

M49

Quarter-Turn Valves:

Head Loss, Torque, and Cavitation Analysis

Third Edition



American Water Works
Association



Documents
Document Sharing Hub

M49

Quarter-Turn Valves: Head Loss, Torque, and Cavitation Analysis

Third Edition

Incorporates errata dated May 1, 2017



**American Water Works
Association**

AVA Documents
Document Sharing Hub

Manual of Water Supply Practices—M49, Third Edition

Quarter-Turn Valves: Torque, Head Loss, and Cavitation Analysis

Copyright © 2001, 2012, 2017, American Water Works Association

All rights reserved. No part of this publication may be reproduced or transmitted in any form or by any means, electronic or mechanical, including photocopy, recording, or any information or retrieval system, except in the form of brief excerpts or quotations for review purposes, without the written permission of the publisher.

Disclaimer

The authors, contributors, editors, and publisher do not assume responsibility for the validity of the content or any consequences of its use. In no event will AWWA be liable for direct, indirect, special, incidental, or consequential damages arising out of the use of information presented in this book. In particular, AWWA will not be responsible for any costs, including, but not limited to, those incurred as a result of lost revenue. In no event shall AWWA's liability exceed the amount paid for the purchase of this book.

Project Manager: Melissa Valentine

Production: Studio Text

Cover Design: Melanie Yamamoto

Manuals Specialist: Sue Bach

Library of Congress Cataloging-in-Publication Data

Names: Bosserman, Bayard E., II, author. | Holstrom, John R., author. | American Water Works Association, editor.

Title: M49, quarter-turn valves : head loss, torque, and cavitation analysis / by B.E. Bosserman and John R. Holstrom.

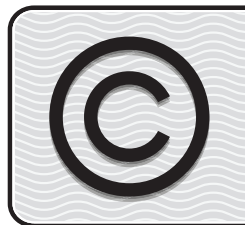
Other titles: Quarter-turn valves

Description: Third edition. | Denver, CO : American Water Works Association, [2017] | Series: Manual M49 | Includes bibliographical references and index.

Identifiers: LCCN 2017001678 | ISBN 9781625762061

Subjects: LCSH: Water-pipes--Valves.

Classification: LCC TD491 .B67 2017 | DDC 628.1/5--dc23 LC record available at <https://lccn.loc.gov/2017001678>



This AWWA content is the product of thousands of hours of work by your fellow water professionals. Revenue from the sales of this AWWA material supports ongoing product development. Unauthorized distribution, either electronic or photocopied, is illegal and hinders AWWA's mission to support the water community.

ISBN-13 978-1-62576-206-1

eISBN-13 978-1-61300-406-7

Printed in the United States of America
American Water Works Association
6666 West Quincy Avenue
Denver, CO 80235-3098
awwa.org



Contents



List of Figures, v

List of Tables, vii

Preface, ix

Acknowledgments, xi

Chapter 1	Introduction	1
	Scope, 2	
	New Definitions, MRST and AST, 2	
	Diameter Assumptions, 3	
	Quarter-Turn Valve Design, 4	
	System Conditions, 9	
	Definitions, 11	
	References, 13	
Chapter 2	Valve Head Loss and Equivalent Resistance System Model.....	15
	Discussion of Head Loss, Choking, and Cavitation, 15	
	Definitions, 16	
	Head Loss Calculations, 20	
	Reducer Installations, 23	
	Inherent and Installed Control Valve Flow Characteristics, 25	
	Equivalent Resistance System Model, 26	
	Variable Head Source Methodology, 31	
	Energy Calculations, 32	
	References, 35	
Chapter 3	Valve Torque	37
	Discussion of Torque Calculations, 37	
	Definitions, 39	
	Combining Torque Components, 42	
	Seating and Unseating Torque, 46	
	Packing and Hub Torque, 50	
	Bearing Torque, 51	
	Center of Gravity Torque, 55	
	Hydrostatic Torque, 56	
	Dynamic Torque, 58	
	Shaft Offset or Eccentricity Torque, 60	
	Other Components of Torque, 65	
	System Characteristics, 72	
	References, 76	
Chapter 4	Valve Cavitation	77
	Definitions, 77	
	Predicting Cavitation, 79	
	Cavitation Calculation Methodology Example, 82	
	Methods of Reducing Cavitation, 84	
	References, 84	

Chapter 5	Valve Testing.....	85
	Uncertainty, 85	
	Testing Requirements, 86	
	Test Procedure, 87	
	Seating/Unseating Test Procedure, 93	
	References, 95	
Chapter 6	Valve Applications.....	97
	Actuator Sizing, 97	
	Extended Bonnet Installation, 100	
	Effects of Pipe Installations, 101	
	Typical Range of Some Coefficients, 111	
	Cautions, 113	
	Summary, 114	
	References, 115	
	Appendix A, 117	
	Appendix B, 119	
	Appendix C, 121	
	Index, 129	
	List of Manuals, 133	

Figures

- 1-1 Valve disc, port, and pipe diameters, 4
- 1-2 Typical ball valve construction, 6
- 1-3 Typical butterfly valve construction, 7
- 1-4 Typical plug valve construction, 8
- 1-5 Typical rotary cone valve construction, 9
- 1-6 Butterfly valve offset designs, 10
- 1-7 Free discharge and reservoir inlet installations of BFVs, 11

- 2-1 Typical quarter-turn valve flow, differential pressure, cavitation, and choking graphical explanation, 17
- 2-2 Smaller-than-line-sized butterfly valve installation, 22
- 2-3 Reducer geometry, 23
- 2-4 Typical inherent valve characteristics, 26
- 2-5 Installed valve characteristics, 26
- 2-6 Equivalent resistance system model, 27
- 2-7 Example equivalent resistance system model diagram, 30
- 2-8 Graphical relationship between head loss and velocity, 31
- 2-9 Comparative operating costs by valve type, 34

- 3-1 Basic closure member design geometries, 40
- 3-2 Horizontal valve shaft in a horizontal pipe, 41
- 3-3 Seat-side, reverse, and shaft-side direct flow direction orientations, 42
- 3-4 Active torque sign convention; positive values tend to close the valve, 42
- 3-5 Dynamic torque (T_d) and bearing torque (T_b) during valve closure, 45
- 3-6 One-, two-, and three-pump system operating curves, 45
- 3-7 Multiple-pump installation; example shown with butterfly valves, 46
- 3-8 Butterfly valve seating torque, 46
- 3-9 Plug valve seat perimeter and port area, 48
- 3-10 Butterfly valve packing and hub seal torque (T_p), 50
- 3-11 Bearing friction torque (T_b), 52
- 3-12 Center of gravity torque (T_{cg}), 55
- 3-13 Hydrostatic torque (T_h), 57
- 3-14 Dynamic torque (T_d) for a symmetrical disc butterfly valve, 59
- 3-15 Dynamic torque coefficient (C_t) graph for butterfly valves with symmetrical and offset closure members, 60
- 3-16 Dynamic torque (T_d) for a butterfly valve with symmetric and offset discs, 61
- 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), 61

- 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), 62
- 3-19 Total closing torque (T_{tc}) for a 20-in. to 30-in. butterfly valve with symmetric and offset discs, 62
- 3-20 Total closing torque (T_{tc}) for a 78-in. to 96-in. butterfly valve with symmetric and offset discs, 63
- 3-21 Shaft offset or eccentricity torque (T_{ecc}), 63
- 3-22 Bearing torque caused by closure member and shaft(s) weight orientation angles, 66
- 3-23 Center of gravity torque pipe angle definition, 68
- 3-24 Valve shaft and pipe orientation from vertical axis for center of gravity torque, 68
- 3-25 Valve shaft and pipe orientation from vertical axis for hydrostatic and bearing torque, 70
- 3-26 Example torque calculation summary graph, 75

- 4-1 Cavitation zones downstream of a butterfly valve disc, 78
- 4-2 Typical cavitation index levels and acceleration readings, 80
- 4-3 Flow rate and acceleration readings, 80
- 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), 81
- 4-5 Example cavitation analysis summary graph, 83

- 5-1 Basic flow test system, 87
- 5-2 Butterfly valve test installation, 88

- 6-1 Typical actuator torque characteristics, 99
- 6-2 Typical ball valve construction, 99
- 6-3 Scotch yoke traveling nut actuator cutaway, 99
- 6-4 Link-and-lever traveling nut actuator cutaway, 100
- 6-5 Actuator sizing characteristics graph, 101
- 6-6 Typical extended bonnet construction, 102
- 6-7 Vertical elbow upstream of a butterfly or other quarter-turn valve preferred installation orientation, 104
- 6-8 Upstream expansion orientation preferences, 105
- 6-9 Upstream orifice orientation preferences, 106
- 6-10 Upstream reducer orientation preferences, 107
- 6-11 Upstream check valve preferred installation orientation, 107
- 6-12 Series-mounted valve preferred installation orientation, 108
- 6-13 Combined improper orientations, 108
- 6-14 Plug valve preferred installation orientation for suspended solids applications, 111

- B-1 Seating torque free body diagram, 119

Tables



2-1 Reducer and Increaser Calculations, 25
2-2 Valve Reducer, and Increaser Assembly Adjusted Flow Coefficients, 25
2-3 Head Loss Calculation Data for Constant Head Source Example, 30
2-4 Annual Pumping Cost Example Calculation, 33
3-1 Torque Component Category, 38
3-2 Torque Calculation Data for Constant Head Source Example, 75
4-1 Cavitation Calculation Data for Constant Head Source Example, 83
6-1 Typical Bearing Friction Coefficients, 112
6-2 Typical Packing Coefficients, 112
6-3 Typical Seating Coefficients, 112
6-4 Typical Full-Open Flow Coefficients, 113
C-1 Nomenclature, terms, and symbols, 121
C-2 Conversion of units, 128
C-3 Abbreviations, 128

This page intentionally blank.

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.

This page intentionally blank.

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 ($AST = MRST \times AF$). 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.

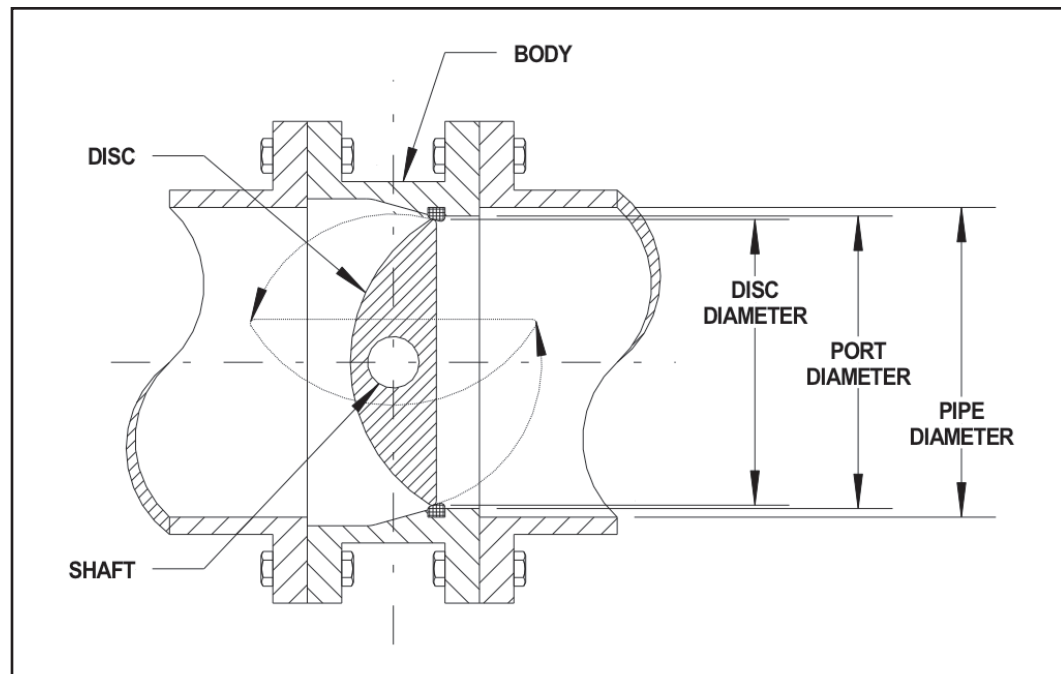


Figure 1-1 Valve disc, port, and pipe diameters

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.

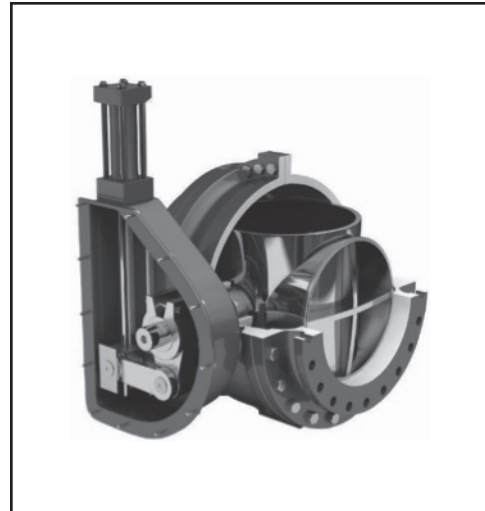
AWWA Butterfly Valve Design

AWWA BFs 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

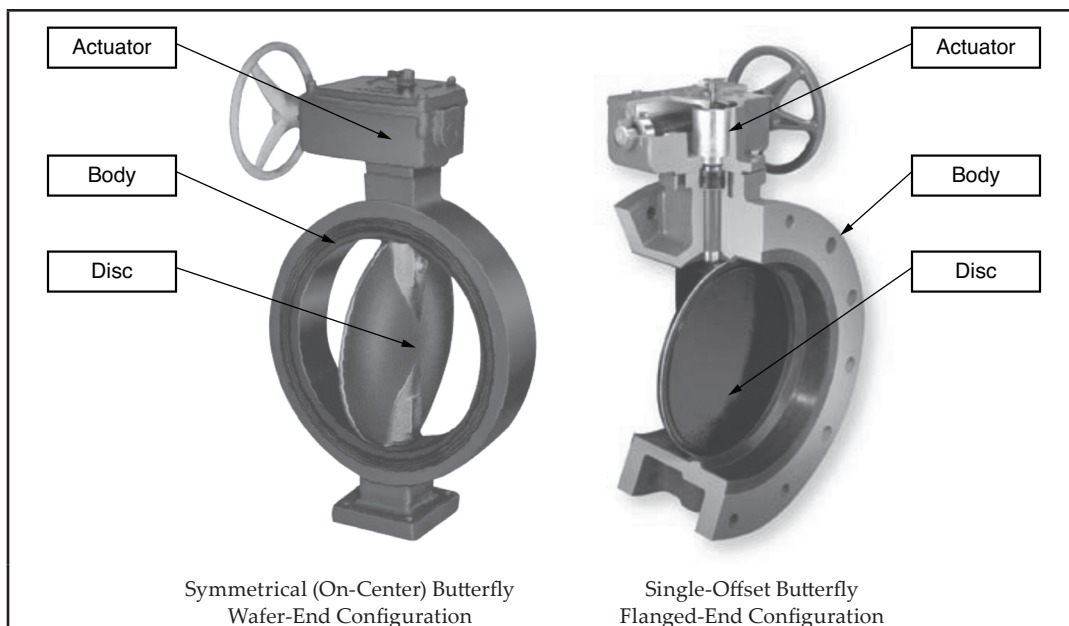
Figure 1-2 Typical ball valve construction

- 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

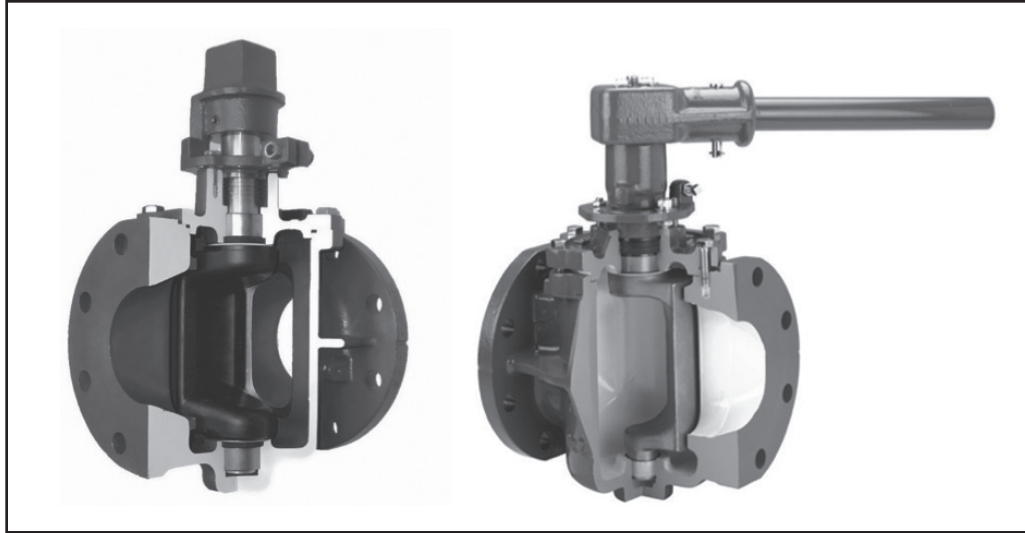
Figure 1-3 Typical butterfly valve construction

- 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 through 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

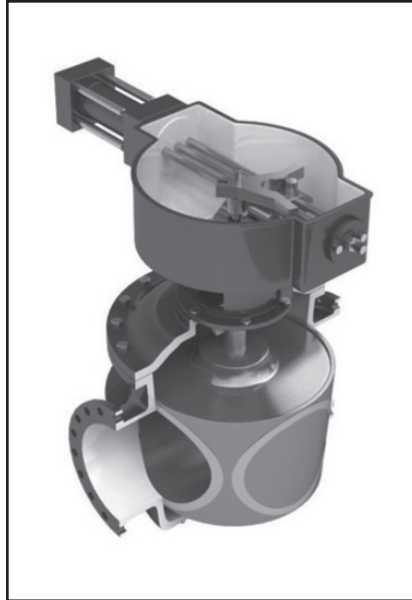
BFVs are available in several different offset designs. The discs, shafts, and seats may be aligned in various relative locations. The popular arrangements are the symmetric (on-center) design and the single-offset, double-offset, or triple-offset design.

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.

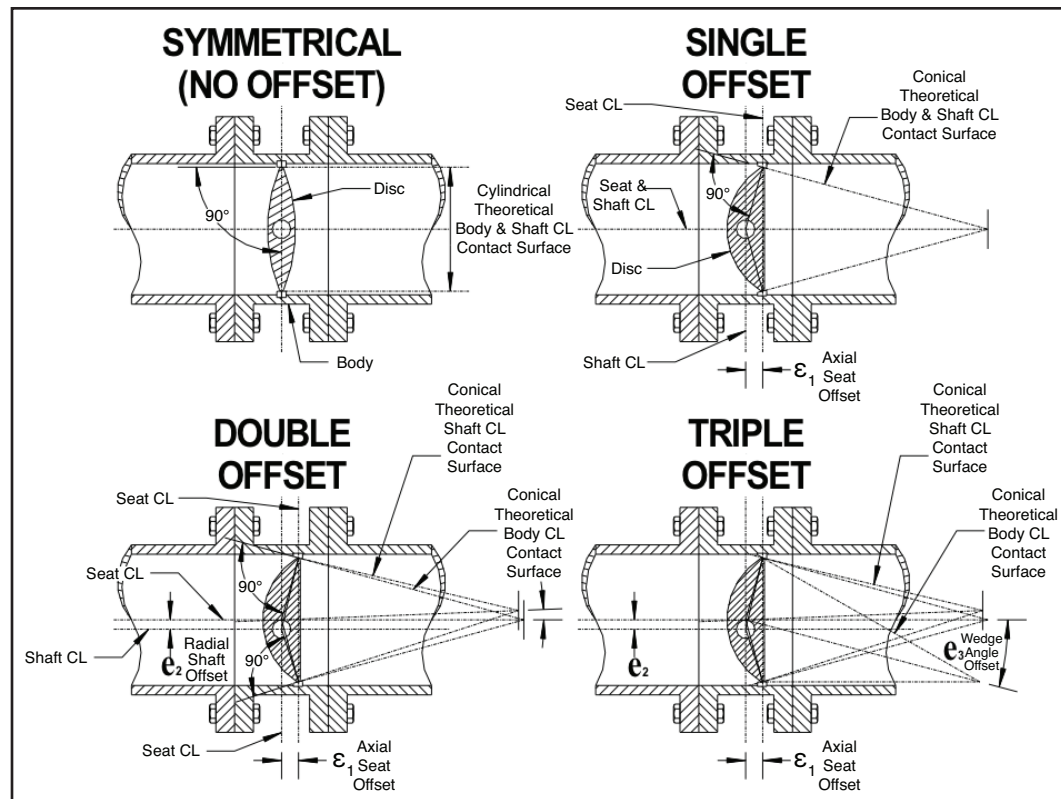


Figure 1-6 Butterfly valve offset designs

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.

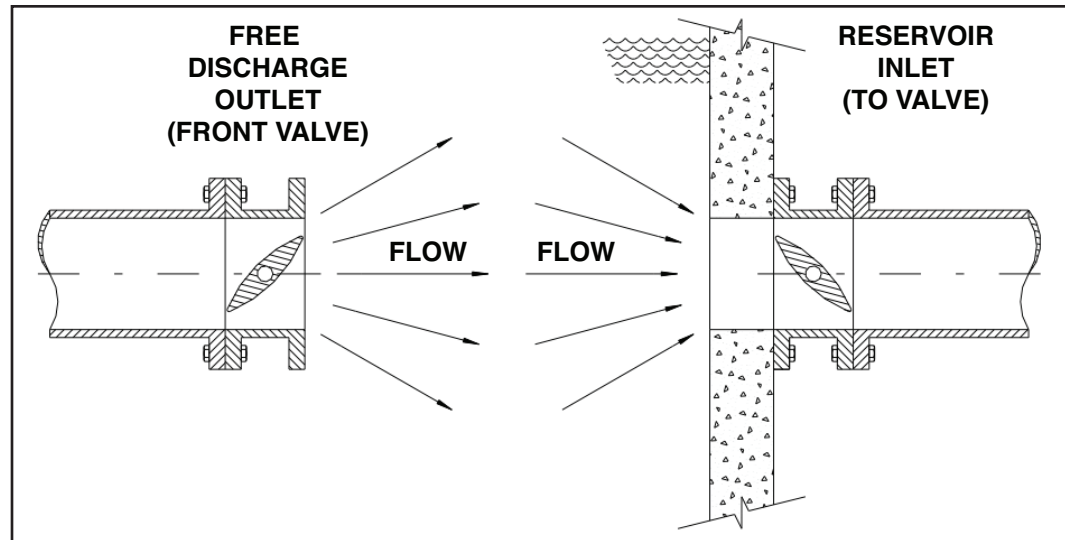


Figure 1-7 Free discharge and reservoir inlet installations of BFVs

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 ($AST = MRST \times AF$). 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 through 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 BFFVs, 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 BFFVs, 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 through 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.

REFERENCES

- American Water Works Association. 2010. ANSI/AWWA C517-16: Standard for Cast-Iron Eccentric Plug Valves. Denver, CO: AWWA.
- American Water Works Association. 2015. ANSI/AWWA C504-15: Standard for Rubber-Seated Butterfly Valves. Denver, CO: AWWA.
- American Water Works Association. 2015. ANSI/AWWA C507-15: Standard for Rubber-Seated Ball Valves. 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.
- Hutchinson, J.W., ed. 1976. *ISA Handbook of Control Valves*, 2nd ed. Research Triangle Park, NC: Instrument Society of America (now known as the International Society of Automation).
- International Society of Automation. 2008. *Control Valve Capacity Test Procedure*, ANSI/ISA S75.02.01-2008.
- International Society of Automation. 2012. *Industrial-Process Control Valves. Part 2-1: Flow capacity: Sizing equations for fluid flow under installed conditions*, ANSI/ISA-S75.01.01-2012.

This page intentionally blank.

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

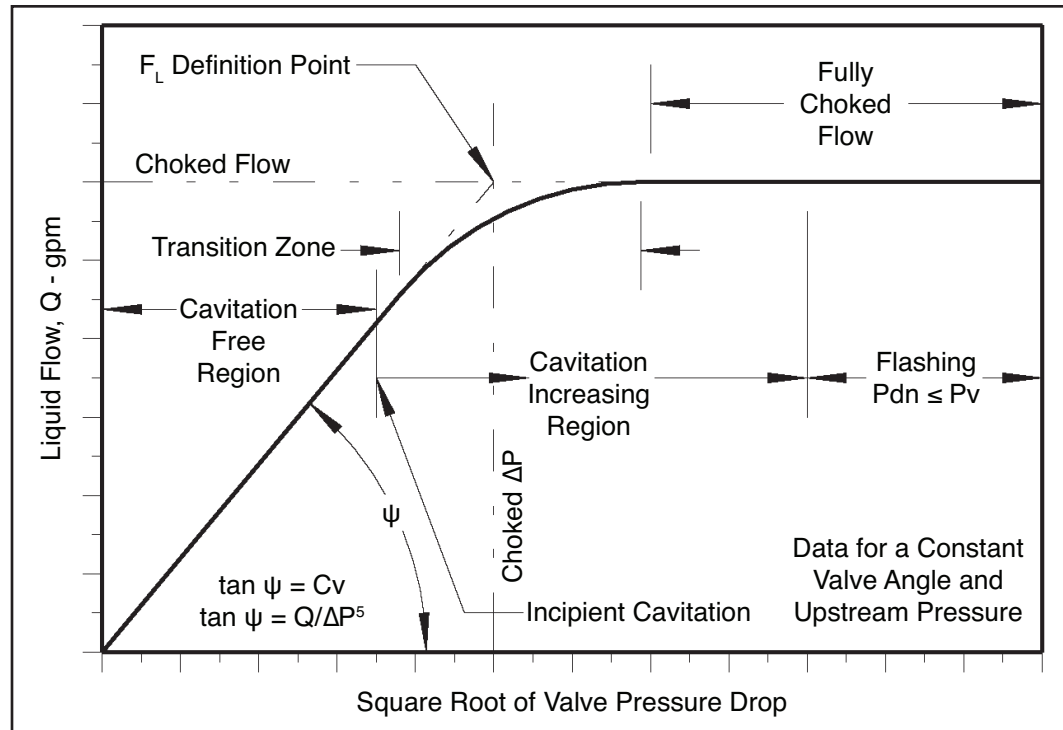


Figure 2-1 Typical quarter-turn valve flow, differential pressure, cavitation, and choking graphical explanation

to satisfy many types of specific flow systems and conditions. There are several variations of flow coefficients, such as K_v , C_v , C_t , 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:

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

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

$$C_{vm} = C_v \times 0.86 \quad (2-3)$$

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 m³/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):

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

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

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

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

$$\text{(in SI metric units)} \quad K = \frac{8.52 \times (D \times U_{c1})^4}{C_{vm}^2} \quad (2-6)$$

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

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

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

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

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 .	gpm/psi ^{1/2} None
	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} .	

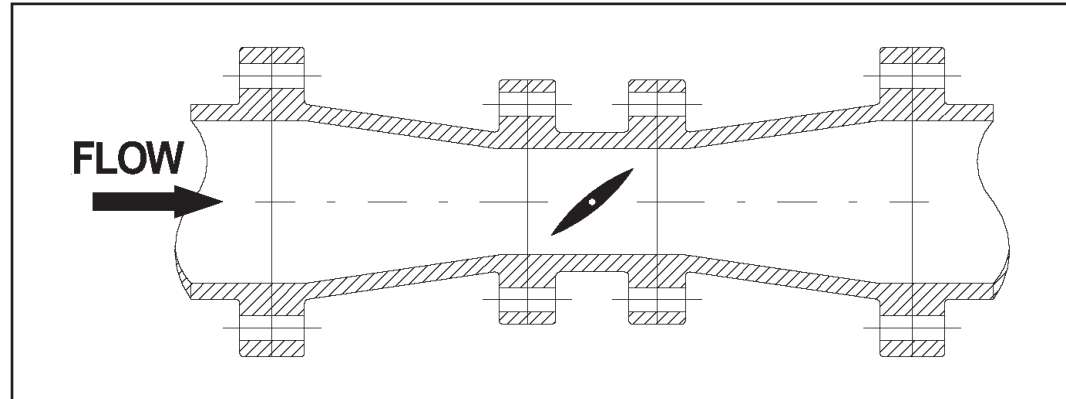


Figure 2-2 Smaller-than-line-sized butterfly valve installation

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 ($m^3/hr/bar^{1/2}$) ($m^3/hr/(100\text{ kPa})^{1/2}$)
D	Nominal valve diameter	in. (mm)
D_{PIPEID}	Pipe ID	in. (mm)
g	Gravitational constant Acceleration due to gravity, 32.2 ft/s ² (9.81 m/s ²)	ft/s ² (m/s ²)
K	Flow resistance coefficient of any component or fitting	dimensionless
Q	Volumetric flow rate	gpm (m^3/hr)
U_{CI}	Units conversion factor: US customary for torque in in.-lb: $U_{CI} = 1$ in./in. US customary for torque in ft-lb: $U_{CI} = 1/12$ (0.0833) ft/in. Metric for torque in N-m: $U_{CI} = 1 \times 10^{-3}$ (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.

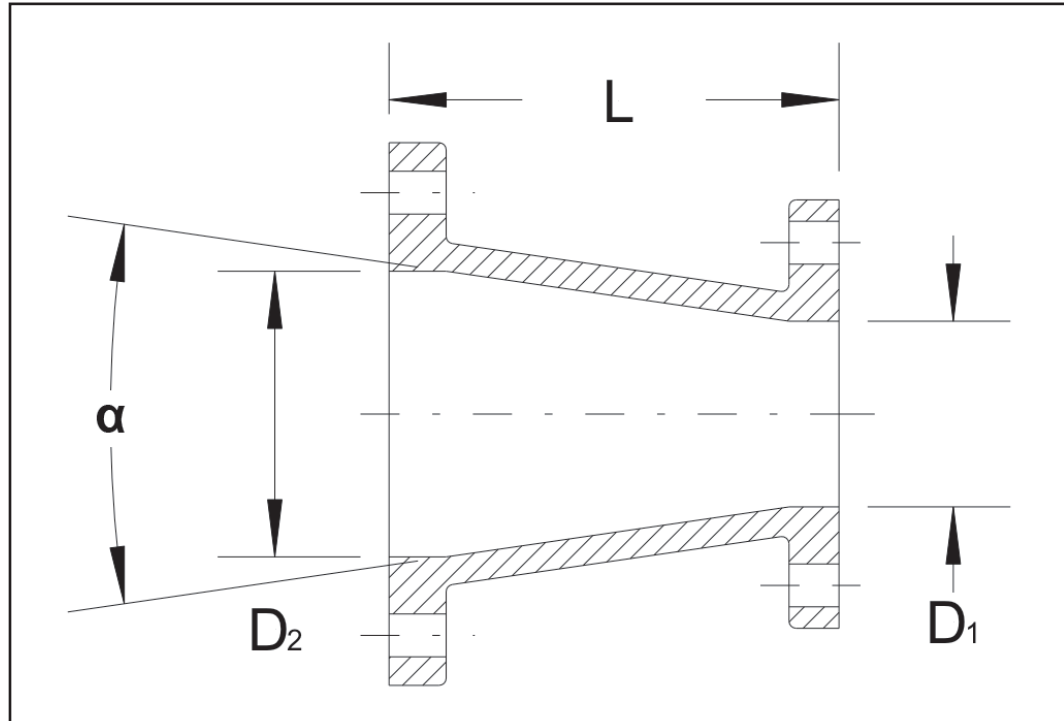


Figure 2-3 Reducer geometry

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_{val} 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):

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

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

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

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

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

$$\beta_r = \frac{D_1}{D_2} \quad (2-12)$$

$$K_2 = \frac{K_1}{\beta_r^4} \quad (2-13)$$

$$K_{vd2} = \frac{K_{vd1}}{\beta_r^4} \quad (2-14)$$

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 $\leq 45^\circ$ ($\pi/4$ 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 _{RA10} " Valve Diameter Basis	Calculated Reducer Assembly Flow Resistance "K _{RA20} " Pipe Diameter Basis	Calculated Reducer Assembly Flow Coefficient "C _{Vassy0} " 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.

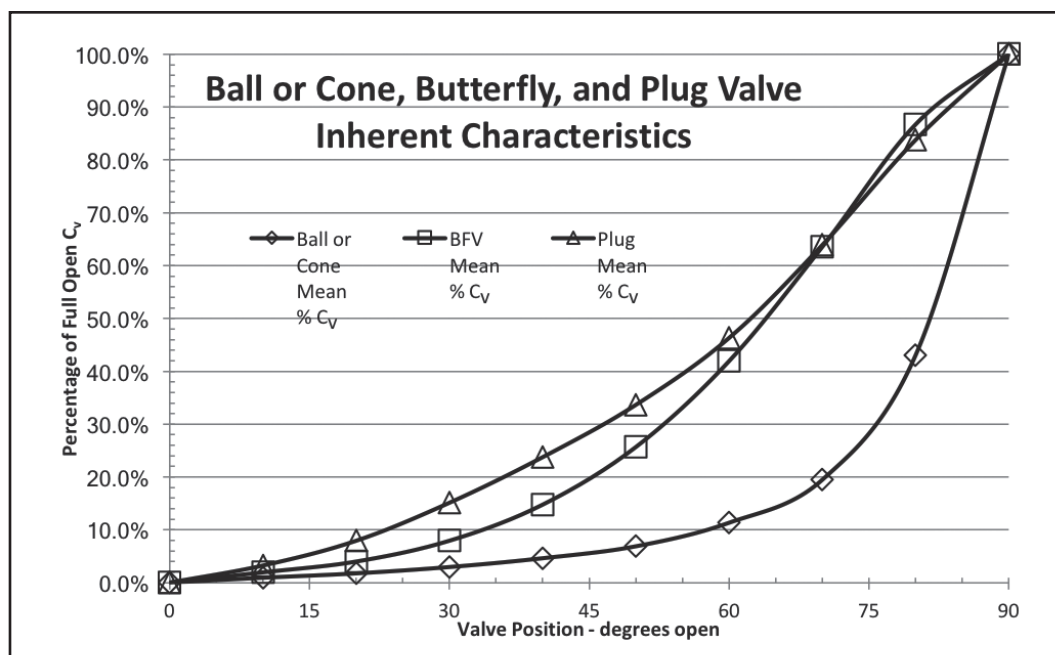


Figure 2-4 Typical inherent valve characteristics

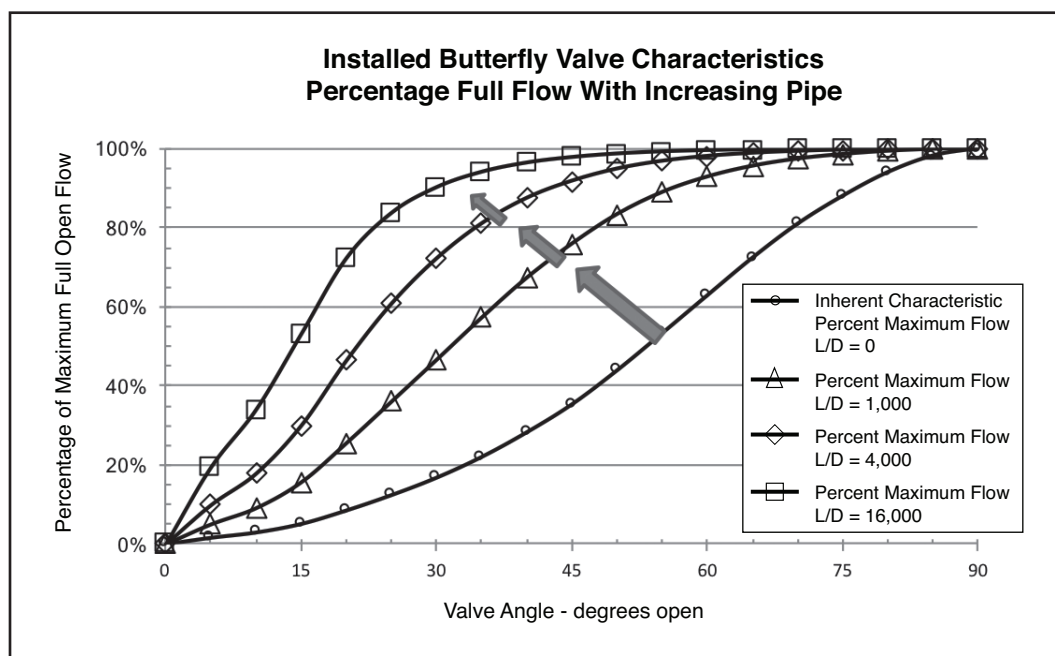


Figure 2-5 Installed valve characteristics

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

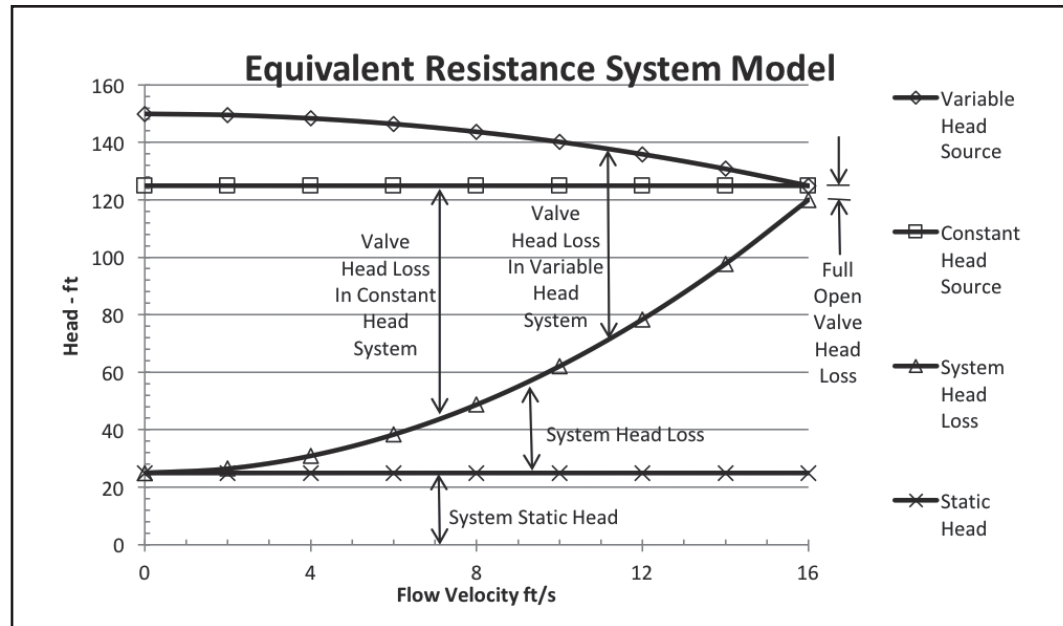


Figure 2-6 Equivalent resistance system model

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:

$$K_{\text{sys}} = \frac{2 \times g \times \Delta H_{\text{MAX}}}{V_{\text{MAX}}^2} - K_{V90} \quad (2-15)$$

where

Variable	Definition or Description	Units US Customary (SI metric)
g	Gravitational constant Acceleration due to gravity, 32.2 ft/s ² (9.81 m/s ²)	ft/s ² (m/s ²)
K_{sys}	System flow resistance coefficient (excluding the valve)	dimensionless
K_{V90}	Flow resistance coefficient of valve at full open ($\approx 90^\circ$, $\approx \pi/2$ 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

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

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):

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

or

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

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):

$$\Delta P_{V\theta} = 0.4335 \times \Delta H_{V\theta} \quad (2-19)$$

or

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

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):

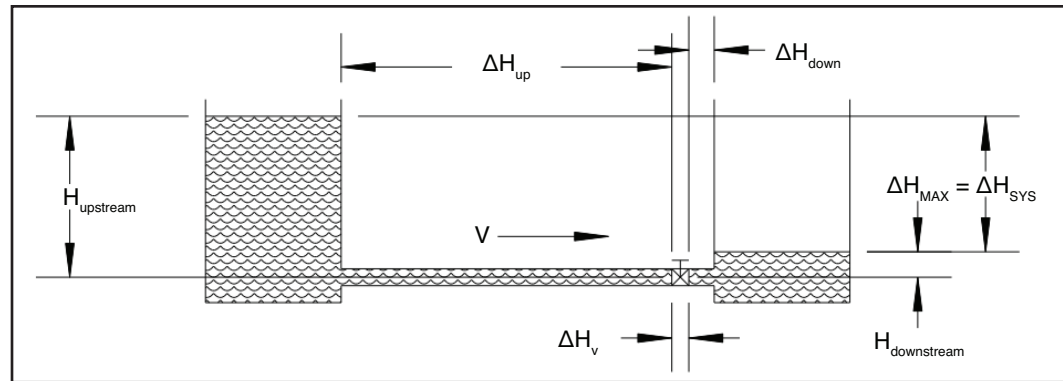


Figure 2-7 Example equivalent resistance system model diagram.

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 ($\Delta P_{MAX} = 43.4$ psid, Eq 2-19)

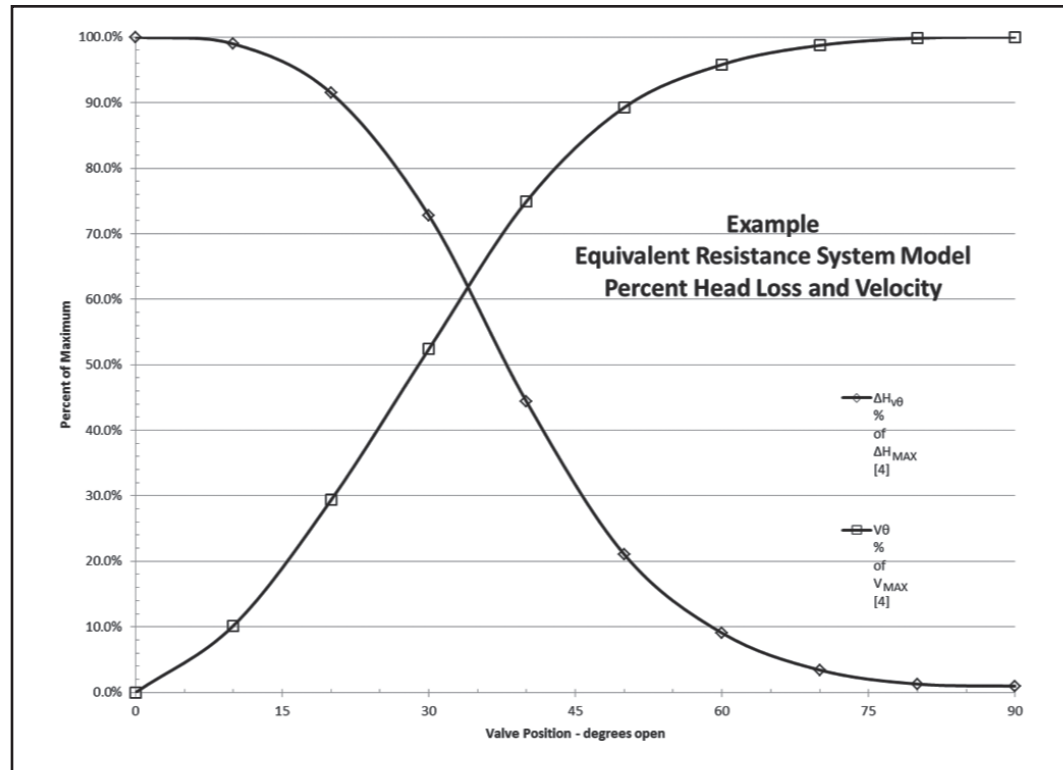


Figure 2-8 Graphical relationship between head loss and velocity

- Closed upstream head (H_{upstream}) is 200 feet of water ($P_{\text{upstream}} = 86.7$ psi, Eq 2-19)
- System downstream static head (H_{dnstream}) is $H_{\text{upstream}} - \Delta H_{\text{MAX}} = 200 - 100 = 100$ feet of water ($P_{\text{dnstream}} = 43.4$ psi, Eq 2-19)
- Maximum system flow rate, $Q_{\text{MAX}} = 20,000$ gpm ($V_{\text{MAX}} = 14.2$ 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
- $K_{\text{sys}} = \frac{2 \times 32.17 \times 100}{16^2} - 0.3 = 31.61$ from Eq 2-15
- Percentage of system upstream of valve = 75% ($K_{\text{sys-up}} = 23.70$)
- 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