open For torque transmitted to the actuator, a positive value when valve shaft torque opposes actuator motion or negative value when valve shaft torque assists actuator motion (actuator is acting as a brake).	dimensionless
T_h	Hydrostatic torque (positive value tends to close the valve; negative value tends to open the valve). Note: Only considered at the seated position during opening or at closing.	in.-lb (or ft-lb) (N-m)
U_{C1}	Units conversion factor: US customary for torque in inch-lb: U_{C1} = 1 in./in. US customary for torque in foot-lb: U_{C1} = $\frac{1}{12}$ (0.0833) foot/inch Metric for torque in N-m: $U_{C1} = 1 \times 10^{-3}$ (0.001) m/mm	in./in. (ft/in.) (m/mm)
ε_2	Closure member lateral offset Note: ε_2 equals 0 for symmetric or single-offset closure member designs. Sign convention note: For hydrostatic torque, ε_2 is positive when oriented above the valve centerline and negative when oriented below the valve centerline See Figure 2-3	in. (mm)
ϱ	Fluid density Note: Standard water density is considered to be 62.4 lb/ft³	lb/ft³ (kg/m³)
ϕ	Valve installed shaft angle from vertical axis, 0° to 90° (or 0 to $\pi/2$ radians)	degrees (radians)
Ω	Pipe angle from vertical axis for hydrostatic and bearing torque, 0° to 90° (or 0 to $\pi/2$ radians) for flow downhill; 90° to 180° (or $\pi/2$ to π radians) for flow uphill	degrees (radians)

Note: ε_2 is equal to zero (0) for symmetric and single-offset valves. The sign of ε_2 is positive when oriented above the valve centerline and negative when oriented below the valve centerline.

If the valve shaft angle, ϕ, equals zero (shaft vertical) or pipe angle, Ω, equals zero (pipe vertical), then $T_h = 0$.

Buoyancy Torque

Another component of torque is generated by the buoyancy force of water displaced by the closure member acting vertically upward at its center of buoyancy. This torque is essentially opposite of the center of gravity torque and is generally low enough to be ignored except in very large (e.g., >60-in. [>1,500-mm]), low-pressure (e.g., <25-psig [<172-kPa]) designs with hollow closure member structures. This methodology is similar to the weight and center of gravity torque.

Thrust Bearing Torque

This component of torque is generated by frictional resistance of the valve thrust bearing, which centers the closure member axially along the shaft while supporting the closure member and shaft weight and the shaft thrust due to internal pressure (a.k.a. piston effect or shaft ejection thrust). This torque is generally negligible except in very high-pressure designs (e.g., 250-psig [1.72-MPa]) and high-friction thrust bearing designs.

Upstream Flow Disturbances and Asymmetric Flow Distributions

Unusual installations in which a butterfly valve is mounted downstream of elbows, pumps, or other valves, for example, may cause unusually high or unstable dynamic torques if the valves are not oriented properly, as chapter 6 explains. If a special orientation is required, flow tests of the actual piping configuration, or a model of it, can be conducted to determine an applicable set of flow and torque coefficients. Also, some researchers have developed approximate methods for estimating the amount of dynamic torque increase in relation to upstream elbows at various installed orientations.

SYSTEM CHARACTERISTICS

As discussed in chapter 2, system characteristics must be known to calculate the flow, ΔP, and torque for each valve position as the valve is opened or closed. This is done using the equivalent resistance system model. Pressure drop in a piping system is caused by friction losses in the pipe, valves, and other components of the system. The system flow rate increases or decreases with changes in the valve position. As the flow rate increases or decreases, the friction losses in the remainder of the system change in relation to the square of the change in flow rate. For example, reducing the flow rate by one half causes the friction loss in the pipe to decrease to one fourth of the original value. Given this relationship, the system design data and the initial flow parameters must be known in order to determine the pressure drop across the valve and the torque values as the valve position changes.

The information needed for a complete model and analysis is as follows;

Description of the System Head Source (Constant or Variable)

If system head is variable, the pump curve should be included. If the pump curve is not available, a constant head source with a source pressure equal to the closed differential pressure across the valve will be assumed.

Maximum Pressure Differential

Maximum differential pressure across the closed valve.

Maximum Design Flow Rate

Maximum design flow through the piping system through the fully open valve.

System Pressure Drop

System pressure drop at the design flow rate. If this information is not provided, a pressure drop equal to the closed pressure differential across the valve is assumed.

System Analysis and Example Calculations. Figure 2-6 shows a basic diagram of important relationships. Calculations involve the following variables:

Variable	Definition or Description	Units US Customary (SI metric)
ΔH	Head loss between any two points in a system	feet of water (meters of water)
ΔP	Pressure drop (or loss) between any two points in a system	psi (kPa)
K	Resistance coefficient	dimensionless
K_{v90}	Resistance coefficient of valve at full open (90°)	dimensionless
g	Gravitational constant (acceleration due to gravity)	ft/s² (m/s²)
D	Valve diameter	Inches (m)
V	Velocity of flow	feet per second, ft/s (meters per second, m/s)

The additional variable subscripts that are applied to variables such as C_t, T_d, T_b, ΔP, ΔH, or K are the following:

Variable	Definition or Description
sys	Subscript indicating system piping and components less the butterfly valve
v	Subscript indicating the quarter-turn valve
0 through 90	Subscript indicating the valve closure member angle

These methods are based on an equivalent resistance model as graphically shown in Figure 2-6.

Torque Calculation Methodology Example

Given the system data and calculations from chapter 2 and the additional valve data as follows:

Valve Input Data

- 24-in. AWWA class 150B butterfly valve with 24-in.-diameter single-offset disc, seat-side flow (valve is installed in a horizontal line with a vertically orientated shaft)
- $K_{v\theta}$ are listed in chapter 2, Table 2-2
- $C_{t\theta}$ from Table 3-2

- C_f for a bronze bearing = 0.25
- d_s = 3.00 in. (per ANSI/AWWA C504)
- T_p = 1,350 in.-lb
- D_d = 24.0 in.
- T_{cg} = 0
- $C_{sc} = C_{usc}$ = 16.0 lb/in.
- $C_{sp} = C_{usp}$ = 0.03 lb/in./psi
- $W_{d\&s}$ = 450 lb
- On-off electric motor AF from AWWA C504 = 1.25

1. From the inputs, T_{cg} and T_h are not applicable because the shaft is vertically oriented. T_{ecc} is not applicable as the design is a single-offset butterfly valve.
2. From chapter 2, calculate the valve differential pressures and flow rates at the valve angle (θ):
3. Calculate dynamic torque $T_{d\theta}$ at the valve angle (θ):

$$T_{d\theta} = U_{C2} \times C_{t\theta} \times D_d^3 \times \Delta P_\theta \quad (3\text{-}26)$$

4. Calculate bearing torque $T_{b\theta}$ at the valve angle (θ):

$$T_{b\theta} = U_{C2} \times \frac{(\pi \times D_d^2 \times \Delta P_\theta + W_{d\&s}) \times d_s \times C_1}{8} \quad (3\text{-}14)$$

Note: This equation is for bearing torque due to differential pressure plus disc and shaft weight as a direct summation.

5. Calculate the seating and unseating torque, T_s and T_{us}, at the valve angle (0):

(ball, butterfly) $$T_s = U_{C2} \times (C_{sc} + C_{sp} \times \Delta P_{max}) \times D_d^2 \quad (3\text{-}5)$$

6. Calculate total torque $T_{t\theta}$ at the valve angle for the opening and closing directions:

$$T_{ts} = T_{b0°} - T_{cg0°} - T_h + T_s + T_p - T_{ecc} \quad \text{at valve angle } \theta \quad (3\text{-}6)$$

$$T_{tus} = T_{b0°} + T_{cg0°} + T_h + T_{us} + T_p + T_{ecc} \quad \text{at valve angle } \theta \quad (3\text{-}7)$$

$$T_{to\theta} = T_{b\theta} + T_{cg\theta} + T_{d\theta} + T_p \quad \text{at valve angles } >\theta \quad (3\text{-}8)$$

$$T_{tc\theta} = T_{b\theta} - T_{cg\theta} - T_{d\theta} + T_p \quad \text{at valve angles } >\theta \quad (3\text{-}9)$$

Note that the T_d torque will be added in the opening direction and subtracted in the closing direction.

7. Repeat steps 2 through 6 for other valve angles.

Other examples are included in Table 3-2 and Figure 3-26.

Table 3-2 Torque Component Category

Input	Ch. 2 Calc.		Input	Calc.	Calc.	Calc.	Calc.	Calc.	Calc.	Calc.
Valve Angle θ (deg.)	Valve Loss $\Delta H_{v\theta}$ (ft)	Valve Loss $\Delta P_{v\theta}$ (psid)	Dynamic Torque Coef. $C_{t\theta}$	Dynamic Torque $T_{d\theta}$ (in.-lb)	Bearing Friction torque $T_{b\theta}$ (in.-lb)	Seating Torque T_s (in.-lb)	Packing Friction Torque $T_{p\theta}$ (in.-lb)	Total Opening Torque $T_{to\theta}$ [MRST] (in.-lb)	Total Closing Torque $T_{c\theta}$ [MRST] (in.-lb)	Actuator Sizing Torque AST (in.-lb)
90	0.9	0.41	–0.3210	–1,809	111	0	1,350	–347	3,270	4,087
80	1.2	0.54	0.0969	726	134	0	1,350	2,210	758	2,762
70	3.4	1.46	0.1250	2,519	290	0	1,350	4,159	–880	5,198
60	8.9	3.87	0.0800	4,282	699	0	1,350	6,331	–2,233	7,914
50	20.8	9.02	0.0511	6,369	1,572	0	1,350	9,290	–3,447	11,613
40	44.0	19.06	0.0341	8,984	3,275	0	1,350	13,610	–4,359	17,012
30	72.5	31.43	0.0219	9,514	5,373	0	1,350	16,237	–2,791	20,297
20	91.3	39.60	0.0128	7,006	6,759	0	1,350	15,116	1,103	18,894
10	99.0	42.90	0.0080	4,744	7,320	0	1,350	13,414	3,925	16,767
0	100.0	43.35	0.0000	0	7,396	9,965	1,350	18,711	18,711	23,389

Table Notes:

[1] The data are for the example only and have no relationship to an actual valve.

[2] Positive $C_{t\theta}$ and $T_{d\theta}$ values indicate a dynamic torque that is acting to close the valve. Negative $C_{t\theta}$ and $T_{d\theta}$ values indicate a dynamic torque that is acting to open the valve.

[3] Negative values for valve closing torque indicate that the valve is self-acting to close at those positions; thus, the valve actuator must resist this torque as a brake.

[4] MRST = minimum required shaft torque

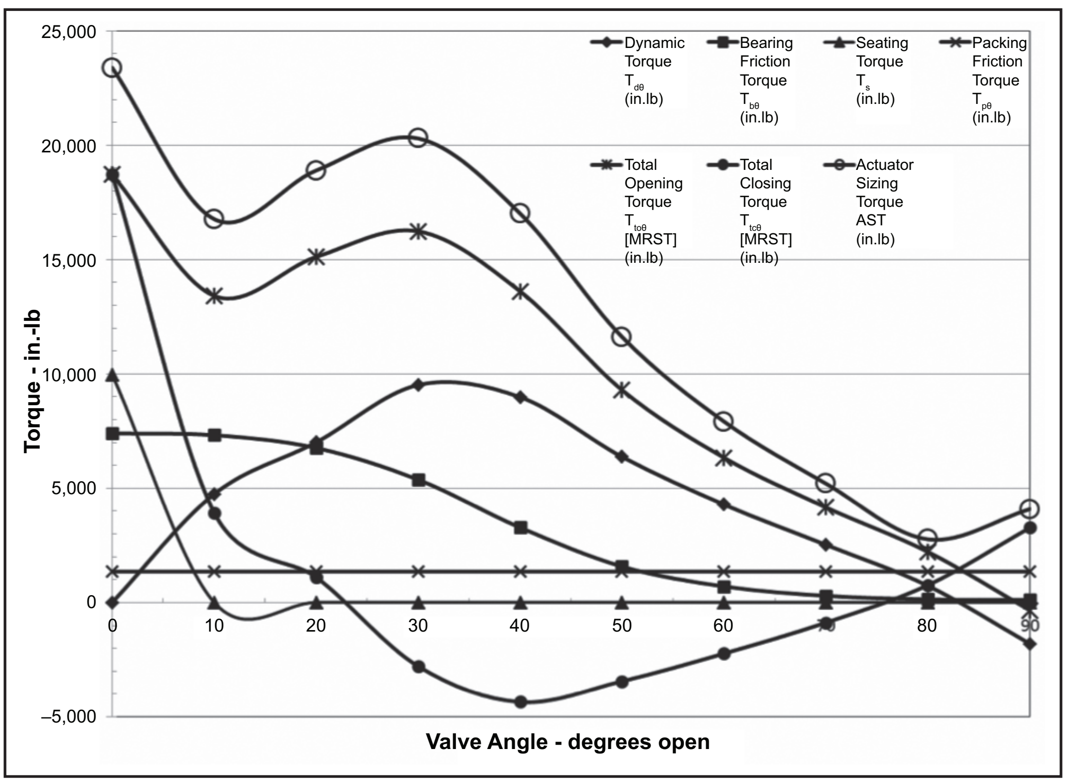

Figure 3-26 Example torque calculation summary graph

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Chapter 4

Valve Cavitation

When a quarter-turn valve is used for throttling or modulating flow rates, the operating conditions should be evaluated to determine if significant cavitation will occur. Cavitation can cause objectionable noise and vibration and can decrease the useful life of a valve and nearby piping components.

The topics in this chapter include an explanation of the conditions that cause cavitation, a method for predicting it, and a listing of methods for minimizing its effects.

DEFINITIONS

Cavitation

Cavitation is the vaporization and subsequent violent condensation of a fluid caused by localized areas of low pressure in a piping system. When water flows through a partially open quarter-turn valve, a localized low-pressure zone may occur immediately downstream of the valve closure member because of the sudden changes in flow velocity and flow separation. When the pressure in this zone falls below the vapor pressure of the fluid, the liquid vaporizes, forming a vapor pocket or vapor bubbles. As the bubbles flow downstream and the pipeline pressure recovers, the bubbles violently collapse or implode. Bubble collapse near a boundary, valve component, fitting, or pipe wall may result in pitting and material removal. Measurements have shown that localized pressures of 100,000 psi (689 MPa) can be generated by the implosion of the bubbles. These rapid implosions can produce effects varying from a popping sound to rumbling or even a deafening roar approaching 100 dB (Tullis 1989). Finally, when cavitation is fully developed, flow is restricted and no longer proportional to the square root of differential pressure.

Cavitation can form immediately downstream of the quarter-turn valve closure member where a low-pressure zone occurs. Figure 4-1 illustrates the low-pressure areas downstream of a butterfly valve disc. Cavitation bubbles can initiate near the disc edges, downstream of the body seat, in the downstream shadow of the disc, or near the downstream pipe wall. Cavitation bubbles can implode just downstream of the valve or many times the pipe diameter downstream, depending on where the pressure recovers. The

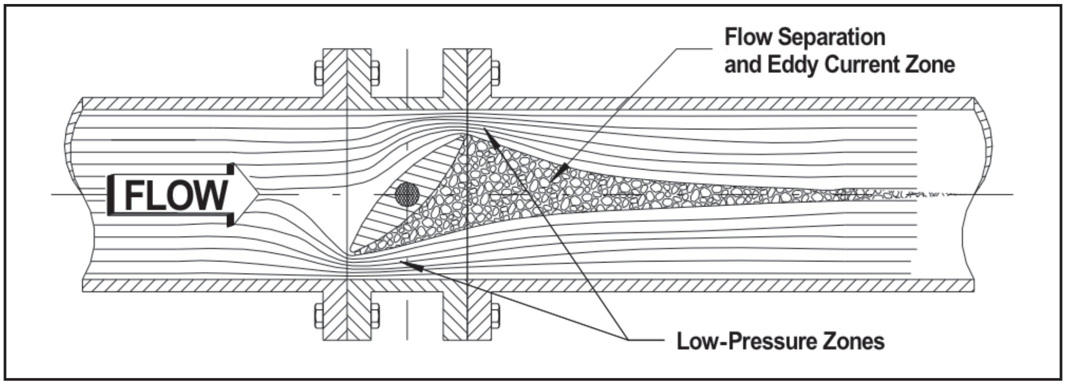


Figure 4-1 Cavitation zones downstream of a butterfly valve disc

process produces an unmistakable vibration and noise that sounds like gravel flowing through the pipe.

Many simple shutoff valve applications produce cavitation when valves are near the closed position where the differential pressure reaches its highest level. Under normal use, a shutoff valve is at a near-closed angle for a few moments before the valve is fully open or fully closed so appreciable damage to the valve or piping does not occur.

When a valve is exposed to continuous cavitation conditions in applications of flow modulation or pressure control, significant damage can occur to the metal surfaces of the valve or downstream piping in a short period of time. Hence, modulating and throttling applications warrant evaluation of cavitation conditions.

Three terms are commonly used to classify cavitation in valves according to the International Society of Automation (ISA-RP75.23-1995):

1. Incipient cavitation
2. Constant cavitation
3. Choked cavitation

The start of steady cavitation, termed incipient cavitation, can be indicated by an intermittent popping sound in the flow stream (Point A in Figures 4-2 and 4-3). Incipient cavitation typically does not cause damage or objectionably loud noise. If the pressure differential increases, however, the constant cavitation level is reached, which can be indicated by a continuous popping similar to the sound of gravel flowing through the pipe or bacon frying (Point B in Figure 4-3). Flowing conditions up to the constant cavitation condition will not cause advanced damage or degradation to the valve or piping. Continuous flow above the constant cavitation level is often accompanied by objectionable noise and valve or piping damage. Finally, the choking cavitation level (Point C in Figures 4-2 and 4-3) occurs when the valve is passing the maximum flow possible for a given upstream pressure. The vapor pocket may become extremely long, causing damage far downstream from the valve. Choking cavitation may cause a reduction in noise, but this change is usually preceded by the highest level of noise and vibration. Valves operating at the choking cavitation level usually allow short bursts of flow accompanied by high velocities and potentially high operating torques. Conditions that produce choking should be reviewed with the valve manufacturer. Reference Figure 2-1 for an overall picture of when cavitation and choking can occur in an operating system.

PREDICTING CAVITATION

Tests have shown that conditions likely to produce cavitation in a quarter-turn valve can be predicted and possibly modified or prevented. The cavitation index is typically used as a predictor of valve damage and is expressed quantitatively at each valve angle as follows (International Society of Automation, ISA-RP75.23-1995):

$$\sigma = \frac{P_u - P_v}{P_u - P_d} \tag{4-1}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
P_d	Reference downstream pressure for cavitation analysis	psi (kPa)
P_u	Reference upstream pressure for cavitation analysis	psi (kPa)
P_v	Vapor pressure adjusted for temperature and atmospheric pressure (Example: P_v = –14.4 psig (–99.6 kPa) for water at 60°F (16°C), measured at sea level.	psi (kPa)
σ	Cavitation index, general form	dimensionless
σ subscripts	Subscripts used for the σ index values include i Valve's incipient cavitation index c Valve's constant cavitation index ch Valve's choking cavitation index Operating system installed and operating index	

Note: Pressures may be gauge or absolute but must be consistent. Pressure may also be taken as feet of head as long as consistent.

The operating cavitation index can be compared to the cavitation indices for valves to predict what level of cavitation will occur (incipient, constant, or choking). It should be noted that in some earlier texts the constant index is referred to as *critical*. Later texts changed this nomenclature to *constant* to be more descriptive of the condition without implying a crucial operating condition.

A valve's cavitation indices for incipient, constant, and choked levels can be determined from flow testing in a laboratory environment. Cavitation can be observed using a hydrophone or accelerometer during the flow test. Audible detection by one with a trained ear can readily identify the incipient or constant levels of cavitation. The results of a flow test on a 6-in. (150-mm) butterfly valve at a reference test pressure of 70 psi are shown in Figure 4-4. Similar data can be prepared for any quarter-turn valve using the test methodology given in chapter 5.

Once the valve is installed into a system and operating under a set of conditions, the operating or system index ($\sigma_{operating}$) can be calculated. The lower the calculated operating cavitation index (see Eq 4-1) value is, the greater the likelihood of cavitation damage.

For example, if a valve is throttled at 45° open with a calculated system or operating index of ($\sigma_{operating}$) 6.0, then cavitation will not likely occur (referencing Figure 4-4). If, however, the valve is closed further to 30° open with a calculated system index ($\sigma_{operating}$) of 2.2, then the cavitation in the range between incipient and constant will occur. Sounds of cavitation will be heard, but serious damage will occur only after a prolonged period of time under those conditions.

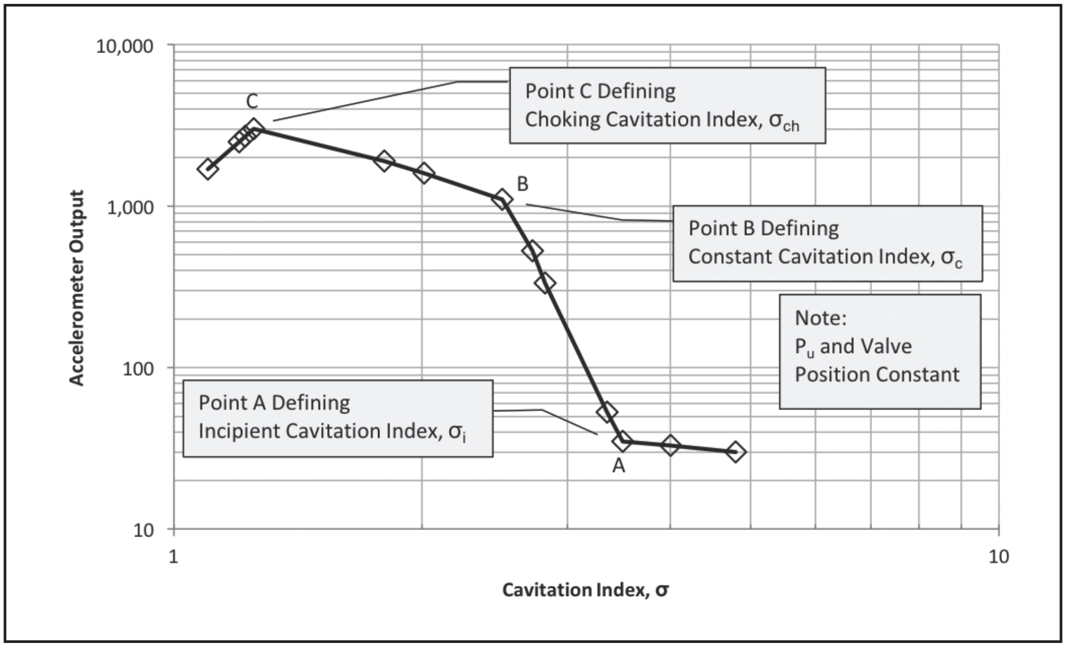


Figure 4-2 Typical cavitation index levels and acceleration readings (Tullis, 1989, p. 135)

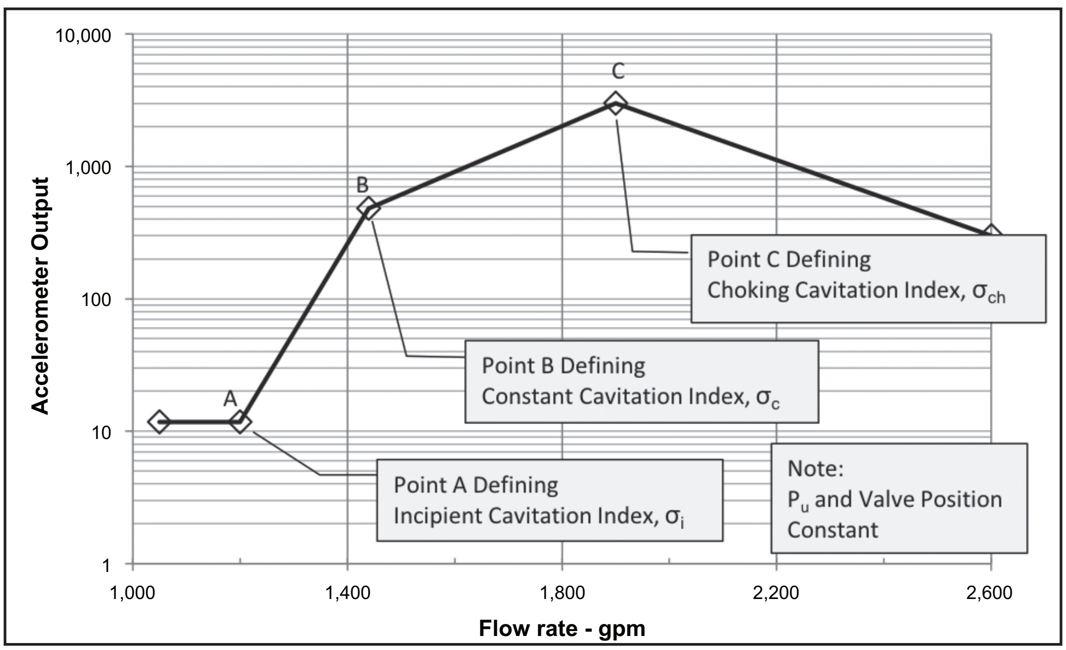
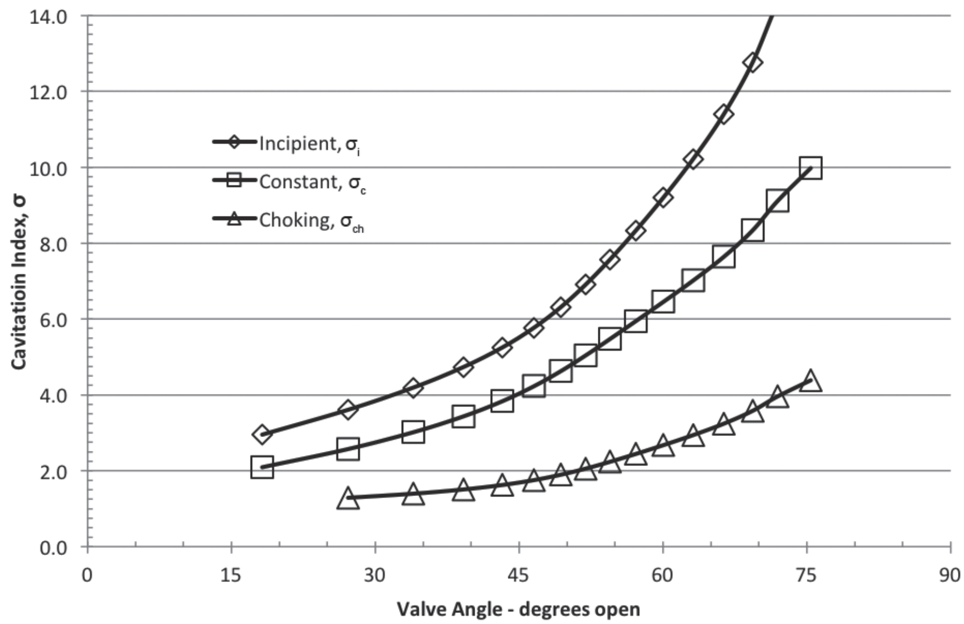


Figure 4-3 Flow rate and acceleration readings

Cavitation data are typically reported for a given valve size and upstream test pressure. Scale and pressure factors can be applied to the data to adjust coefficients from one size and pressure to another.

Incipient and constant cavitation indices (σ_i and σ_c) can be corrected for size and pressure scale effects by these equations (Tullis, 1989, pp. 144–148, and Tullis, 1993, pp. 47–55):



Note:

[1] *Critical* annotation was changed to *constant* in the figure to be more descriptive of the condition and match current practice.

[2] The data were adjusted to upstream pressure correlation instead of downstream pressure reference to match current practice and the methodology used in this manual.

[3] The abscissa (x-axis) was modified from the flow coefficient, C_d, to the corresponding valve position angle for ease of use.

[4] Based on Tullis, 1989, page 140

Figure 4-4 Typical cavitation index values for a 6-in. (150-mm) butterfly valve (Reference upstream pressure from laboratory test = P_{ut} = 70 psi, vapor pressure from laboratory test = P_{vt} = –12 psi)

$$\sigma_i = (\sigma_{it} - 1) \times PSE \times SSE + 1 \tag{4-2}$$

$$\sigma_c = (\sigma_{ct} - 1) \times PSE \times SSE + 1 \tag{4-3}$$

$$PSE = \left(\frac{P_u - P_v}{P_{ut} - P_{vt}}\right)^{0.28} \quad \text{Note [1]} \tag{4-4}$$

$$SSE = \left(\frac{D}{d_t}\right)^{Y} \quad \text{Note [2]} \tag{4-5}$$

$$Y = 0.3 \times K_v^{-0.25} \tag{4-6}$$

Notes:

[1] Results may be quite conservative for pressures above 300 psia.

[2] For SSE calculations, D is limited to ≤36 in., for valves greater than 36 in.

where

Variable	Definition or Description	Units US Customary (SI metric)
D	Nominal valve diameter	in. (mm)
d_t	Size of model or test valve	in. (mm)
K_v	Flow resistance coefficient of the valve	dimensionless
PSE	Pressure scale effects factor for cavitation analysis	dimensionless
P_u	Reference upstream pressure for cavitation analysis	psi (kPa)
P_{ut}	Upstream pressure from laboratory test for cavitation analysis	psi (kPa)
P_v	Vapor pressure adjusted for temperature and atmospheric pressure (Example: P_v = –14.4 psig (–99.6 kPa) for water at 60°F (16°C), measured at sea level.	psi (kPa)
P_{vt}	Vapor pressure from laboratory test	psi (kPa)
SSE	Sizing scale effects factor for cavitation analysis	dimensionless
Y	Size scale exponent for cavitation analysis	dimensionless
σ_c	Constant cavitation index at a reference pressure, P_u	dimensionless
σ_{ct}	Constant cavitation index from laboratory testing	dimensionless
σ_i	Incipient cavitation index at a reference pressure, P_u	dimensionless
σ_{it}	Incipient cavitation index from laboratory test	dimensionless
$\sigma_{operating}$	System installed and operating cavitation index	dimensionless

CAVITATION CALCULATION METHODOLOGY EXAMPLE

Given the system data and calculations from chapter 2, the valve data from chapter 3, and cavitation index values from Figure 4-5 and Table 4-1,

1. From the inputs, operating vapor pressure is –14.4 psig.
2. From chapter 2, calculate the valve upstream and differential pressures at the valve angle (θ).
3. Calculate the valve's operating cavitation index, σ_θ, at valve angle (θ) using Eq 4-1.
4. Calculate the pressure scale effect factor, PSE_θ, at valve angle (θ) using Eq 4-4.
5. Calculate the size scale exponent, Y_θ, at valve angle (θ) using Eq 4-6.
6. Calculate the size scale effect factor, SSE_θ, at valve angle (θ) using Eq 4-5.
7. Calculate the reference-corrected incipient cavitation index, σ_i, at valve angle (θ) using Eq 4-2.
8. Calculate the reference-corrected constant cavitation index, σ_c, at valve angle (θ) using Eq 4-3.
9. Repeat steps 3 through 8 for other valve angles (θ).
10. Compare the valve's operating cavitation index to the reference-corrected incipient cavitation index and the reference-corrected constant cavitation index at each valve angle (θ). When the operating cavitation index is at or below the

Table 4-1 Cavitation Calculation Data for Constant Head Source Example

Input	Input	Ch. 1 Calc.	Calc.	Calc.	Calc.	Calc.	Calc.	Calc.	Calc.	Calc.
Valve Angle θ (deg.)	$K_{V\theta}$	$\Delta H_{v\theta}$	ΔH_{SYS} Up-Stream of Valve (75%)	σ Operating	PSE	Y	SSE	6" test Incipient Cav. Index σ_i	6" test Constant Cav. σ_c	SSE & PSE Corrected Incipient Cav. Index σ_i	SSE & PSE Corrected Constant Cav. σ_c
90	0.30	0.9	74.3	165.79	0.888	0.41	1.754	27.18	14.72	41.77	22.36
80	0.40	1.2	74.1	124.92	0.888	0.38	1.687	19.09	11.32	28.11	16.46
70	1.10	3.4	72.5	46.91	0.891	0.29	1.501	13.21	8.57	17.32	11.12
60	3.10	8.9	68.3	18.15	0.898	0.23	1.368	9.13	6.40	10.98	7.63
50	8.30	20.8	59.4	8.24	0.912	0.18	1.278	6.47	4.75	7.37	5.36
40	24.80	44.0	42.0	4.31	0.938	0.13	1.205	4.83	3.54	5.33	3.87
30	83.30	72.5	20.6	2.92	0.968	0.10	1.148	3.83	2.70	4.14	2.89
20	333.30	91.3	6.5	2.48	0.986	0.07	1.102	3.08	2.18	3.26	2.28
10	3,000.00	99.0	0.8	2.35	0.993	0.04	1.058	2.18	1.90	2.24	1.94

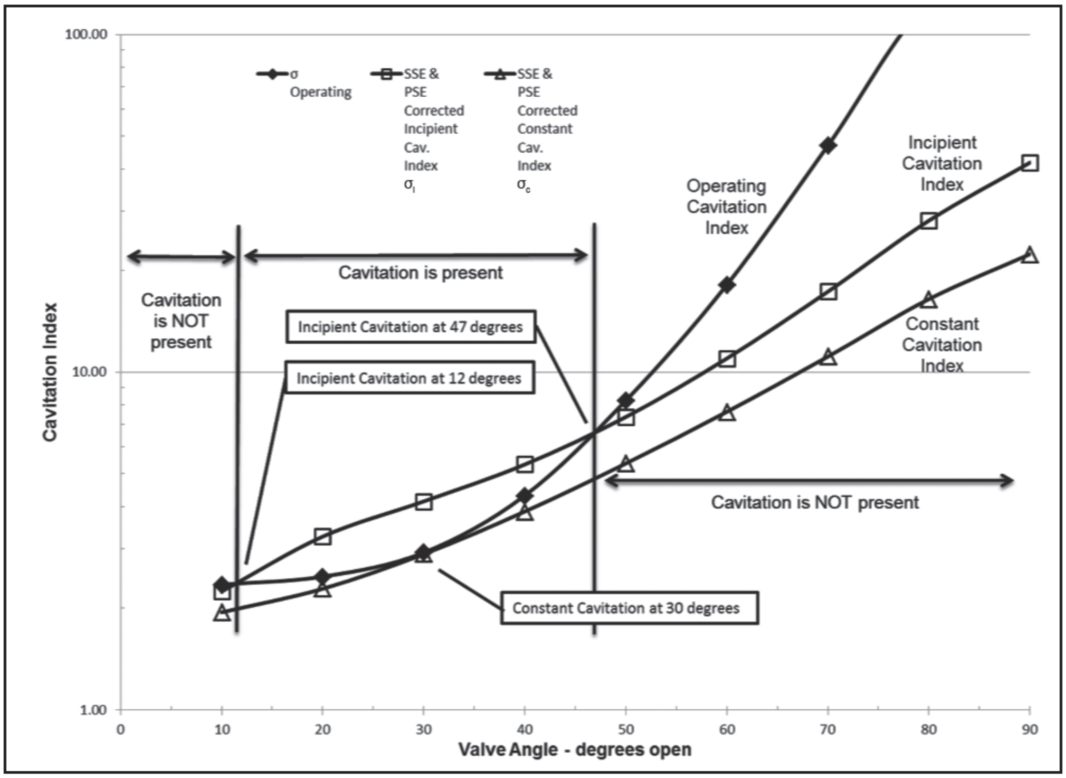


Figure 4-5 Example cavitation analysis summary graph

reference-corrected incipient cavitation index, cavitation is starting and increasing as the operating cavitation index drops further. When the operating cavitation index is between the reference-corrected incipient cavitation index and the reference-corrected constant cavitation index, cavitation exists and is audible within the system, but cavitation-related damage is unlikely. When the operating cavitation index is below the reference-corrected constant cavitation index, cavitation is constant and fully developed. As the operating cavitation index drops further, the risk of cavitation-related damage increases.

METHODS OF REDUCING CAVITATION

Design provisions for water systems completely without cavitation are beyond the scope of this manual, but some general recommendations to reduce cavitation can be considered. A detailed look at the cavitation index equation (Eq 4-1) may offer clues about how cavitation can be reduced.

To reduce cavitation, the value of the operating cavitation index, $\sigma_{operating}$, must be increased above the constant cavitation index for the valve, σ_c, similar to that shown in Figure 4-4. One way to do this is to increase the downstream pressure, P_d, which will increase the value of the cavitation index. Another strategy is to decrease the differential pressure across the valve, $(P_u - P_d)$. The value of the constant cavitation index, σ_c, can also be changed by using the valve at a different opening position or using a different valve type or model. This includes valves that incorporate cavitation-reducing trim that are available for many valves. Finally, air can be introduced to mitigate cavitation.

In practice, these changes can be achieved in some cases by using one or more of the following methods (Tullis 1989, pp. 145–165; Skousen 1998, pp. 511–517).

1. Increase the downstream pressure by relocating the quarter-turn valve in the system or providing additional restriction downstream using another valve or permanent restriction such as an orifice.
2. Decrease the differential pressure $(P_u - P_d)$ by using two or more valves in series to lower the differential pressure across each valve.
3. Throttle the valve at a different valve opening position by changing the size of the valve. To maintain the same differential and flow rate, a smaller valve may be used at a more open position or a larger valve may be used at a more closed position. Control valves are often reduced below the line sized for better control when large energy losses are necessary.
4. Install a smaller bypass line around the main valve to handle low-flow conditions.
5. Install air inlet ports immediately downstream of the valve shaft to admit air and reduce the zone of pressure differential in the pipe. The system must be capable of withstanding air, or provision should be made to remove the air (i.e., incorporating an air-release valve or valves). This technique is discussed in several sources (Tullis 1989, pp. 145–165).
6. Manufacturers may offer cavitation-reducing trim, which can be added to quarter-turn valves.

REFERENCES

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Chapter 5

Valve Testing

The importance of torque, head loss, and cavitation calculations has been demonstrated. These calculations are only as accurate as the coefficients used in the equations. The purpose of this chapter is to present a practical methodology for testing quarter-turn valves and developing flow, torque, and cavitation coefficients. Testing is recommended but is not a requirement of this manual or any other ANSI/AWWA standard. Coefficients can also be developed through analytical methods and based on geometric similarities. However, greater accuracy can be expected from testing. Any flow, torque, and cavitation coefficients developed using analytical methods, similarity comparisons, or mathematical or computational fluid dynamic models should be verified by testing to determine their accuracy.

UNCERTAINTY

All valves are individuals, just like people. Although we have many common traits that can be anticipated and categorized, we all perform differently in the present and over time. Uncertainty is a complicated subject and not a topic of this manual. Although there is an experimental accuracy associated with any data collection and reduction technique, most laboratories and researchers use measurements of good accuracy and replicated data to minimize the uncertainty in the results. The use of this methodology is intended to provide a best estimate of normal operating head loss, torque requirements, and cavitation potential under the conditions analyzed or specified.

It is expected that equipment measurement accuracy is recorded and a quality control program is used to ensure accurate test results, but specific uncertainty is not designated by this standard practice. The reader is directed to the International Society of Automation (ISA, formerly the Instrument Society of America) and the American Society of Mechanical Engineers (ASME), as both of these standards organizations have multiple documents covering this topic. The flow and pressure measurement uncertainty is provided in the following sections of this chapter.

TESTING REQUIREMENTS

The following requirements must be met in designing and conducting flow and torque tests:

1. The test media should be clean water in the range of 35°F to 80°F (10°C to 21°C). Different temperatures normally do not affect head loss measurements, but variation can affect torque when rubber materials are involved. It is recommended to perform additional testing to predict the extent of torque variation with respect to temperature.
2. The upstream and downstream piping should consist of a straight, horizontal run of pipe with the same nominal size as the test valve for a minimum length of 20 times the pipe diameter upstream and 10 times the pipe diameter downstream of the valve. Alternatively, the upstream length of straight piping may be as long as required to provide a fully developed uniform, tested, and documented flow profile at a minimum of three times the pipe diameter upstream of the tested valve. Flow conditioners may be used to improve flow conditions in the approach pipe. The piping friction head loss, determined before testing begins, must be subtracted from the measured head loss across the piping run for determining net flow, torque, and cavitation coefficients.
3. Flow, pressure, and torque measurements should be taken at a minimum of 10 positions in the valve's range of travel: typically, 0° (closed), 10°, 20°, 30°, 40°, 50°, 60°, 70°, 80°, and 90° (open). Many investigators find it best to collect data at 5° intervals. Measurements should also include additional data around the peak dynamic torque coefficient position or where a torque direction change (zero transition) occurs. Additional travel position tests are made at the discretion of the researcher, test technician, or manufacturer. Valve positions can be measured with a precision protractor, potentiometer, rotary variable differential transformer, or similar device connected directly to the valve shaft. The position reading on the valve actuator should not be used as a reliable indicator of precise position because of hysteresis in the connections and gearing.
4. Model valves may be used for testing, but they must be large enough that their Reynolds numbers exceed 100,000, and they must be geometrically similar to production valves. Model valves must not be smaller than 6 in. (150 mm) in nominal diameter. The manufacturer must verify the dimensional accuracy of the model to be tested to within 2 percent of actual scaled dimensions.
5. Flow testing determines flow resistance coefficient values, K or K_V; flow coefficient values, C_V; cavitation indices, σ; and dynamic torque coefficients, C_t. The flow testing must be conducted in accordance with ANSI/ISA S75.02.01-2008 except that piping manifold losses are subtracted to determine the net K_v for a valve. K values based on the direct measurements without subtracting the piping manifold losses will be referred to as gross K_v but will not be used in this methodology.
6. Pressures are measured in the pipe run two times the pipe diameter upstream and six times the pipe diameter downstream of the valve (Figure 5-1). Measurements are taken through static-wall, piezometric pressure taps located on opposite sides of the pipe at each location. The design of the taps must conform to ANSI/ISA S75.02.01-2008. There is one exception, as follows: For this standard practice, the valve shaft will be mounted vertically so that the effects of closure member weight, buoyancy, and center of gravity do not affect torque measurements and bias the torque data.

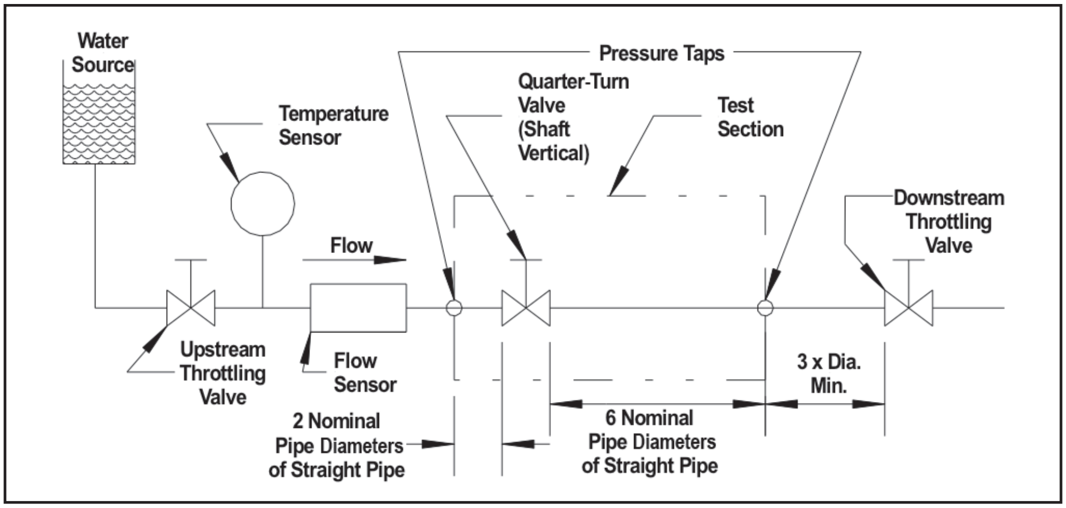

Figure 5-1 Basic flow test system

7. The volumetric flow is measured with National Institute of Standards and Technology traceable weight tanks, volumetric tanks, or flow nozzles within error limits not exceeding 2 percent.
8. The accuracy of the pressure measurements must remain within an error range of ±2% of the measured pressure differential.

TEST PROCEDURE

The following steps represent a generic procedure for flow testing quarter-turn valves. Because of testing constraints and unusual valve configurations, deviations or anomalies sometimes occur. Such conditions should be explained in the final test report.

Figures 5-1 and 5-2 show the test valve installation. The upstream pressure tap is visible to the right of the valve in Figure 5-2; however, the downstream tap (to the left of the test valve) is not visible. Shaft torque is measured by a strain gauge installed between the actuator and the valve. Flow rate is measured by calibrated flowmeters, weight tanks, or volumetric tanks (not shown).

1. The quarter-turn valve design or scale model should be checked to verify that it has the minimum amount of packing torque on the shaft to provide a reasonable seal. A small amount of packing leakage can be tolerated if it assists in lowering the packing torque, provided it does not affect the flow measurement accuracy or other instrumentation during the test. Rotate the valve in mid-range, measure, and record any running, packing, and/or hub seal torque, T_{pt}, with the valve full of water but zero flow.
2. Before installation in the test line, equip the valve with a device to provide a precise indication of valve angle. Mount a strain gauge or torque transducer to the valve shaft to record operating torque. Dynamic torque tending to close the valve should be recorded as positive, resulting in a torque being required by the actuator to open the valve. Dynamic torque tending to open the valve should be recorded as negative, resulting in a torque being required by the actuator to close the valve.
3. The valve should be mounted in the test line with the shaft vertical to avoid the effects of weight and buoyancy torque. This is contrary to the ISA requirement 4.6 regarding pressure tap and shaft location orientations. For nonsymmetrical

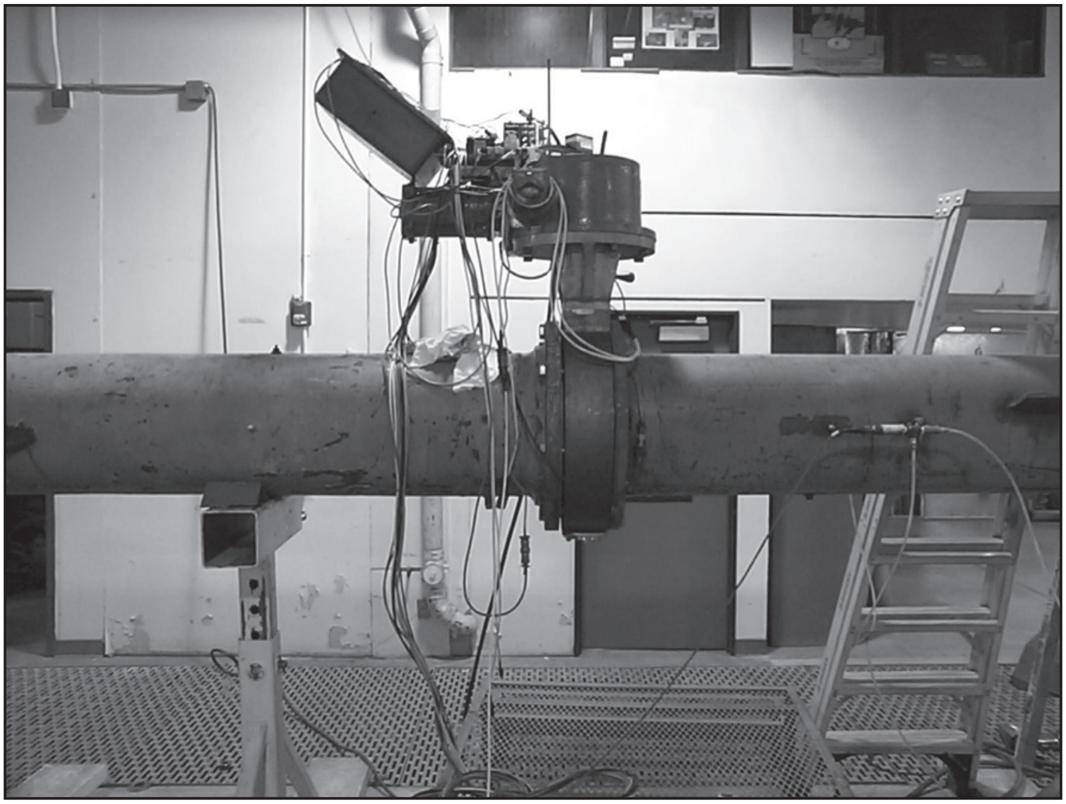

Figure 5-2 Butterfly valve test installation

Courtesy of Utah Water Research Laboratory

closure member designs, the flow direction orientation should be recorded. For double- or triple-offset valves, the rotation direction of the lateral offset or eccentricity torque (tending to open or close the valve) should be recorded.

4. The pipe run should be equipped with appropriate flow and pressure measurement devices, such as flow tubes, manometers, and pressure transducers or transmitters. The pipe should be pretested to determine the head loss over the eight diameters (two upstream plus six downstream) of the test pipe run .
5. The instrumentation equipment data, including any identification numbers, manufacturer, model numbers, scale, accuracy, and calibration last and due dates, should be recorded.
6. With the quarter-turn valve at the fully open position, subject the valve to flow in the range of 4 to 16 ft/s (1.3 to 5.2 m/s) and record flow and head loss. Repeat the test of fully open flow at a minimum of three different flow rates. Calculate the flow coefficient, K_v, for the valve at each test point. Subtract the pipe head loss from the pressure measurement to obtain ΔH_v. The flow equation is Eq 5-1. Compute the mathematical average of the calculated K_v values and round the result to two decimal places (e.g., 0.32). Repeat the flow coefficient test for lower angles at 10° or 10% (or smaller) increments.
7. Measure the torques required to rotate the valve shaft in the opening and closing directions at each increment of valve position. Torque readings must be taken with the valve rotating so that bearing torques are measured. A rise in opening torque indicates that dynamic torque is tending to close the valve. Measured torques combine dynamic torque, T_d; bearing torque, T_b; and packing and hub

torque, T_P. Calculate the dynamic torque based on Eq 5-2 (note: use torque in in.-lb or N-m only). Repeat the torque coefficient test for the same lower angles at 10° or 10 percent (or smaller) increments as performed for the flow coefficient testing.

8. The bearing torque (and the apparent bearing coefficient) at a small closure member angle at which the closure member does not interfere with the seat can be determined by calculating the difference of the two torque values (opening and closing) and subtracting the packing and hub torque measured in step 1 (note: use torque in in.-lb or N-m only).
9. The quarter-turn valve should be tested at a maximum of 10° intervals to determine incipient, constant, and choking cavitation indices by adjusting upstream and downstream control valves. The upstream pressure is held constant (typically at 70 psig [481 kPa]) while the flow is increased in small increments until the desired cavitation limits are identified. A graph of the logarithmic accelerometer output versus flow rate or cavitation index is helpful. The slope of this curve normally changes at the points of incipient, constant, and choking cavitation as shown in Figure 4-2. If it is not possible to hold the upstream pressure constant, the audible method may be used to identify incipient and constant cavitation, and then the laboratory result can be scaled to a predetermined upstream pressure. Additional information on cavitation testing can be found in ISA-RP75.23-1995.
   - Incipient cavitation is indicated by an intermittent popping noise or increase in vibration above the flow turbulence.
   - Constant cavitation is indicated by a steady noise and vibration increasing at a slower rate. The intensity of the cavitation is the same as incipient cavitation, but the occurrence of the cavitation is steady.
   - Choking cavitation occurs when the flow rate no longer increases with further opening of the downstream control valve while not increasing the upstream pressure. The choking limit may exceed the flow capability of the test loop. When this occurs, this data may be ignored or alternate means (such as use of a free discharge test pipe and including the liquid pressure recovery factor) may be used to obtain reasonable results.
10. The flow tests should be repeated with flow oriented in the opposite direction for a valve with an offset or asymmetric closure member.
11. Incipient and constant cavitation coefficients (σ_i and σ_c) should be corrected for size scale effects (SSE) and pressure scale effects (PSE) according to Eqs 5-8 and 5-6 and reported in a summary table at the same upstream pressure and size (i.e., 12 nominal valve size and 70 psig [481 kPa]) (Tullis 1989). These data should include (1) the SSE-corrected size (12 in.) used, (2) the nominal size of the valve tested, (3) the PSE-corrected pressure (70 psig), and (4) the vapor pressure of the water during the test. When all data are adjusted to the same size (12 in.) and upstream pressure (70 psig), then the results of different valve styles and designs can be directly compared. If the data are not SSE- or PSE-adjusted, the actual size, upstream pressure, and water vapor pressure should be reported.
12. Alternative equations may be used to present and predict cavitation data provided they are clearly shown and understood.
13. Summarize the test data for each data point and report at least these results:
   - Valve model and materials
   - Construction drawing and revision level or date
   - Valve angle in degrees open
   - Flow velocity (based on nominal valve size)

- Total head loss (measured)
- Head loss (piping)
- Net head loss (valve only)
- Average $K_{v\theta}$, T_p, T_b, $T_{d\theta}$, $C_{t\theta}$, C_f, values
- Opening and closing torque (consisting of $T_d + T_b + T_p$)
- $\sigma_{i\theta}$, $\sigma_{c\theta}$, or other cavitation coefficients at the reference pressure (see Figure 4-4)

$$K_V = \frac{2 \times g \times \left(\Delta H_t - \Delta H_{pipe}\right)}{V^2} \tag{5-1}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
g	Gravitational constant Acceleration due to gravity, 32.2 ft/s² (9.81 m/s²)	ft/s² (m/s²)
K_v	Flow resistance coefficient of the valve	dimensionless
V	Velocity of fluid flow	feet per second, ft/s (meters per second, m/s)
ΔH_{pipe}	Measured head loss across the pipe during testing without the valve	feet of water (meters of water)
ΔH_t	Measured head loss across the valve and pipe during testing	feet of water (meters of water)

$$T_{dt} = \frac{\text{Opening torque} + \text{Closing torque}}{2} \tag{5-2}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
Closing torque	Test measured torque in the closing direction	in.-lb (N-m)
Opening torque	Test measured torque in the opening direction	in.-lb (N-m)
T_{dt}	Measured dynamic torque from testing (a positive value indicates a tendency to close the valve)	in.-lb (N-m)

Calculate the dynamic torque coefficient based on the formula

$$C_t = \frac{1}{U_{C2}} \times \frac{T_{dt}}{D_d^3 \times \Delta P_t} \tag{5-3}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
C_t	Coefficient of dynamic torque (positive value tends to close valve), general form	dimensionless
D_d	Disc diameter	in. (mm)
T_{dt}	Measured dynamic torque from testing (a positive value indicates a tendency to close the valve.)	in.-lb (N-m)
U_{C2}	Units conversion factor: US customary for torque in in.-lb: U_{C2} = 1 in./in. US customary for torque in ft-lb: U_{C2} = 1⁄12 (0.0833) in./ft Metric for torque in N-m: $U_{C2} = 1 \times 10^{-6}$ (0.000001) m²/mm²	in./in. (in./ft) (m²/mm²)
ΔP_t	Measured pressure drop across the disc from testing	psid (kPa)

Calculate the bearing torque based on the formula

$$T_{bt} = \frac{\text{Opening torque} - \text{Closing torque}}{2} - T_{pt} \tag{5-4}$$

Compute the bearing coefficient of friction

$$C_f = \frac{1}{U_{C2}} \times \frac{8 \times T_{bt}}{\pi \times D_d^2 \times d_s \times \Delta P_t} \tag{5-5}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
C_f	Coefficient of friction between the shaft or trunnion and bushing, dimensionless. (This value may be obtained from a flow test, engineering handbooks, the bearing manufacturer, or the valve manufacturer.)	dimensionless
Closing torque	Test measured torque in the closing direction	in.-lb (N-m)
D_d	Disc diameter	in. (mm)
d_s	Shaft diameter	in. (mm)
Opening torque	Test measured torque in the opening direction	in.-lb (N-m)

Variable	Definition or Description	Units US Customary (SI metric)
T_{bt}	Measured bearing torque from testing	in.-lb (N-m)
T_{pt}	Measured packing and hub torque from testing	in.-lb (N-m)
U_{C2}	Units conversion factor: US customary for torque in in.-lb: $U_{C2} = 1$ in./in. US customary for torque in ft-lb: $U_{C2} = 1/12$ (0.0833) in./ft Metric for torque in N-m: $U_{C2} = 1 \times 10^{-6}$ (0.000001) m²/mm²	in./in. (in./ft) (m²/mm²)
ΔP_t	Measured pressure drop across the disc from testing	psid (kPa)

$$PSE = \left(\frac{P_u - P_v}{P_{ut} - P_{vt}}\right)^{0.28} \tag{5-6}$$

$$Y = 0.3 \times K_v^{-0.25} \tag{5-7}$$

$$SSE = \left(\frac{D}{d_t}\right)^{Y} \tag{5-8}$$

$$\sigma_i = (\sigma_{it} - 1) \times PSE \times SSE + 1 \tag{5-9}$$

$$\sigma_c = (\sigma_{ct} - 1) \times PSE \times SSE + 1 \tag{5-10}$$

Note: See Figure 4-3 for typical graphs of cavitation indices.

where

Variable	Definition or Description	Units US Customary (SI metric)
PSE	Pressure scale effects factor for cavitation analysis	dimensionless
SSE	Sizing scale effects factor for cavitation analysis	dimensionless
Y	Size scale exponent for cavitation analysis	dimensionless
P_u	Reference upstream pressure for cavitation analysis	psi (kPa)
P_{ut}	Upstream pressure from laboratory test for cavitation analysis	psi (kPa)
P_v	Vapor pressure adjusted for temperature and atmospheric pressure. Example: $P_v = -14.4$ psig (–99.6 kPa) for water at 60°F (16°C), measured at sea level.	psi (kPa)

Variable	Definition or Description	Units US Customary (SI metric)
P_{vt}	Vapor pressure from laboratory test	psi (kPa)
σ_c	Constant cavitation index at a reference pressure, P_u	dimensionless
σ_{ct}	Constant cavitation index from laboratory testing	dimensionless
σ_i	Incipient cavitation index at a reference pressure, P_u	dimensionless
σ_{it}	Incipient cavitation index from laboratory test	dimensionless

Note: Pressures may be gauge or absolute but must be consistent.

SEATING/UNSEATING TEST PROCEDURE

The following steps represent a generic procedure for performing a seating/unseating torque test for a quarter-turn valve. Because of testing constraints and unusual valve configurations, deviations and anomalies sometimes occur. These conditions should be explained in the final test report. Tests should be performed for both flow directions for most valves unless the valve design is completely symmetrical. Valves with offset and asymmetrical closure members should be tested in both directions.

1. The valve model, type, and materials of construction should be recorded. The bearing material and friction coefficient, C_f, or the measured bearing friction torque from step 8 of the preceding flow test will be needed for report calculations. The quarter-turn valve should first undergo a shell and seat leak test to verify proper adjustments of the seat and packing (if applicable).
2. Rotate the valve into a mid-travel position and measure any packing and hub seal torque ($T_p + T_h$) with the valve full of water but no flow or pressure. Repeat this measurement three times and average the results.
3. With a blind test head on one flange and water in the valve, slowly close the valve with no differential pressure applied and record the highest total closing (seating) torque ($T_s + T_b + T_p \pm T_{ecc}$). Slowly open the valve and record the highest total opening (unseating) torque ($T_{us} + T_b + T_p \pm T_{ecc}$). Repeat this test three times and average the results for a no-pressure condition. For some valves, this should be done at a low differential pressure to ensure the seat tightness is achieved.
4. With a blind test head on one flange, pressurize the valve to its rating. Slowly open the valve and record the highest total opening (unseating) torque ($T_{us} + T_b + T_p + T_h \pm T_{ecc}$). Slowly close the valve with the differential pressure applied and record the highest total closing (seating) torque ($T_s + T_b + T_p + T_h \pm T_{ecc}$). Repeat this test three times and average the results.
5. Repeat step 4 above at one or more intermediate pressures to determine the pressure-dependent seating coefficients.
6. Compute the unseating torque (T_{us}) and seating torque (T_s) by subtracting T_b (from calculation or from the flow test) and T_p (including hub seal torque if applicable, measured in step 2) from total measured torque. If the valve design is double or triple offset, the eccentricity torque (T_{ecc}) should be added or subtracted depending on rotation direction.

7. Graphically plot the seating and unseating torques on the ordinate (y-axis) and the differential pressure on the abscissa (x-axis). Perform a linear regression of the data to determine the seating and unseating constant and pressure-dependent coefficients.

Although often based on the higher of the opening or closing torque values, separate seating and unseating coefficients may be determined individually when large differences exist. (Note: Use torque in in.-lb or N-m only.)

Compute the seating coefficient, C_{sc}, and unseating coefficient, C_{usc}, based on the graphical data and linear regression analysis of step 7 above and the following equations. If the seating and/or unseating torque are reasonably constant at all pressures, then it is only necessary to determine the pressure-independent coefficients. If these data show a definite positive or negative slope relative to the operating pressure, then the pressure-dependent coefficients should be developed.

The following example and equations are based on the ball and butterfly valve seating and unseating methods (Eq 3-5) shown in chapter 3. For double- and triple-offset butterfly valves, the alternate coefficients may be developed consistent with the nomenclature provided in Eq 3-6. For plug valves, the coefficients need to be developed consistent with the nomenclature provided in Eq 3-7.

$$C_{sc} = \frac{1}{U_{C2}} \frac{T_{st}}{D_d^2} \tag{5-11}$$

$$C_{usc} = \frac{1}{U_{C2}} \frac{T_{ust}}{D_d^2} \tag{5-12}$$

where

Variable	Definition or Description	Units US Customary (SI metric)
C_{sc}	Constant or pressure-independent coefficient of seating torque	lb/in. (N/m)
C_{usc}	Constant or pressure-independent coefficient of unseating torque	lb/in. (N/m)
D_d	Disc diameter	in. (mm)
T_{st}	Measured seating torque (always positive)	in.-lb (N-m)
T_{ust}	Measured unseating torque (always positive)	in.-lb (N-m)
U_{C2}	Units conversion factor: US customary for torque in in.-lb: U_{C2} = 1 in./in. US customary for torque in ft-lb: U_{C2} = 1⁄12 (0.0833) in./ft Metric for torque in N-m: $U_{C2} = 1 \times 10^{-6}$ (0.000001) m²/mm²	in./in. (in./ft) (m²/mm²)

$$T_s = \left(C_{sc} + C_{sp} \times \Delta P_V\right) \times D_d^2 \tag{5-13}$$

and

$$T_{us} = \left(C_{usc} + C_{usp} \times \Delta P_V\right) \times D_d^2 \qquad (5\text{-}14)$$

where

Variable	Definition or Description	Units US Customary (SI metric)
C_{sc}	Constant or pressure-independent coefficient of seating torque	lb/in. (N/m)
C_{sp}	Pressure-dependent coefficient of seating torque	lb/in./psi (N/m/kPa)
C_{usc}	Constant or pressure-independent coefficient of unseating torque	lb/in. (N/m)
C_{usp}	Pressure-dependent coefficient of unseating torque	lb/in./psi (N/m/kPa)
D_d	Disc diameter	in. (mm)
T_s	Seating torque (always positive)	in.-lb (or ft-lb) (N-m)
T_{us}	Unseating torque (always positive)	in.-lb (or ft-lb) (N-m)
ΔP_V	Pressure drop (or loss) across the valve, general form	psid (kPa)

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Chapter 6

Valve Applications

This chapter provides recommendations for actuator sizing and valve installations. Some piping configurations encountered in water systems can dramatically affect head loss through a valve and its operation. These effects should be understood by the system designer. The chapter also includes cautions that should be observed when quarter-turn valves are used for throttling service, when they are subject to unusual upstream flow conditions, and when actuators are removed.

ACTUATOR SIZING

The formulas for determining total break, opening, and closing torques are presented in chapter 3. Actuator sizing should be based on a comparison of the highest torque values for the valve with the torque rating and output capability of the actuator. The chapter 3 "Combining Torque Components" section total operating torque is the minimum required shaft torque (MRST) at each travel position. This torque is the most probable torque requirement for the valve when operating under the system conditions analyzed.

The torque formulas do not include application factors that should be taken into account when sizing actuators. Refer to the applicable AWWA valve standard for application factors (AFs) and other considerations necessary to properly size manual, cylinder, and electric actuators. The MRST values are multiplied by the appropriate AFs given in these standards to determine the actuator sizing torque (AST) (AST = MRST × AF). In the AWWA standards, the authors have used the term *application factors*, as there are different values associated with the actuator type and application function. Other industry standards use the terms *safety factor* or *margin* in a similar manner.

Manual Actuator Sizing

Manual actuators are sized based on two criteria. First, the actuator rating must exceed the MRST at all valve positions. Second, the actuator must be sized to allow operation of the valve without exceeding certain limits on handwheel or chainwheel pull force or input torque (such as 80-lb [356-N] handwheel rim pull force or 150 ft-lb [219 N-m] input torque of the current ANSI/AWWA C504-15). The calculation of rim pull force or input torque requires a review of the characteristic torque curve for the actuator (see Figure 6-1).

The typical AWWA maximum handwheel or chainwheel pull requirement of 80 lb has been found by some operating personnel to be a high exertion of effort, and lesser pulls of 40 to 60 lb (18.1 to 24.2 kg) have sometimes been found to be beneficial. This may require more input turns, a larger handwheel or chainwheel, or perhaps larger or more efficient actuators.

Typical torque multiplier curves are presented for both a worm-gear actuator and a traveling nut–type actuator in Figure 6-1. The torque multiplier is the expected ratio of output torque to input torque, considering the efficiency of the mechanism and/or gearing. For example, if an actuator has a torque multiplier of 20 and a butterfly valve requires an operating torque of 8,000 in.-lb (666 ft-lb, 973 N-m), then the actuator will require an input torque of 8,000 in.-lb/20 (666 ft-lb/20, 973 N-m/20) or 400 in.-lb (33.3 ft-lb, 49 N-m).

The worm-gear actuator has a constant torque multiplier throughout the full travel, and the “curve” in Figure 6-1 is a horizontal straight line at all valve opening positions. Conversely, the traveling nut–type actuator, which includes scotch yoke or link-and-lever designs, have variable torque multipliers depending on valve opening position. When calculating the input torque requirements of an actuator with a variable torque characteristic curve, use the operating torque at each valve opening position and the corresponding actuator torque multiplier.

The input torque is calculated by dividing the required valve torque by the actuator’s torque multiplier. Handwheel rim pull force is calculated by further dividing the input torque by the radius of the handwheel.

Example cutaway images of a worm gear actuator (Figure 6-2), a scotch yoke traveling nut actuator (Figure 6-3), and a link-and-lever traveling nut actuator (Figure 6-4) are provided, and they display the internal mechanisms. The scotch yoke and the link-and-lever traveling nut actuators are often used on quarter-turn valves as the torque multiplier or mechanical advantage is greatest at the closed position. In smaller valves, this is advantageous as the seating and unseating torque is often the greatest torque requirement. In larger valves in which the greatest operating torque may be at mid-stroke positions, this is still an advantageous characteristic as this slows the rotation speed as the valve approaches the closed position and assists in reducing the risk of a pressure transient caused by too fast a valve closure.

Cylinder Actuator Sizing

There are five basic types of pneumatic or hydraulic actuators used on AWWA quarter-turn valves. These include rack-and-pinion, scotch yoke, link-and-lever, pivoting cylinder, and vane. The first four types require a separate mechanism (gear or lever) to convert the cylinder’s linear thrust output into torque for operating the quarter-turn valve. The vane actuator is the only one that converts fluid pressure directly into torque.

The mechanical gear or lever portion of a cylinder actuator should be designed to handle the torques calculated as in the preceding for manual actuators. Selection of the cylinder bore size (or area) should be based on the minimum supply pressure to the cylinder. Additionally, ANSI/AWWA C504-15, ANSI/AWWA C507-15, ANSI/AWWA C516-14, and future ANSI/AWWA C517 recommend AFs on the basis of the type of cylinder controls when sizing the actuator. (Note: The current edition of ANSI/AWWA C517-16 does not currently use the term AF. The newer editions are anticipated to conform to this nomenclature.)

The AF is needed to allow for pressure drop in cylinder control valves and speed control devices. Higher application factors are used for cylinders powered by air and used for throttling flow because the cylinder is moved by a floating differential pressure across the piston, which is created by a small orifice in the positioner. Because of the compressibility of the air, cylinder throttling without a positioner should be avoided.

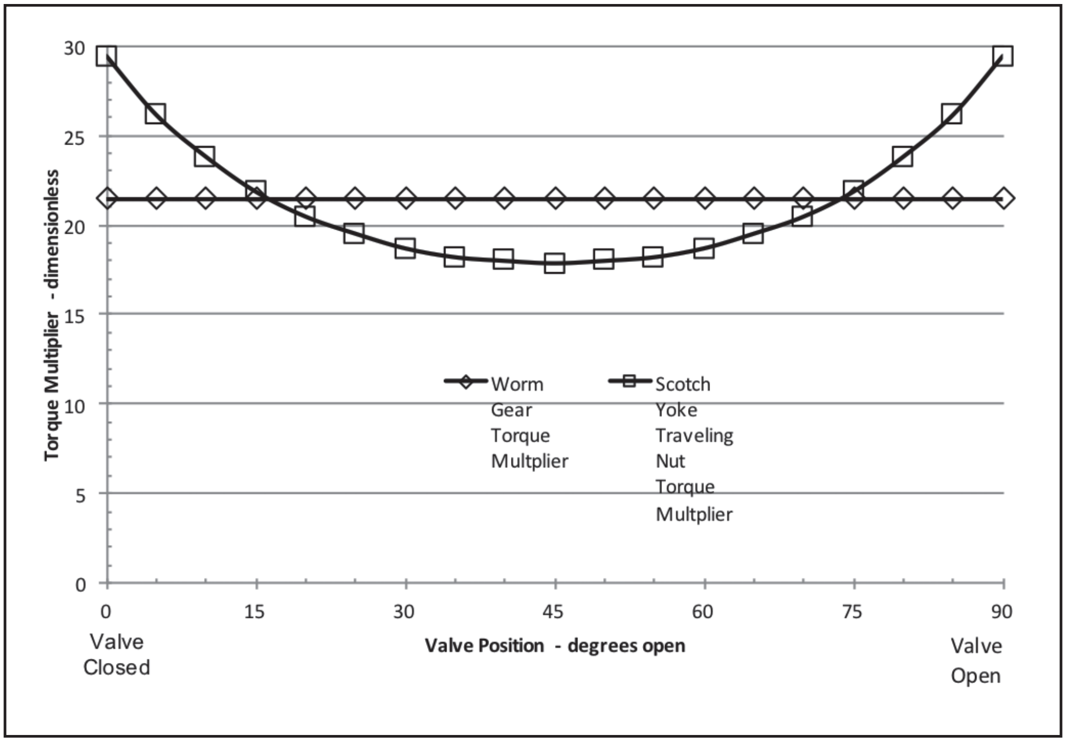
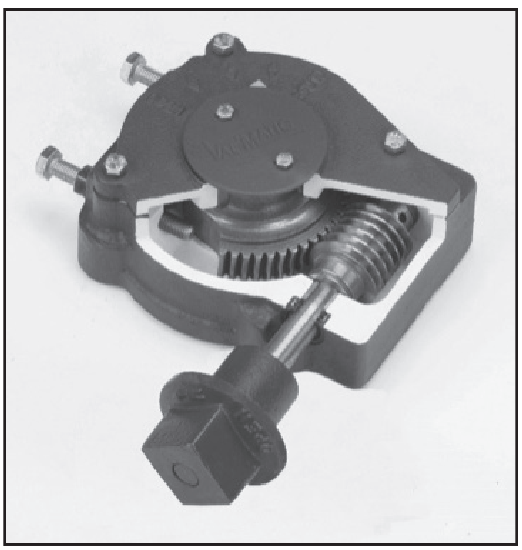


Figure 6-1 Typical actuator torque characteristics



Courtesy of Val-Matic

Figure 6-2 Worm gear actuator

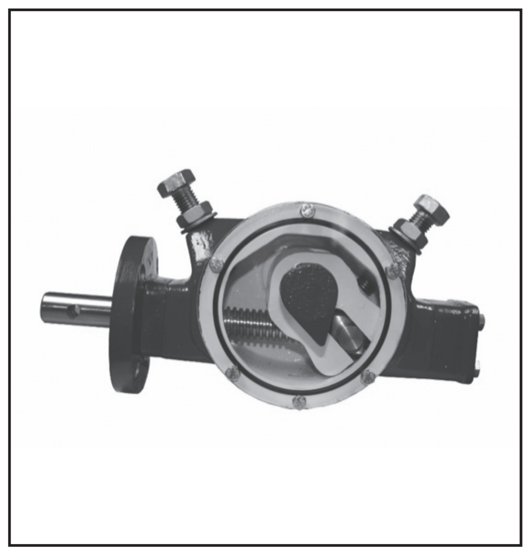
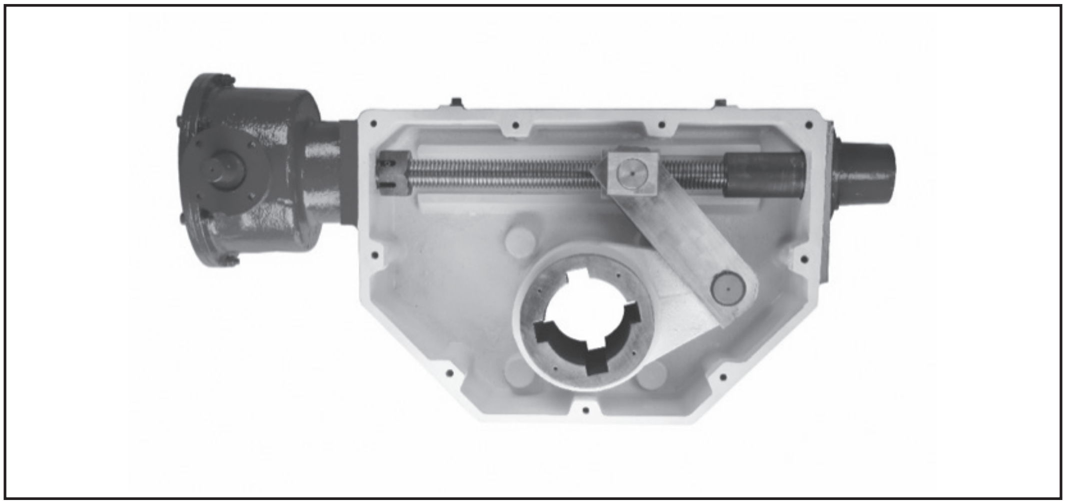


Figure 6-3 Scotch yoke traveling nut actuator cutaway

Some cylinder actuators are equipped with springs for fail-safe closure or fail-safe opening. A spring-return cylinder requires more analysis of sizing on the basis of calculated valve required operating torques, cylinder AFs, and the variable torque generated by the spring. In all cases, actuator manufacturer's data and sizing information should be consulted for proper actuator sizing of these actuators.



Courtesy of Val-Matic

Figure 6-4 Link-and-lever traveling nut actuator cutaway

Many cylinder actuators have a nonlinear variable output characteristic relative to valve position. When this occurs, the actuators AFs (or margins) should be evaluated at each calculated valve position as illustrated in Figure 6-5.

Refer to standard ANSI/AWWA C541-16, Hydraulic and Pneumatic Cylinder and Vane-Type Actuators for Valves and Slide Gates, for the design and construction of these actuators and the AWWA Manual M66 Water Supply Practices, *Cylinder Actuators and Controls – Design and Installation*. Appendix A of ANSI/AWWA C541-16 has a useful actuator data form for selecting and specifying actuator requirements.

Electric Actuator Sizing and Switch Settings

The gear portion of the electric actuator is sized on the basis of the break and running torques provided by this manual of practice. Additionally, the motor should be sized on the basis of the minimum expected motor voltage plus the ANSI/AWWA C504-15, ANSI/AWWA C507-15, ANSI/AWWA C516-14, or ANSI/AWWA C517-16 recommended AFs.

An electric actuator for a quarter-turn valve may be wired for limit switch position seating or torque seating as indicated by the manufacturer. Symmetric and single-offset rubber-seated valves are usually position seated, and double- and triple-offset valves may be torque seated. For position-seated valves, the torque switch is used for over-torque protection. It can then be set above the AST and below the actuator's rating or the torque that stalls the motor. This protects the unit from burnout should an unusually high torque be encountered (such as an obstruction in the pipeline).

Refer to standard ANSI/AWWA C542-16, Electric Motor Actuators for Valves and Slide Gates, for further design and construction requirements of these actuators.

EXTENDED BONNET INSTALLATION

When the quarter-turn valve must be operated from a significant distance above because of a buried or submerged condition, the valve may be equipped with an extended bonnet. An extended bonnet consists of an outer pipe that is rigidly attached to the valve body and extends up and rigidly supports the actuator. The valve shaft is extended up through the extended bonnet with a shaft or inner pipe and connects to the actuator mechanism. The inner pipe then rotates 90° to operate the quarter-turn valve. The typical construction is illustrated in Figure 6-6.

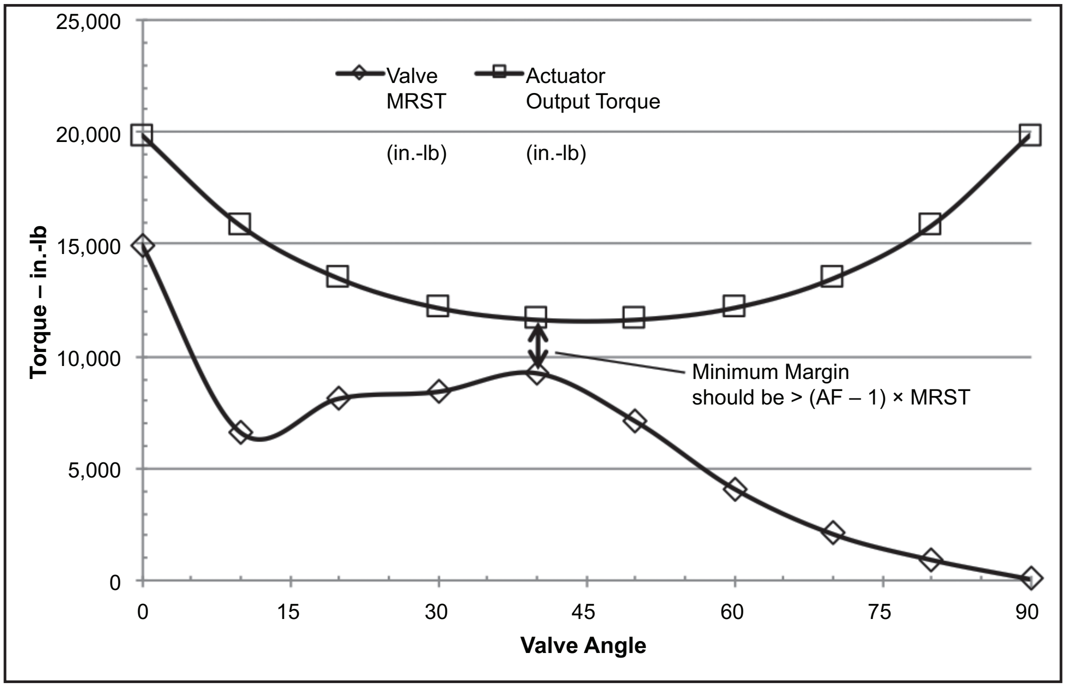


Figure 6-5 Actuator sizing characteristics graph

To assure tight shutoff of the seated quarter-turn valve, the actuator and extended bonnet must position the valve closure member to within a limited angle depending on the valve design. For example, ball and butterfly valves typically need to position the closure member to within about ±1° of the true closed position. Because plug valves have eccentric action and may be torque seated, the angle may be higher, typically ±2° or ±3°. Because both the outer and inner pipes see the full operating torque of the valve, they are subject to torsional deflection. The total torsional deflection is the sum of the deflections of the outer and inner pipes. As the length of the bonnet increases, the deflection increases proportionally. Hence, it is often more important to size the extended bonnet pipes to limit the torsional deflection rather than strength and allowable stress. During the mid-stroke positions, the torsional deflection is not as important and stress becomes more critical. However, any torsional deflection at mid-stroke positions should be limited to about 3° to 5°.

With longer extensions (greater than approximately 6 ft [1.8 m]), it may become impractical to limit the torsional deflection to low values because the required bonnet pipe sizes become exceptionally large. As an alternative, some extended bonnet designs may have a closed stop-limiting bolt near the valve. When a closed-position stop is used, the extended bonnet is allowed to deflect more, and the closed stop will precisely position the valve disc in the closed position. In this case, the extended bonnet can be allowed to deflect up to 3° or 5°, depending on the allowable over-travel in the actuator. Whether position seated or torque seated, it is good practice to base the design of extended bonnets on deflection requirements as well as material strength requirements.

EFFECTS OF PIPE INSTALLATIONS

Proper installation can prevent serious problems with valve performance and life expectancy. Many operating conditions should be reviewed with the manufacturer, such as flow rate, differential pressure, temperature, and so on. A comprehensive list of information

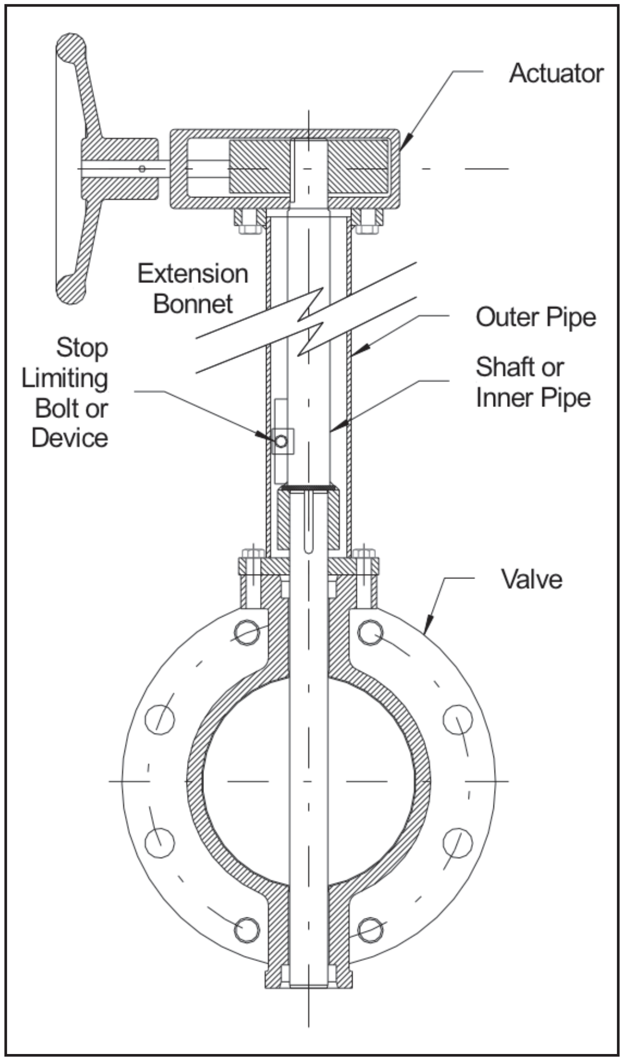


Figure 6-6 Typical extended bonnet construction

that should be included when placing orders is given in the forewords of ANSI/AWWA C504-15, ANSI/AWWA C507-15, ANSI/AWWA C516-14, and ANSI/AWWA C517-16. In addition, the following recommendations should be followed in the placement and installation of a quarter-turn valve in a piping system. Appendix A is a specification and data sheet that can be used to communicate quarter-turn valve design requirements in accordance with the AWWA quarter-turn valve standards.

The design of quarter-turn valves is based on straight pipe testing in laboratory conditions. Any upstream piping components that cause asymmetric flow conditions or increased turbulence at the valve will alter the valve's performance as well as the system's performance. In many cases, these changes are benign, but there are several installation conditions that can severely alter quarter-turn valve performance. The following examples provide some basic guidelines for the use and installation of quarter-turn valves. Although many figures depict a butterfly valve, the same effects apply to all quarter-turn valves.

Flow and Pressure Direction

Butterfly valves with symmetrical discs can typically be installed with flow and pressure in either direction. Butterfly and plug valves with asymmetric closure members have different flow as well as seating, unseating, and dynamic torque properties, depending on whether the shaft is upstream or downstream of the seat. Also, many asymmetric valves tend to seal better with the shaft on the upstream side of the valve seat (review Figures 3-1 and 3-3). Although a valve may have a preferred orientation on the basis of the pressure assisting the seal performance, the seating, unseating, and dynamic torque requirements may require the opposite orientation. Check with the manufacturer before installing a valve for the preferred direction of flow or pressure if not indicated on the valve body or in the manuals or drawings.

If system configuration or operating modes require a valve to be installed opposite to the valve's preferred installation orientation, consult the manufacturer for recommendations or modifications.

The valve installation can also be affected by the actuator configuration. Conditions may favor orienting the handwheel or operating nut in a specific direction. If the desired actuator orientation does not match the required flow direction and system orientation, then the valve manufacturer should be consulted about reorienting an actuator.

General Upstream Flow Disturbance

The first consideration concerns the flow rates. If the flow rate and corresponding fluid velocity are low, then the effect of an upstream flow disturbance is likely inconsequential to the valve's dynamic performance (head loss and torque). As the local velocities that approach the valve become greater, their effect on the dynamic torque and head loss increases, and atypical loads and stresses are developed in the valve. The severity of the discontinuity in the local velocity patterns at the valve increases the uncertainty about the valve's performance. Some research has been done on the effects of elbows and pumps upstream of quarter-turn valves, but the data are limited as there are many variables to consider. These variables include valve type, elbow or pump orientation relative to the valve shaft axis, distance between the disturbance and the valve, closure member rotation direction, and fluid velocity (or flow rate).

These conditions affect the fluid dynamic responses of the valve (head loss and dynamic torque). For valves that have low friction-related torque components, these dynamic effects can be of greater consequences than for valves that have high friction-based torque requirements. As an example, a valve using nonstick packing, a clearance seat, and roller bearings will have little operating friction torque requirements, and any change in the dynamic torque will be of major significance to the total operating torque. A valve constructed of asbestos-free braided pull-down packing, low-leakage metal seats, and stainless-steel bearings and shafts will have significant operating friction torque, and the changes in dynamic torque will not greatly affect the total operating torque requirement.

For most water works applications, some general rules of thumb can be used provided that the maximum velocities are less than the AWWA class B velocity designation of about 16 ft/s. For best dynamic performance (head loss and torque), a length of straight pipe between the disturbance and the valve greater than eight times the pipe diameter is sufficient to provide normal flow through the valve. For good performance, there should be two or three pipe diameters of straight pipe upstream of the valve. Less than two pipe diameters of upstream straight pipe may impair valve performance at higher fluid velocities. Dynamic torque can be doubled or tripled by an improperly oriented valve installation. The valve shaft should be oriented such that the fluid flow on either side of the shaft centerline is essentially equal.

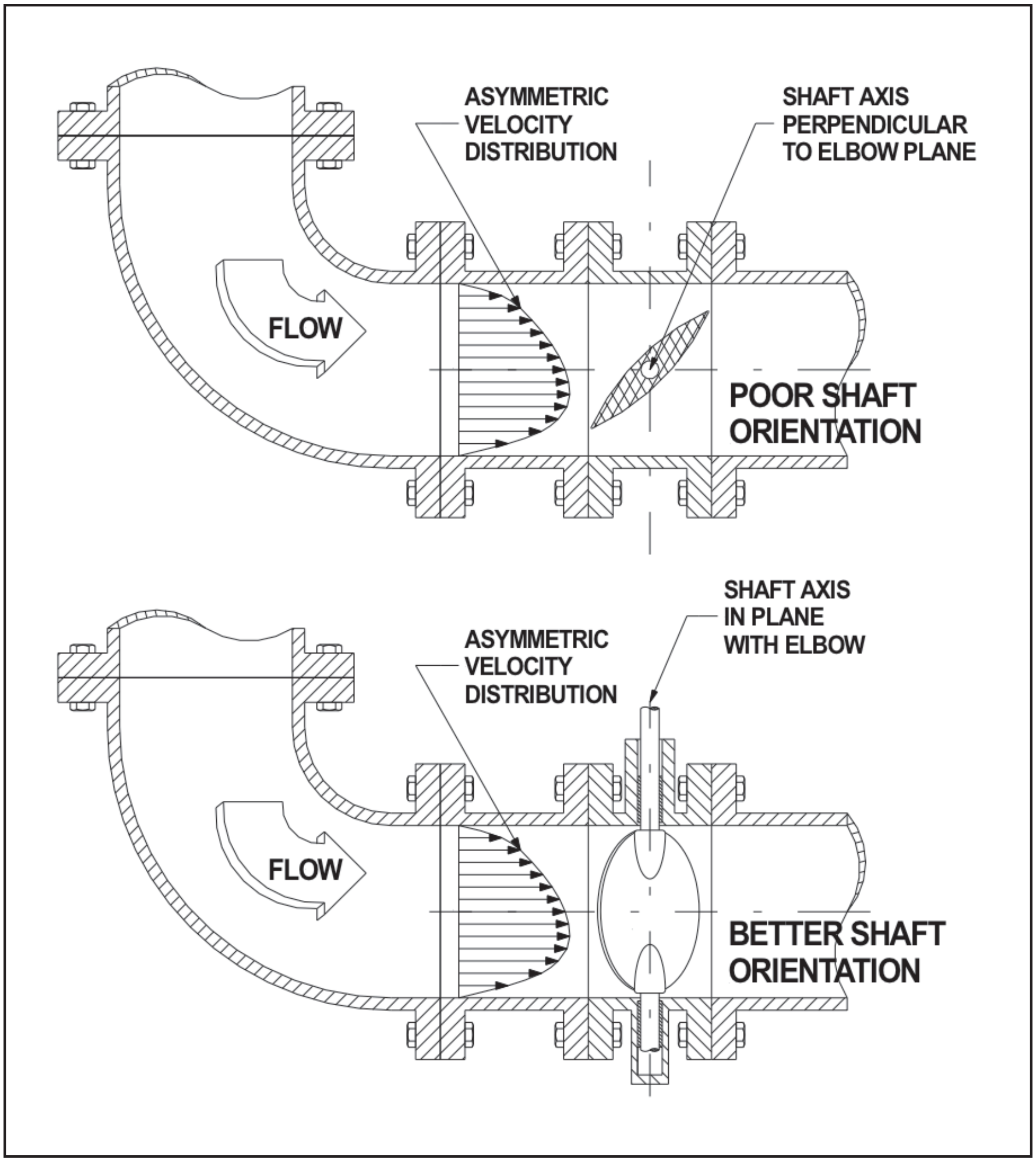


Figure 6-7 Vertical elbow upstream of a butterfly or other quarter-turn valve preferred installation orientation

When operating conditions are extreme or operation and function are critical, it may be best to look for similar operating experience or perform model testing of the installation.

Upstream Elbow or Branch Tee

Elbows and branch tees cause asymmetrical fluid velocity in the pipe (Figure 6-7), which affects quarter-turn valve operation. This figure and others in this section depict butterfly valves as these are generally more vivid visual representations. However, these principles apply to all quarter-turn valve installations. Dynamic torque can be doubled by an improperly oriented valve and an upstream elbow. The valve shaft must be positioned vertically when installed downstream of a vertical elbow or tee. For a horizontal elbow or tee, the valve shaft should be positioned horizontally.

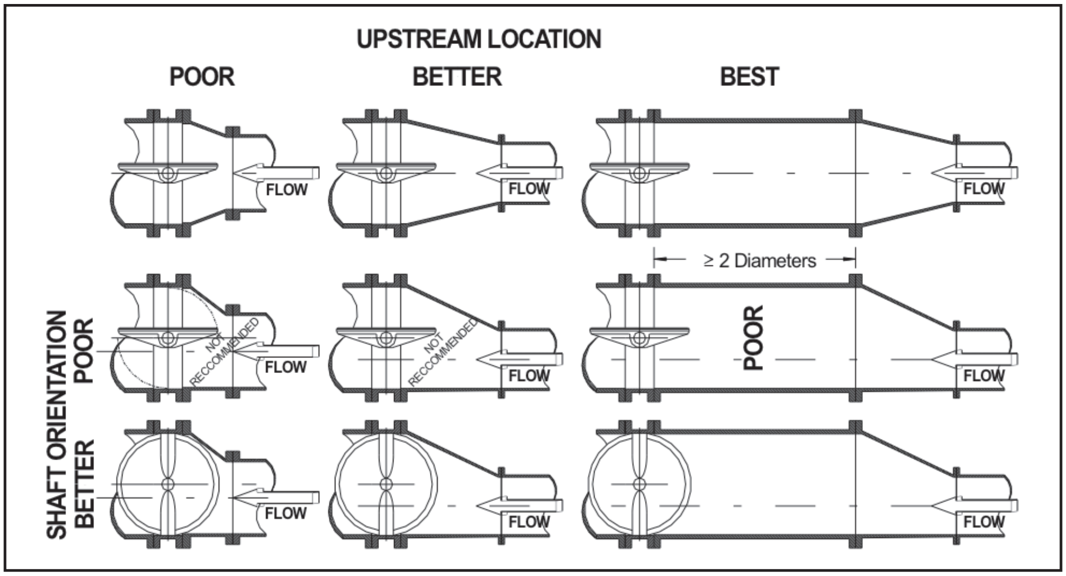


Figure 6-8 Upstream expansion orientation preferences

Upstream Expansion

Expansion sections cause major fluid pattern changes and turbulence as the flow approaches a downstream quarter-turn valve. When short expander laying lengths are used and close coupled, the disc of butterfly valves may interfere with the expander inside diameter when operated. Expanders upstream of quarter-turn valves should be concentric rather than eccentric whenever possible. Similar to the upstream elbows, the distance upstream should be two or more diameters for good performance. If an eccentric arrangement is necessary, the shaft axis should be oriented in the direction of the offset as shown in the lowest orientations of Figure 6-8. Figure 6-8 illustrates preferences and compatibility for several orientation arrangements. Although depicted with butterfly valves, these scenarios apply to all quarter-turn valve installations.

Upstream Orifice and Venturi Flow Meters

Upstream orifices also cause major fluid pattern changes and turbulence as the flow approaches a downstream quarter-turn valve. Orifices upstream of quarter-turn valves should be concentric rather than eccentric whenever possible. Similar to the upstream elbows and expansions, the distance upstream should be two or more diameters for good performance of both the quarter-turn valve and the orifice. In this case, the orifice performance will be altered as well as the quarter-turn valve performance if there is insufficient spacing. This is especially true when the orifice is used as a meter. A close-coupled control valve can alter the pressure tap reading and affect the accuracy of the meter. Venturi meters, which use upstream and throat pressure taps, are unlikely to be affected by a close-coupled downstream control valve. However, Venturi meters, whether full-length flanged or insert, often have reduced exit diameters and can interfere with valve closure member operation if not sufficiently separated.

If an eccentric orifice arrangement is necessary, the shaft axis should be oriented in the direction of the offset as indicated in the lowest line of Figure 6-9. See Figure 6-9 for additional preferences and compatibility. Again, these depictions with butterfly valves apply to all quarter-turn-valve installations.

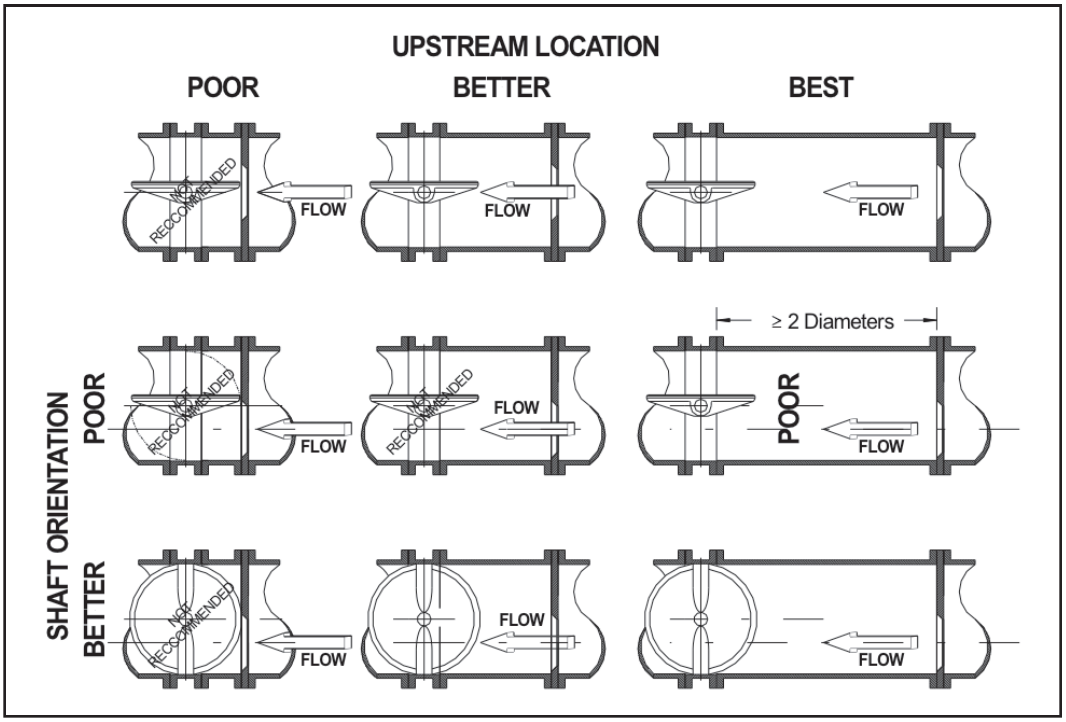


Figure 6-9 Upstream orifice orientation preferences

Upstream Reducers

Upstream reducers can also cause poor valve behavior if not oriented properly. Concentric reducers are best, and again, it is better to maintain a greater distance upstream for good performance. As with the other orientations, it is best that the flow is essentially equal on either side of the shaft axis. Again, these depictions with butterfly valves apply to all quarter-turn valve installations.

If an eccentric upstream reducer arrangement is necessary, the shaft axis should be oriented in the direction of the offset as indicated in the lowest line of Figure 6-10. See Figure 6-10 for additional preferences and compatibility.

Upstream Valves

A common type of valve positioned upstream of a quarter-turn valve is a check valve (Figure 6-11). Less than one diameter of straight pipe between the check valve and the quarter-turn valve will cause these valves to interact. When the check valve has a horizontal pivot shaft (or hinge pin), the quarter-turn valve should be installed with a vertical shaft. This positioning allows the high localized fluid velocities to be divided evenly across the quarter-turn valve's shaft. When two quarter-turn valves are placed in close proximity (less than three diameters), their shafts should be perpendicular to each other so that the upstream valve does not cause excessive torques in the downstream valve. This orientation is demonstrated in Figure 6-12 and is relevant to all quarter-turn valves.

Many valves and other piping components have internal parts or closure members that may extend beyond the end connection during operation. Series-mounted valves and other components with internal parts or closure members should be placed far enough apart to avoid any structural interference. For instance, the centerlines of two series-piped butterfly valves should be at least one diameter apart so that their discs do not touch

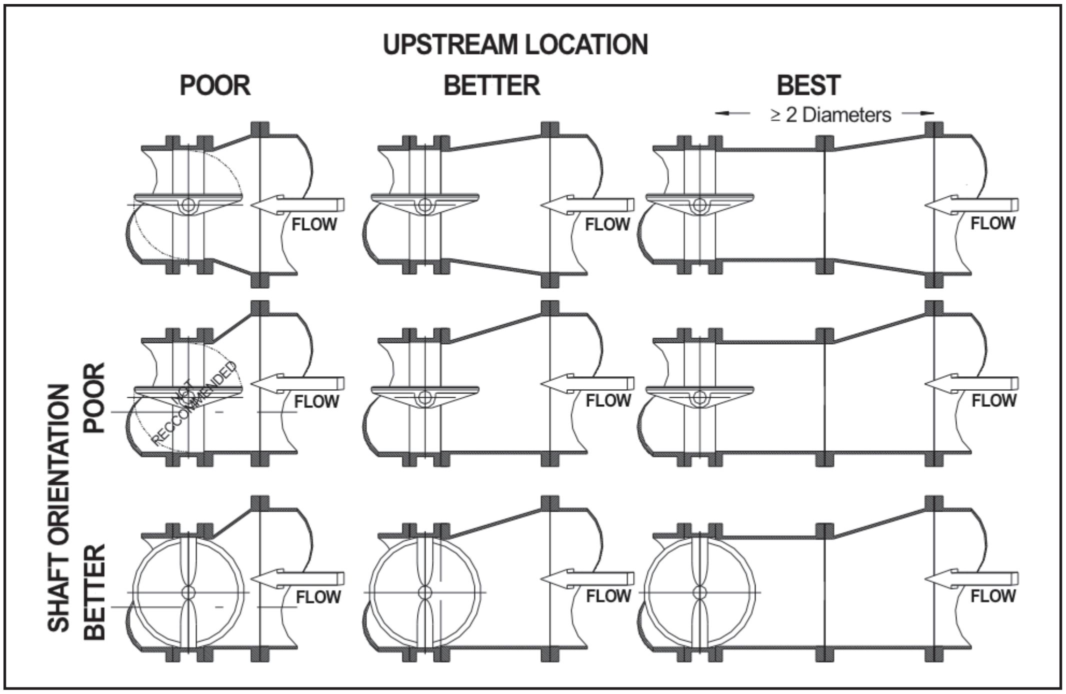


Figure 6-10 Upstream reducer orientation preferences

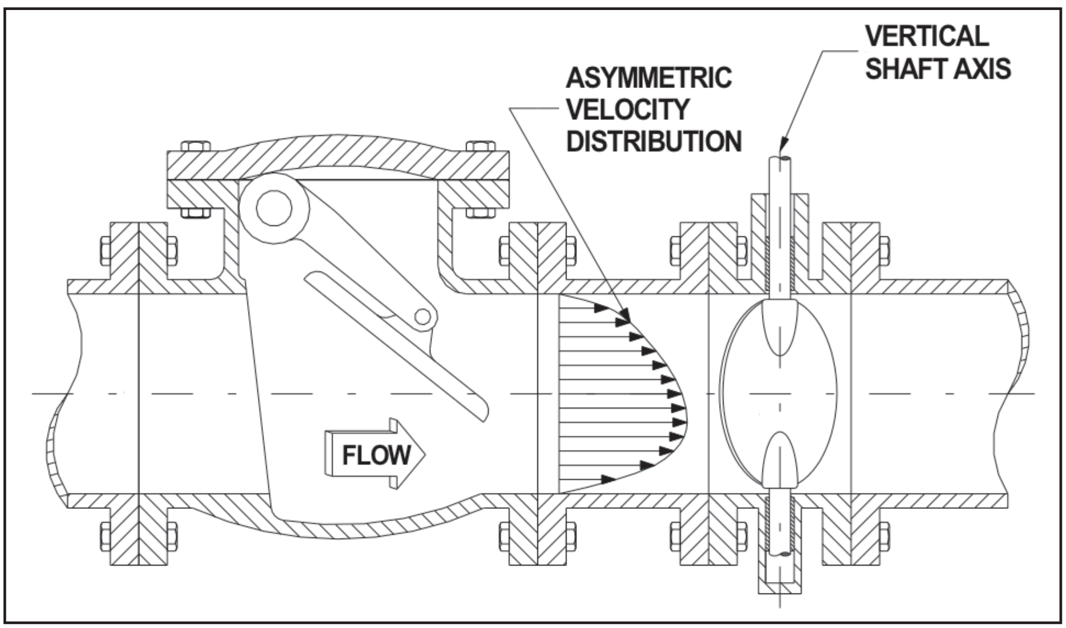


Figure 6-11 Upstream check valve preferred installation orientation

during operation. These valves would function better with two- or three-diameter spacing between centers to reduce the possibility of fluid dynamic interaction.

Further, never combine two or more poor orientations, such as the following: (1) short expansion, (2) improperly oriented shafts, and (3) a highly throttled upstream valve, as illustrated in Figure 6-13.

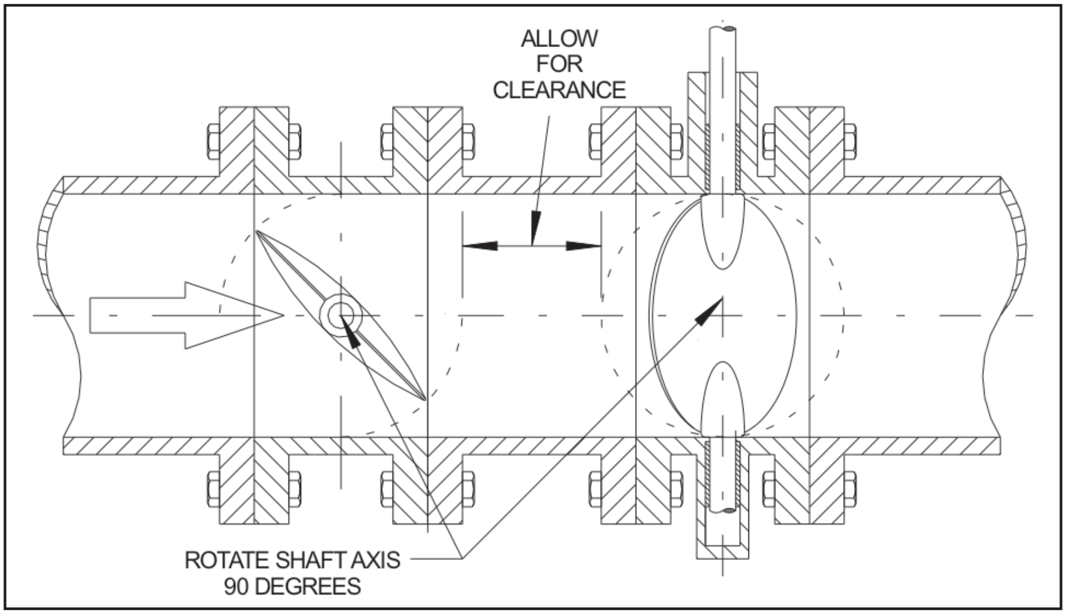


Figure 6-12 Series-mounted valve preferred installation orientation

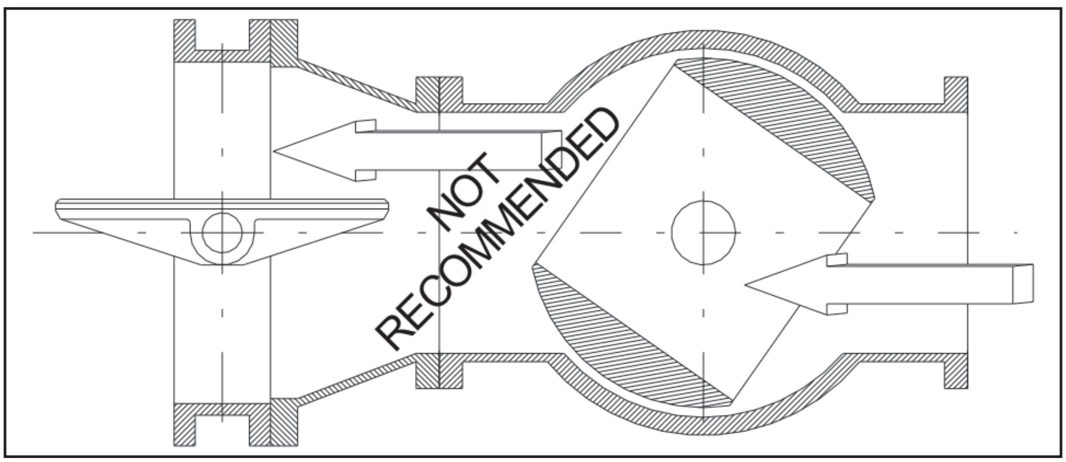


Figure 6-13 Combined improper orientations

Free Inlet or Discharge

Quarter-turn valves are often mounted on the walls or partitions of basins and tanks as in Figure 1-7. The valve manufacturer must be aware of such an installation because special torque and flow coefficients should be used to evaluate these conditions. Also, the direction of flow may be more critical for an asymmetric valve. The valve shaft is usually mounted in the vertical orientation so that partial flows are equally divided across both sides of the shaft.

External and Pipe Loads

Quarter-turn valves are self-contained devices that may not function properly or remain tight if subjected to external forces. Quarter-turn valves are not designed or intended to be pipe supports and should not be used to transmit piping loads to static or dynamic pipe supports. Quarter-turn valves are designed to withstand pressure vessel and normal operating loads and to be stronger than the adjacent piping. Additional external and piping applied loads may affect valve performance and operation. Quarter-turn valves rely on a precise fit between the body and closure member seats to form a low-leakage or leak-tight seal. Although valve bodies are designed to be stronger than the standard or normal mating pipe, they are still susceptible to deflection or distortion from externally imposed and piping applied loads. Any valve deflection or distortion caused by these loads impairs the valve's ability to seat and seal properly. Even if deflection or distortion does not cause the leakage rate to be observably degraded or impaired, the valve's service life and operating margins could be affected.

If a valve is rigidly installed in a pipeline using flanged joints, the whole assembly of pipe and valve can be stressed by temperature changes, settlement, pipe support and (if buried) exceptional surface loads. To prevent a valve being overly externally stressed, strained, and/or distorted, there should be at least one flexible joint installed next to or near the valve. Never connect a flanged valve to an out-of-tolerance, out-of-round pipe or flange. Refer to the flange standards for the bolting dimensional tolerances. The connection and support structure next to the valve shall prevent or at least minimize the piping loads and bending moments from being transmitted through the valve body to stop the potential for overstressing the valve flanges and deflecting the body. Having two flexible couplings near and one on either side of a valve is best if the section between the couplings can be restrained and simply supported, reducing or relieving the transmitted pipe loads to the valve.

For buried applications of large-diameter flanged valves, the designer should consider all pipe, valve, soil, and surface live loads in designing the supports for the adjacent piping. The designer may consider using a common foundation slab and concrete pipe cradles to support the adjacent piping, including the use of flexible pipe joints on either side of the valve to ensure that any pipe or external loads do not distort the valve body. It is the responsibility of the system or piping designer to determine the loading conditions and the required valve and pipe supports for the specific installation. Although these and many other constructions are suitable, it is important that the system and piping designer understand that pipe loads transmitted through the valve causing a small distortion of the valve seat may affect the valve's ability to seal, change the operating torque requirements, alter system fluid flow characteristics, and reduce the valve's service life.

Buried valves normally should use mechanical joint or push-on joint-type fittings that allow for some flexibility and relief of the pipe loads. These joints reduce the piping loads transferred to the valve, but thrust blocks or other restraints must be used to keep the joints from separating. When flanged valves are buried, the valve should be bolted to flanges of the adjacent piping with no other rigid support provided under or around the valve body. In no case should the valve body be supported directly by a saddle or other structure. In this condition, the valve becomes an anchor or a support for the piping system and must transmit piping loads.

Aboveground, exposed, or plant piping support design should carefully avoid additional thrust or bending moments to the valve's end connections. Buried applications should include either valve boxes or vaults that do not transmit piping, traffic, shock, shifting soil, or earth loads and stresses to either the valve or actuator and should not apply additional thrust or bending moments to the valve's end connections.

If external and piping loads are severe enough, the valve may bind at various locations, including shafts, trunnions, seats, packing, and bearings, and the valve may become inoperable.

Installation Concerns and Precautions

When quarter-turn valves have adjustable seating, install the seat-adjustment side of the valve for access and adjustment in service unless the manufacturer's drawings or installation procedure indicate a different required orientation.

When practical, quarter-turn valves in buried installations should be located in vaults. Many types of buried pipes are designed to deflect 2–5 percent of pipe diameter, which is harmful to the valve integrity. Adjacent pipe should be supported and/or stiffened to provide a round mating connection that does not distort the valve body.

Foreign material in a quarter-turn valve can damage the seat when valves are operated. Be sure valve interiors and adjacent piping are cleaned of foreign material prior to making up valve-to-pipe joint connections. Foreign material may also prevent the valve from operating and may damage the actuator.

Prepare pipe ends and install quarter-turn valves in accordance with the pipe manufacturer's instructions for the joint used. Do not deflect a pipe–valve joint. Do not use a quarter-turn valve as a jack to pull pipe into alignment. The installation procedure should minimize the bending of the quarter-turn valve with installation pipe loadings.

When quarter-turn valves are installed in vaults, the vault design should provide adequate space for removal or servicing of the valve–actuator assembly for purposes of repair and for access to adjust the thrust-bearing assembly. The vault should be designed and constructed wide enough such that the doors, covers, or panels on the actuator housing can be opened without the doors or panels striking an adjacent wall, thereby limiting or preventing access to the inside of the actuator housing or control system electronics. The possibility of groundwater or surface water entering the valve and the disposal of the water should be considered. The valve operating nut should be accessible from the top opening of the vault with a tee wrench.

As discussed earlier, single-seated quarter-turn valves may have different shutoff characteristics based on seat orientation and should be installed with the seat oriented for shutoff in the direction preferred or required. Also, valves with adjustable, replaceable, or repairable seats may require access from a specified end of the valve to facilitate the rework. When this orientation, the installed piping arrangement, and the shutoff direction are in conflict, the designer may need to compensate, select a different valve, or select an orientation that is the most preferred of the choices.

Quarter-turn valves in pump control applications should be installed such that the seat orientation best prevents flow back toward the pump.

Buried quarter-turn valves should be installed with the shaft horizontal and the actuator input drive nut upward.

Valve box or extension pipe should be installed so that the actuator input drive nut and extension stem turn freely.

Plug valves in fluids free of suspended solids may be installed in any orientation relative to vertical unless there is an actuator orientation limitation. If practical, plug valves shall be installed so that the pipeline pressure is exerting force on the plug from opposite the seat end of the valves (e.g., direct pressure orientation).

Plug valves used for fluids containing suspended solids should be installed as shown in Figure 6-14. When installed in horizontal pipes, the axis of the closure member shaft is to be horizontal with flow entering the valve body from the seat end, thus keeping the plug in the upper half of the body when rotated (counterclockwise).

Quarter-turn valves in horizontal pipes used with fluids containing suspended solids should be installed with their shafts horizontal so that the sediment does not collect or penetrate into the shaft bearing area and cause binding of the shaft. Even when used with clean fluids, it is often best to avoid vertically oriented shafts unless there are other

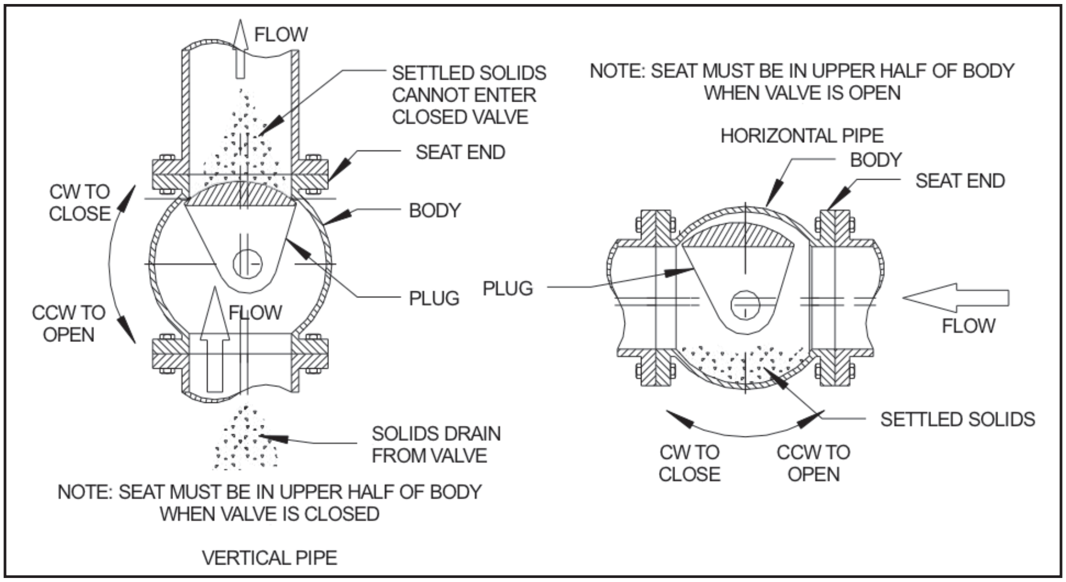


Figure 6-14 Plug valve preferred installation orientation for suspended solids applications

overriding conditions. Valves with horizontally oriented shafts tend to flush the bottom of the pipe for better long-term service.

Whenever possible, the actuator should be mounted to the side or above a horizontal pipe. If it is necessary to mount an actuator below the pipe, the valve or actuator manufacturer should be consulted regarding this orientation.

Double-seated valves may be susceptible to pressure locking between the seats. System designers should review pressure transients or excursions and operating procedures to ensure that extreme pressures or pressure excursions are not captured between the seats that create seat binding, distortion, and high loads are not created. It is not typical for water works systems to have operating thermal cycles with greater than a 50°F temperature change (ΔT). If this should be a system design condition, then the valve construction should be reviewed for both pressure locking and thermal binding caused by differences in material and fluid thermal coefficients of expansion that create higher friction-based torque components that could render the valve inoperable. This is of greatest concern for double-seated valves or with series-redundant valves operated simultaneously. It may be necessary to vent the internal central cavity of the valve or disable one of the seals. Such conditions should be reviewed with the valve manufacturer. Large or frequent thermal cycling can also cause grease to migrate past actuator housing seals. If this is a concern, vents, breathers, or compliant volumes may be needed.

TYPICAL RANGE OF SOME COEFFICIENTS

As many users of this methodology may not have intimate knowledge of the various factors and coefficients involved, the following provides some direction regarding typical values found in the industry. These data were taken from a variety of sources and are not to be considered as being representative of any single valve or manufacturer. These are provided as informational only and should be understood as being only nominal representations.

An attempt to normalize was made, and outliers were removed as being not representative. All values shown are for valves and materials in good, well-maintained condition

Table 6-1 Typical Bearing Friction Coefficients

Material Type	Low Value C_f	Mean Value C_f	High Value C_f
Nonmetallic	0.07	0.12	0.25
Metallic	0.125	0.25	0.35

Table 6-2 Typical Packing Coefficients

Low Value $C_{pck\ f}$	Mean Value $C_{pck\ f}$	High Value $C_{pck\ f}$
30 lb/in. (5,254 N/m)	400 lb/in. (70,000 N/m)	1,200 lb/in. (210,000 N/m)
Low Value $C_{pck\ tq}$	Mean Value $C_{pck\ tq}$	High Value $C_{pck\ tq}$
100 lb (444 N)	450 lb (2,002 N)	700 lb (3,114 N)
Low Value μ_P	Mean Value μ_P	High Value μ_P
0.10	0.20	0.30

Table 6-3 Typical Seating Coefficients

Valve Type	Low Value C_{SC} lb/in. (N/m)	Mean Value C_{SC} lb/in. (N/m)	High Value C_{SC} lb/in. (N/m)
BV, Resilient Seats	10 (0.054)	24 (0.137)	40 (229)
BFV, Resilient Seats	6 (0.034)	16 (0.091)	36 (0.206)
Valve Type	Low Value C_{SP} lb/in./psi (N/m/kPa)	Mean Value C_{SP} lb/in./psi (N/m/kPa)	High Value C_{SP} lb/in./psi (N/m/kPa)
BV, Resilient Seats	0.08 (2.03)	0.12 (3.05)	0.26 (6.60)
BFV, Resilient Seats	–0.02 (–0.51)	0.02 (0.51)	0.05 (1.27)

and do not represent heavily degraded or misapplied conditions. These data are only provided for instructional and comparative use.

Bearing Friction

Table 6-1 provides some typical ranges for the bearing coefficient of friction, C_f, which may vary widely with the bearing and shaft materials. All valves within the scope of this report have iron, steel, or stainless-steel shafts, but the bearing material may be of many types. In general, nonmetallic bearings are designed to have a lower coefficient of friction than their metallic counterparts. Also, some nonmetallic bearing materials have coefficients of friction that vary with the applied load.

Packing Friction

Table 6-2 provides typical ranges for the packing coefficients, $C_{pck\ f}$ and $C_{pck\ tq}$, and coefficient of friction, μ_P, which may vary widely with the packing type and materials. Many packing manufacturers provide values and procedures to determine packing loads.

Seating Coefficients

Table 6-3 provides some typical ranges for the seating coefficients, C_{sc} and C_{SP}, which also vary widely by design and materials.

Table 6-4 Typical Full-Open Flow Coefficients

Valve Type	Low Value C_V	Mean Value C_V	High Value C_V
BV & RCV	$118.9 \times D^2$	$139.2 \times D^2$	$188.8 \times D^2$
BFV	$32.4 \times D^2$	$41.0 \times D^2$	$54.5 \times D^2$
PV, Red. Port	$31.5 \times D^2$	$33.5 \times D^2$	$39.9 \times D^2$
PV, Full Port	$38.9 \times D^2$	$44.8 \times D^2$	$57.4 \times D^2$
Valve Type	High Value K_V	Mean Value K_V	Low Value K_V
BV & RCV	0.063	0.046	0.025
BFV	0.85	0.53	0.30
PV, Red. Port	0.90	0.79	0.56
PV, Full Port	0.59	0.44	0.27

Flow Coefficients

Table 6-4 provides typical ranges for the flow coefficients, C_V and K_V, of quarter-turn valves. These coefficients can be expressed, grouped, and graphed in many ways. First, the full-open butterfly valve flow resistance coefficient, K_V, is normally between 0.30 and 0.85. The C_V and K_V are inversely related, so the high K_V corresponds to the low C_V. Low-pressure valves generally have lower full-open flow resistance (K_V) values.

Butterfly valves have inherent characteristic curves as given in Figure 2-4, which approach those of equal percentage control valves. These generalized curves can be used to calculate the C_V and K_V for butterfly valves at intermediate angles between full open and full closed.

Dynamic Torque Coefficients

Figure 3-15 provides typical ranges for butterfly valve dynamic torque coefficients, C_t, which vary widely by design and materials. The combination of the flow and torque coefficients should be from a matched set of data. It should not be assumed that valves with high flow resistance valves have low torque coefficient values. These data are generic and representative, but use of these data will not be representative of a specific valve.

CAUTIONS

Valves installed contrary to the recommendations in the previous section or in configurations subject to significant nonuniform or swirling upstream flow may develop torque requirements or stresses in excess of those generally assumed in sizing valve shafts, disc connections, and actuators. As a consequence, electric actuators may stall, and other components may fail over time as a result of metal fatigue. Failure of any component (shaft or coupling) that connects the disc to the actuator mechanism may cause the disc to slam closed with resulting damage to the valve and possibly severe water hammer and pipe damage.

Such an installation requires actuators with higher torque ratings and stronger valve components when improvements in upstream piping conditions are not feasible.

Actuator Speed of Operation

Fast valve opening and closure rates can impose large pressure transients in piping systems. The AWWA quarter-turn valve standards typically impose an operating time for power actuators of about 2 to 4 s/in. of nominal valve size. This is based on reasonable operating power requirements for the actuator and low transient pressures in a pipeline up to about 4,000 diameters in length. Operating speeds should be reviewed by the user when pipe lengths are near or exceed this limit or when other operational requirements involve the use of faster speeds.

Actuator Servicing and Removal

The actuator should not be serviced or removed from installed valves unless the pipeline on both sides of the valve is de-energized (i.e., no pressure and drained). A quarter-turn valve closure member that is not restrained by the actuator may move, slam open, or slam closed, causing damage. Extreme caution must also be used when examining or working on a quarter-turn valve in the line. The actuator must be locked out to prevent unexpected travel. An installed valve without an actuator is a dangerous valve. An offset-style closure member may tend to open or close because of its offset and its closure member's center of gravity when installed. Similarly, hydrostatic force (discussed in chapter 3) may also open an unsecured valve with a horizontal shaft in a horizontal line with water on one side of the closure member. Personnel should not enter a large-diameter pipe where any valve in the pipe has an unsecured actuator or a removed valve actuator. Personnel should not enter a large-diameter pipe where the actuator has not been locked and tagged out.

Seat Servicing and Removal

Valve seats should only be serviced or replaced with both sides of the valve de-energized. Many industrial safety jurisdictions or authorities consider valves to be pressure vessels that require the valve to be in a state in which there is no pressure and no fluid on both sides when working on any pressure element or appurtenant component. Should valve seats be serviced or repaired inline, consideration must be given to confined-space entry requirements of the governing municipality, jurisdiction, or safety authority, particularly the regulations concerning single- or double-block-and-bleed isolation. When possible, it is recommended that the valve be removed from the adjacent piping to repair or replace the valve seats.

Throttling Flow

Quarter-turn valves have good flow characteristics and are often used for throttling flow. However, quarter-turn valves are usually limited to a throttling range of 15° to 75° open (15 percent to 85 percent). Operating valves at positions less than 15° (15%) open may cause high localized velocities and cavitation, which can damage the seating surfaces.

As discussed in chapter 4, cavitation can be observed by detecting a rumbling noise immediately downstream from the valve similar to rocks flowing through the line or by the use of an accelerometer attached to the pipe. Cavitation is a result of excessive pressure drop across the valve combined with low downstream pressure. When the localized pressure downstream of the disc falls below the vapor pressure of water (typically about 0.5 psia [3.5 kPa] for cold water), water vapor bubbles will form and then violently implode downstream as the pressure recovers (see Figure 4.1).

SUMMARY

The issues presented in this manual will assist users, system engineers, and designers to understand quarter-turn valve characteristics. The calculations, recommendations, applications, and valve installation precautions presented herein will provide the user and designer with the most effective and trouble-free application of this type of valve. Because of the engineering complexity of this subject matter and the size dependency of the many variables, their individual impact increases with valve size. Greater effort should be placed on the details, the methods, the effects, and the results as the valve size increases.

REFERENCES

American Water Works Association. 2008. ANSI/AWWA C541-16: Standard for Hydraulic and Pneumatic Cylinder and Vane-Type Actuators for Valves and Slide Gates. Denver, CO: AWWA.

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American Water Works Association. 2015. *Cylinder and Vane Actuators and Controls – Design and Installation*. AWWA Manual M66. Denver, CO: AWWA.

American Water Works Association (AWWA). 2015. ANSI/AWWA C504-15: Standard for Rubber-Seated Butterfly Valves. Denver, CO: AWWA.

American Water Works Association (AWWA). 2015. ANSI/AWWA C507-15: Standard for Ball Valves, 6 In. Through 60 In. (150 mm Through 1,500 mm). Denver, CO: AWWA.

American Water Works Association. 2015. ANSI/AWWA C516-14: Standard for Large-Diameter Rubber-Seated Butterfly Valves, 78 In. (2,000 mm) and Larger. Denver, CO: AWWA.

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Appendix A

SPECIFICATION AND DATA SHEET

System

1. Owner/client: ____________________
2. Project/site: ____________________
3. Service/system description: ____________________
4. Equipment ID or tag no.(s): ____________________
   ____
5. Type of installation: (select one)
   ☐ buried ☐ submerged ☐ non-buried/submerged
6. Installed location type: (select one)
   ☐ indoors ☐ outdoors ☐ vault
   ☐ other/data: ____________________
7. System fluid: (select one)
   ☐ water ☐ wastewater ☐ gas ____________________
   ☐ other/data: ____________________
8. Max. non-shock line pressure: ____________________
9. Max. non-shock closed differential pressure: ____________________
10. If known, maximum transient pressure, rate, characteristics: ________
11. Port fluid flow or velocity through valve & units:
    a. Normal conditions: ____________________
    b. Maximum flow conditions: ____________________
    c. Emergency opening: ____________________
    d. Emergency closing: ____________________
12. Fluid temperature range: ____________________
13. Flow direction (select a + i, a + ii, a + iii, b, or c):
    a. ☐ Unidirectional flow
       i. ☐ Manufacturer's best or preferred (TBD)
       ii. ☐ Direct pressure/shaft-side flow
       iii. ☐ Reverse pressure/seat-side flow
    b. ☐ Bidirectional flow
    c. ☐ Pump check operation
14. Service: (select one)
    ☐ on/off ☐ modulating ☐ throttling (set & held)
    For modulating or throttling provide (with units):
    a. Condition 1: flow: ____________________
       Upstream P: ____________________
       Downstream P: ____________________
    b. Condition 2: flow: ____________________
       Upstream P: ____________________
       Downstream P: ____________________
    c. Condition 3: flow: ____________________
       Upstream P: ____________________
       Downstream P: ____________________
    d. Condition 4: flow: ____________________
       Upstream P: ____________________
       Downstream P: ____________________

Installation (if known)

1. Description of connecting piping: ____________________
   ____
2. Installed pipe orientation: ☐ vertical ☐ horizontal
   ☐ other: ____________________
3. Installed shaft orientation: ☐ vertical ☐ horizontal
   ☐ other: ____________________

Valve

1. Type: (select one) ☐ ball (BV) ☐ butterfly (BFV)
   ☐ cone (RCV) ☐ plug (PV)
2. Size of the valve & units: ____________________
3. Press. class/vel. designation (AWWA): ____________________
4. Quantity required: ____________________
5. Type of body end connection: (select one)
   ☐ flanged ☐ wafer ☐ MJ
   ☐ other: ____________________
6. Special materials or material requirements
   a. Body: ____________________
   b. Closure member: ____________________
   c. Metal seat(s): ____________________
   d. Resilient seat: ____________________
   e. Bearing: ____________________
   f. Shaft: ____________________
   g. Shaft seals: ____________________
7. If BV or RCV, ☐ double-seated or ☐ single-seated

Actuator Mounting

1. Type (select one):
   ☐ direct ☐ replaceable packing yoke or bonnet
   ☐ extended actuator yoke or bonnet
   C/L valve to C/L actuator input: ____________________
   ☐ extended actuator floor stand & torque tube
   C/L valve to floor: ____________________
   ☐ other: ____________________

☐ Manual Actuator

1. Type (select one or more):
   ☐ direct (lever / nut) ☐ worm gear ☐ traveling nut ☐ other
2. Input (select one or more):
   ☐ lever ☐ crank ☐ 2-in. nut ☐ chain lever
   ☐ handwheel ☐ chainwheel
3. Opening direction (select one):
   ☐ std.-CCW (open lft) ☐ non-std.-CW (open rt)
4. Input accessories/devices (select one or more):
   ☐ Extension shaft, length: ____________________
   ☐ Floor stand
   ☐ Special handwheel diameter: ____________________
   ☐ Special position indicator: ____________________
5. Output accessories/devices (select one or more):
   ☐ Limit switches, qty./type: ____________________
   ☐ Position transmitter: ____________________

☐ Power Actuator

Attach separate data sheet. See AWWA C541 or C542.

Coatings & Other Special Requirements

____
____
____
____
____

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Appendix B

EXAMPLE OF FIRST-PRINCIPLES DERIVATION OF SEATING TORQUE FOR CIRCULAR SEAT

Let:

C_s = Seating friction load, force/unit length (i.e., lb force/in. or Newtons/m)

D = Diameter of the line of closure member seat contact = 2 × R, unit length (i.e., in. or m)

L = Effective lever length = R × sinθ, unit length (i.e., in. or m)

R = Radius of the line of disc seat contact, unit length (i.e., in. or m)

T_{seat} = Seating torque, force – unit length (i.e., lb force – in. or Newton m)

W = Friction load at seat contact circle, unit force = $C_S \times R \times \delta\theta$

Θ = Angle about the seat contact circle, radians

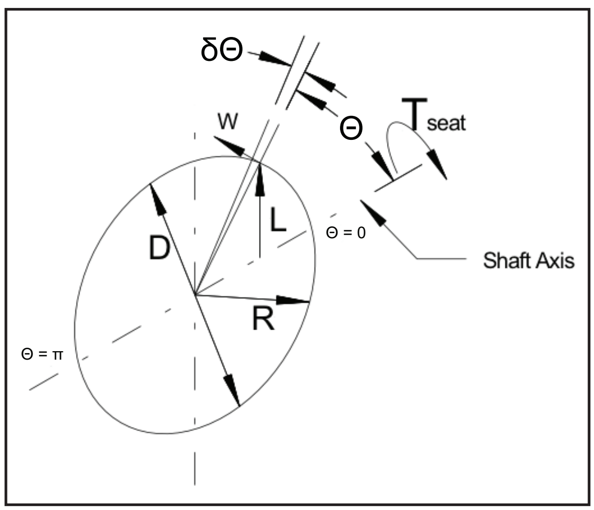

Figure B-1 Seating torque free body diagram

$$T_{seat} = 2 \times \int_0^{\pi} L \times W$$

$$T_{seat} = 2 \times \int_0^{\pi} (R \sin\theta) \times (C_S \times R \times d\theta)$$

$$T_{seat} = 2 \times C_S \times R^2 \int_0^{\pi} \sin\theta d\theta$$

$$T_{seat} = 2 \times C_S \times R^2 \times [-\cos\theta]_0^{\pi}$$

$$T_{seat} = 2 \times C_S \times R^2 \times (1) - (-1)$$

$$T_{seat} = 4 \times C_S \times R^2$$

$$T_{seat} = C_S \times D^2$$

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Appendix C

NOMENCLATURE

Table 1 provides the variable symbols and definitions used in the manual. Table 2 provides unit conversion between US customary and SI (metric). Table 3 provides abbreviations used in this manual.

Table C-1 Nomenclature, terms, and symbols

Term, Variable, or Symbol	Definition or Description	Units US customary (SI metric)
±	Indicates a torque sign convention that may be assigned by the user. For valve-active torque components, + indicates when the torque tends to close the valve, and – indicates when the torque tends to open the valve. For torque transmitted to the actuator, + indicates when valve shaft torque opposes actuator motion, and – indicates when valve shaft torque assists actuator motion (actuator acts as a brake).	dimensionless
0 though 90	Subscript indicating the valve closure member angle. The convention used in this manual is that closed = 0° (0 radians); full open ≈ 90° (π/2 radians)	degrees (radians)
A_{COST}	Annual energy cost	$/year
A_{PIPEID}	Pipe inside flow area	in.^{2} (mm^2)
A_{port}	Valve mean port area based on the center or mean seating point around the perimeter of the seat.	in.^{2} (mm^2)
C	Cost of electricity	$/kWh
C_f	Coefficient of friction between the shaft or trunnion and bushing, dimensionless. (This value may be obtained from a flow test, engineering handbooks, the bearing manufacturer, or the valve manufacturer.)	dimensionless
C_g	Valve disc center of gravity distance from shaft centerline	in. (mm)
Closing torque	Test measured torque in the closing direction	in.-lb (N-m)
$C_{pck\ f}$	Packing force coefficient	lb/in. (N/m)
$C_{pck\ t}$	Packing torque coefficient	lb (N)
C_s	Coefficient of seating, general form used in the first edition	lb/in. (N/m)
C_{sc}	Constant or pressure-independent coefficient of seating torque	lb/in. (N/m)
C_{sp}	Pressure-dependent coefficient of seating torque	lb/in./psi (N/m/kPa)
C_t	Coefficient of dynamic torque (positive value tends to close valve), general form	dimensionless

Table continues next page

Table C-1 Nomenclature, terms, and symbols (continued)

Term, Variable, or Symbol	Definition or Description	Units US customary (SI metric)
$C_{t\theta}$	Coefficient of dynamic torque at valve angle θ (positive value tends to close valve)	dimensionless
C_{usc}	Constant or pressure-independent coefficient of unseating torque	lb/in. (N/m)
C_{usp}	Pressure-dependent coefficient of unseating torque	lb/in./psi (N/m/kPa)
C_v	Valve flow coefficient. The flow of water through a valve at 60°F in US gallons per minute at a pressure drop of 1 psi (lb/in.^{2}) Metric units note: In metric units, this variable is often identified as K_V. However, this is not used in this manual as it is easily confused with the resistance coefficient, K. When the resistance coefficient, K, is the resistance coefficient of the valve, it is subscripted with a "V" to indicate this reference, K_V. The metric flow coefficient, Kv, is defined as the flow of water with temperature ranging 5–30°C through a valve in cubic meters per hour (m^3/h) with a pressure drop of 1 bar (1 bar = 100 kPa). For purposes of this manual, the metric unit version of C_v will be identified by the variable symbol C_{vm}.	gpm/psi$^{1/2}$ None
C_{vm}	The metric equivalent to C_v (referred to as K_v in other texts)	None (m^3/hr/Bar$^{1/2}$) (m^3/hr/(100 kPa)$^{1/2}$)
D	Nominal valve diameter	in. (mm)
D_1	Reducer reduced pipeline diameter	in. (mm)
D_2	Reducer large pipeline diameter	in. (mm)
D_d	Disc diameter for butterfly valves or nominal diameter (D) or port diameter (D_{port}) for ball valves	in. (mm)
D_{PIPEID}	Pipe inside diameter	in. (mm)
D_{REF}	Reference valve size	in. (mm)
d_s	Shaft diameter. For trunnion-mounted valves, use trunnion diameter (d_{tr}) in bearing friction calculations.	in. (mm)
d_{sm}	Mean shaft diameter. Top or actuator shaft and the bottom or blind shaft may be different diameters.	in. (mm)
d_t	Size of model or test valve	in. (mm)
d_{tr}	Trunnion diameter for trunnion-mounted obturators	in. (mm)
E	Efficiency of pump and motor set (80%; 0.80, typical)	%
$F_{L\theta}$ (and $F_{L\theta}^2$)	Liquid pressure recovery factor of a valve without attached fittings. This experimentally determined factor depends upon the internal valve geometry.	dimensionless
F_R	Resultant force vector for hydrostatic torque	lb (N)

Table continues next page

Table C-1 Nomenclature, terms, and symbols (continued)

Term, Variable, or Symbol	Definition or Description	Units US customary (SI metric)
g	Gravitational constant Acceleration due to gravity, 32.2 ft/sec^2 (9.81 m/sec^2)	ft/sec^2 (m/sec^2)
G	Additional seismic acceleration loading multiplier. For applications involving additional seismic loading, component weight may be multiplied by G (horizontal) or G ± 1 (vertical), where G is the additional seismic acceleration.	dimensionless
H_P	Packing height	in. (mm)
K	Flow resistance coefficient of any component or fitting	dimensionless
K_1	Reducer resistance coefficient based on the reduced diameter, D_1	dimensionless
K_2	Reducer resistance coefficient based on the large diameter, D_2	dimensionless
K_{sys}	System flow resistance coefficient (excluding the valve)	dimensionless
K_v	Flow resistance coefficient of the valve	dimensionless
K_{V90}	Flow resistance coefficient of valve at full open (≈ 90°, ≈ π/2 radians). Note: Use of KV90 assumes the valve travels 90° to full open.	dimensionless
K_{Vd1}	Valve resistance coefficient based on the reference diameter, D_1 (typically the nominal diameter, D)	dimensionless
K_{Vd2}	Valve resistance coefficient based on the diameter, D_2	dimensionless
$K_{V\theta}$	Flow resistance coefficient of the valve at valve angle θ	dimensionless
L	Reducer end to end length	in. (mm)
Opening torque	Test-measured torque in the opening direction	in.-lb (N-m)
P_C	Pressure class or maximum design pressure (the greater of)	dimensionless (or psig)
P_d	Reference downstream pressure for cavitation analysis	psi (kPa)
P_{EW}	Pressure equivalent to disc and shaft weight	psi (kPa)
P_{ml}	Mean seat perimeter length, see Figure 2-11	in. (mm)
P_{SE}	Pressure scale effects factor for cavitation analysis	dimensionless
P_u	Reference upstream pressure for cavitation analysis	psi (kPa)
P_{ut}	Upstream pressure from laboratory test for cavitation analysis	psi (kPa)
P_v	Vapor pressure adjusted for temperature and atmospheric pressure (Example: P_v = –14.4 psig (–99.6 kPa) for water at 60°F (16°C), measured at sea level.	psi (kPa)
P_{vt}	Vapor pressure from laboratory test	psi (kPa)
Q	Volumetric flow rate	gpm (m^3/h)
Q_{MAX}	Maximum volumetric flow rate	gpm (m^3/h)

Table continues next page

Table C-1 Nomenclature, terms, and symbols (continued)

Term, Variable, or Symbol	Definition or Description	Units US customary (SI metric)
R_p	Swing radius of the plug valve's rubber plug face, see Figure 3-9	in. (mm)
S_c	Sign convention variable: For torque active components, +1 when the torque tends to close the valve or –1 when the torque tends to open the valve. For center of gravity torque, the sign convention variable is +1 when the center of gravity is above the horizontal when the disc is in the full open position or –1 when the center of gravity is below the horizontal when disc is full open. For torque transmitted to the actuator, a positive value when valve shaft torque opposes actuator motion or negative value when valve shaft torque assists actuator motion (actuator is acting as a brake).	dimensionless
S_g	Specific gravity of liquid relative to water at 60°F (16°C) (water = 1.0)	dimensionless
SSE	Sizing scale effects factor for cavitation analysis	dimensionless
sys	Subscript indicating system piping and components less the butterfly valve	
$T_{b0°}$	Bearing torque at valve angle 0° (always positive)	in.-lb (or ft-lb) (N-m)
T_{bt}	Measured bearing torque from testing	in.-lb (N-m)
$T_{btotal\theta}$	Total bearing torque at valve angle θ with addition of disc weight relative to installation orientation (always positive)	in.-lb (or ft-lb) (N-m)
T_{bw}	Bearing torque from disc weight relative to installation orientation (always positive)	in.-lb (or ft-lb) (N-m)
$T_{b\theta}$	Bearing torque at valve angle θ (always positive)	in.-lb (or ft-lb) (N-m)
$T_{cg\theta}$	Center of gravity torque at valve angle θ (positive value tends to close the valve; negative value tends to open the valve)	in.-lb (or ft-lb) (N-m)
$T_{cg0°}$	Center of gravity torque at valve angle 0° (positive value tends to close the valve; negative value tends to open the valve)	in.-lb (or ft-lb) (N-m)
$T_{d\theta}$	Dynamic torque at valve angle θ (positive value tends to close the valve; negative value tends to open the valve)	in.-lb (or ft-lb) (N-m)
T_{dt}	Measured dynamic torque from testing (A positive value indicates a tendency to close the valve.)	in.-lb (N-m)
T_{ecc}	Eccentricity torque (positive value tends to close the valve; negative value tends to open the valve) Note: Only considered at the seated position during opening or at closing.	in.-lb (or ft-lb) (N-m)
T_h	Hydrostatic torque (positive value tends to close the valve; negative value tends to open the valve) Note: Only considered at the seated position during opening or at closing.	in.-lb (or ft-lb) (N-m)
T_p	Packing and hub torque (always positive)	in.-lb (or ft-lb) (N-m)
T_{pt}	Measured packing and hub torque from testing	in.-lb (N-m)
T_s	Seating torque (always positive)	in.-lb (or ft-lb) (N-m)

Table continues next page

Table C-1 Nomenclature, terms, and symbols (continued)

Term, Variable, or Symbol	Definition or Description	Units US customary (SI metric)
T_{st}	Measured seating torque from testing (always positive)	in.-lb (N-m)
$T_{t\theta}$	Total operating torque at valve angle θ (positive value opposes actuator motion; negative value assists actuator motion), general form	in.-lb (or ft-lb) (N-m)
$T_{tc\theta}$	Total closing torque a valve angle θ (positive value opposes actuator motion; negative value assists actuator motion)	in.-lb (or ft-lb) (N-m)
$T_{to\theta}$	Total opening torque a valve angle θ (positive value opposes actuator motion; negative value assists actuator motion)	in.-lb (or ft-lb) (N-m)
T_{ts}	Total seating torque (positive value opposes actuator motion; negative value assists actuator motion)	in.-lb (or ft-lb) (N-m)
T_{tus}	Total unseating torque (positive value opposes actuator motion; negative value assists actuator motion)	in.-lb (or ft-lb) (N-m)
T_{us}	Unseating torque (always positive)	in.-lb (or ft-lb) (N-m)
T_{ust}	Measured unseating torque from testing (always positive)	in.-lb (N-m)
U	Pump usage percentage, 100% (1.0) equals 24 hours per day	%
U_{C1}	Units conversion factor: US customary for torque in in. lb: U_{C1} = 1 in./in. US customary for torque in ft lb: U_{C1} = 1⁄12 (0.0833) ft/in.h Metric for torque in N-m: $U_{C1} = 1 \times 10^{-3}$ (0.001) m/mm	in./in. (ft/in.) (m/mm)
U_{C2}	Units conversion factor: US customary for torque in in. lb: U_{C2} = 1 in./in. US customary for torque in ft lb: U_{C2} = 1⁄12 (0.0833) ft/in. Metric for torque in N-m: $U_{C2} = 1 \times 10^{-6}$ (0.000001) m^2/mm^2	in./in. (ft/in.) (m^2/mm^2)
V	Velocity of fluid flow	feet per second, fps or ft/s (meters per second, mps or m/s)
v	Subscript indicating relative to the quarter-turn valve	
V_{MAX}	Maximum full-open velocity, ft/s (Note: Based on nominal valve diameter if converted from a quantity flow rate)	feet per second, ft/s (meters per second, m/s)
V_V	Approach fluid velocity of the valve	feet per second, ft/s (meters per second, m/s)
V_θ	Approach velocity of fluid flow at valve angle θ	feet per second, ft/s (meters per second, m/s)
W_d	Weight of valve disc	lb (kg)

Table continues next page

Table C-1 Nomenclature, terms, and symbols (continued)

Term, Variable, or Symbol	Definition or Description	Units US customary (SI metric)
$W_{d\&s}$	Weight of the disc and shaft(s) assembly (banjo). For applications involving additional seismic loading, Wd&s may be multiplied by G or (G ± 1), where G is the additional gravitational acceleration multiplier.	lb (kg)
W_s	Weight of shaft(s)	lb (kg)
Y	Size scale exponent for cavitation analysis	dimensionless
α	Reducer (increaser) included angle, degrees; for angles ≤ 45° (◎/4 radians]	degrees (radians)
β	Pipe angle from vertical axis for center of gravity relative to seat location, 0° to 90° (or 0 to π/2 radians) when seated position is above horizontal; >90° to 180° (or π/2 to π radians) when seated position is below horizontal.	degrees (radians)
β_r	For reducer flow resistance calculation: Reducer (increaser) diameter ratio	dimensionless
γ	Center of gravity offset angle (nonsymmetric disc designs) May also include an adjustment for valve designs in which the seating angle is not perpendicular to the pipe axis.	degrees (radians)
ΔH	Head loss between any two reference points in a system	feet of water (meters of water)
ΔH_{MAX}	Head loss across the closed valve or total system with valve closed	feet of water (meters of water)
ΔH_{pipe}	Measured head loss across the pipe during testing without the valve	feet of water (meters of water)
ΔH_{SYS}	Head loss across the system	feet of water (meters of water)
ΔH_t	Measured head loss across the valve and pipe during testing	feet of water (meters of water)
ΔH_V	Head loss across the valve, general	feet of water (meters of water)
$\Delta H_{V\theta}$	Head loss across the valve at angle θ	feet of water (meters of water)
ΔP	Pressure drop (or loss) between any two reference points in a system	psid (kPa)
ΔP_{MAX}	Maximum pressure drop (or loss) across the closed valve or total system with valve closed	psid (kPa)
ΔP_t	Measured pressure drop across the disc from testing	psid (kPa)
ΔP_V	Pressure drop (or loss) across the valve, general form	psid (kPa)
$\Delta P_{V\theta}$	Pressure drop (or loss) across the valve at valve angle θ	psid (kPa)

Table continues next page

Table C-1 Nomenclature, terms, and symbols (continued)

Term, Variable, or Symbol	Definition or Description	Units US customary (SI metric)
ΔP_θ	Pressure drop (or loss) while at disc angle θ	psid (kPa)
ε_1	Disc axial offset (Note: ε_1 equals 0 for symmetric disc designs) See Figure 2-3.	in. (mm)
ε_2	Disc lateral offset Note: ε_2 equals 0 for symmetric or single offset disc designs. Sign convention note: For hydrostatic torque, ε_2 is positive when oriented above the valve centerline and negative when oriented below the valve centerline. See Figure 2-3	in. (mm)
θ	Valve opening position angle, closed = 0° (0 radians); full open ≈ 90° (π/2 radians) Or, as a subscript indicating valve opening position angle for a variable; fully closed equals 0° (0 radians), fully open normally equals 90° (π/2 rad) Note: Some designs may not travel the full 90° (π/2 rad) to the full open position.	degrees (radians)
μ_P	Packing coefficient of friction (typically 0.1 to 0.3)	dimensionless
ν	Packing radial stress to axial stress transfer ratio (typically assumed to be ◎ 0.5)	dimensionless
ϱ	Fluid (weight) density Note: Standard water density is considered as 62.43 lb/ft^3 (1,000 kg/m^3) See Crane. 2009. Flow of Fluids Through Valves, Fittings and Pipe, Technical Paper No. 410	lb/ft^3 (kg/m^3)
σ	Cavitation index, general form	dimensionless
σ_c	Constant cavitation index at a reference pressure, P_u	dimensionless
σ_{ct}	Constant cavitation index from laboratory testing	dimensionless
σ_i	Incipient cavitation index at a reference pressure, P_u	dimensionless
σ_{it}	Incipient cavitation index from laboratory test	dimensionless
ϕ	Valve installed shaft angle from vertical axis, 0° to 90° (or 0 to π/2 radians)	degrees (radians)
Ω	Pipe angle from vertical axis for hydrostatic and bearing torque, 0° to 90° (or 0 to π/2 radians) for flow downhill; 90° to 180° (or π/2 to π radians) for flow uphill	degrees (radians)
W_{seat}	Seat width	in. (mm)
K	Seating compression coefficient, typically equal to 2	none
μ_{seat}	Seat friction coefficient	none

Table C-2 Conversion of Units

US Customary	=	SI (Metric)	SI (Metric)	=	US Customary
1 ft	=	0.3048 m	1 m	=	3.28084 ft
1 ft-lb	=	1.355818 N-m	1N-m	=	0.737562 ft-lb
1 gal (liquid)	=	3.785412×10^{-3} m^3	1 m^3	=	264.172037 gal (liquid)
1 gal (liquid)	=	3.785412 L	1 L	=	0.264172 gal (liquid)
1 gpm (liquid)	=	0.227124 m^3/hr	1 m^3/hr	=	4.402867 gpm (liquid)
1 in.	=	25.4 mm	1 mm	=	0.03937 in.
1 in.	=	0.0254 m	1 m	=	39.370078 in.
1 in.-lb	=	0.112985 N-m	1 N-m	=	8.850745 in.-lb
1 lb	=	4.448222 N	1 N	=	0.224809 lb
1 lb	=	0.4535924 kg	1 kg	=	2.204622 lb
1 lb/ft^3	=	16.01846 kg/m^3	1 kg/m^3	=	0.062428 lb/ft^3
1 lb/in.	=	175.12685 N/m	1 N/m	=	0.00571015 lb/in.
1 lb/in./psi	=	25.4 N/m/kPa	1 N/m/kPa	=	0.03937001 lb/in./psi
1 psi	=	6.894757 kPa	1 kPa	=	0.145038 psi
1 psi	=	0.06894757 Bar	1 Bar	=	14.5038 psi

Table C-3 Abbreviations

Abbreviation	Description
AF	application factor
ANSI	American National Standards Institute
API	American Petroleum Institute
ASME	American Society of Mechanical Engineers
AST	actuator sizing torque
AWWA	American Water Works Association
BFV	butterfly valve
BV	ball valve
CWP	cold working pressure
ISA	International Society for Automation
ISO	International Standards Organization
MRST	minimum required shaft torque
MSS	Manufacturers Standardization Society of the Valve and Fittings Industry
NPS	nominal pipe size
PV	plug valve
RCV	rotary cone valve

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NOTE: *f.* indicates figure; *t.* indicates table

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M2, *Instrumentation and Control*, #30002
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M4, *Water Fluoridation Principles and Practices*, #30004
M5, *Water Utility Management*, #30005
M6, *Water Meters—Selection, Installation, Testing, and Maintenance*, #30006
M7, *Problem Organisms in Water: Identification and Treatment*, #30007
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M11, *Steel Pipe—A Guide for Design and Installation*, #30011
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M14, *Backflow Prevention and Cross-Connection Control: Recommended Practices*, #30014
M17, *Installation, Field Testing, and Maintenance of Fire Hydrants*, #30017
M19, *Emergency Planning for Water Utilities*, #30019
M20, *Water Chlorination/Chloramination Practices and Principles*, #30020
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M24, *Planning for the Distribution of Reclaimed Water*, #30024
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