FLOWSERVE

Edward Valves

EV100
5th Edition

Technical G

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Equations and calculations outlined in this manual are available in a proprietary Edward Valves computer program. Consult your Edward Valves sales representative for more information.

Figure and Table Number Index

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1. Stop and Check Valve Applications Guide

1.1 Stop Valve Applications

Foreword

Edward stop valves are used primarily as isolation valves in medium and high pressure piping systems. They are offered in a broad range of sizes, pressure ratings, and types, and they are used in an immense array of diverse applications. Only a few are listed for illustration:

- Normally open valves in main steam lines; used only for equipment isolation, e.g. during maintenance.

- Normally open valves to provide for emergency shutoff due to failure of downstream piping or other equipment; closed periodically for verification of operability.

- Normally open valves that are throttled to varying degrees during start-up or shutdown of plants or systems.

- Frequently cycled valves that are opened and closed for control of batch processes or for start-up and shutdown of equipment (e.g., equipment that is on-stream daily but shut-down at night).

- Normally closed valves; used only for filling or draining systems during outages.

Stop valves are sometimes referred to as “on-off valves.” They should not normally be considered as “control valves,” but they are suitable for moderate or infrequent flow-control functions. Valves that must open and close under high differential pressure and flow conditions (such as “blowdown” service) inherently function as flow-control devices while they are stroking.

Considering the diversity of stop valve applications, it is not surprising that there is no universal valve type that is best for all services. Users’ experience with specific applications is a valuable basis for selecting the best valves.

The goal of this guide is to supplement users’ experience with information based on decades of Edward Valves’ laboratory tests and field experience.

Introduction

While many other types of valves (ball, plug, butterfly) are used as stop valves where service conditions permit, emphasis in this guide is on selection and application of Edward valves with forged- and cast-steel bodies and bonnets. Comparisons are presented with other similar valves where appropriate.

Edward stop valves are typically of metal-seated construction and, where necessary, use gaskets and stem seals designed for severe high-pressure, high-temperature service. While special designs with “soft seats” and O-ring seals are supplied for unique specific applications, the standard products are designed to stand up to tough service conditions with minimum requirements for maintenance or parts replacement.

Edward stop valves fall into two basic categories – **globe valves** and **gate valves**. The following sections of this guide will address the principal features of each type and the design variations within the types.

Globe valves are offered in stop, stop-check, and check versions. Stop-check valves can also be used for isolation in unidirectional flow applications. These valves are discussed in the Check Valves Applications section (1.2).

The FLOW PERFORMANCE section of this catalog provides equations and coefficients for the calculation of pressure drop across any of these valves. This information can be used to evaluate the effects of different valve sizes and types on system energy efficiency.

1.1.1 Stop Valve Types and Typical Uses

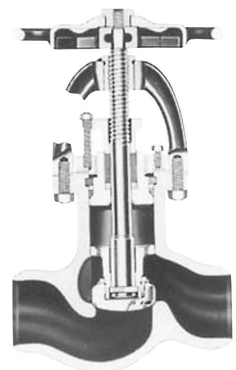
Brief notes on the advantages, disadvantages, applications and limitations of the various types of Edward stop valves are presented in the Stop Valve Applications Chart (section 1.1.4). Some additional highlights of the features of these valves and some comparisons with similar valves are presented in the following paragraphs.

Globe Valves

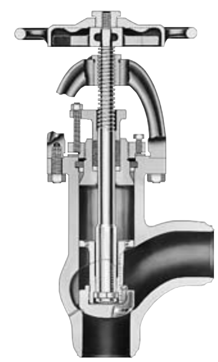
A globe valve employs a poppet or disk that opens and closes by moving linearly along the seat axis. There are many types of globe valve bodies, seats and methods of guiding the disk to and from the seat.

- **Bodies** –Edward stop, stop-check and check type globe valves are offered with three basic body styles:

Conventional or 90°-bonnet globe valves are usually the most compact, and the stem and yoke position allow easy handwheel or actuator access and convenience for maintenance. Relatively short stem travel allows fast actuation. Multiple direction changes in the flow stream result in higher pressure drop than with other types, but streamlined flow passages in Edward valves generally yield lower pressure drop than competitive valves of this type.



Angle valves are otherwise similar to conventional globe valves, but the less tortuous flow path yields lower pressure drop. Angle valves are particularly economical in piping layouts where use of this configuration eliminates an elbow and associated flanged or welded joints.



Inclined bonnet or “Y type” valves, such as Univalves® and Flite-Flow® valves, yield lower pressure drop than other styles, because they permit a more nearly straight-through flow path. Typically, they require a longer stem travel. In large sizes, this body shape is heavier and requires a greater end-to-end length than conventional globe valves.

1.1 Stop Valve Applications Guide (con't.)

• **Seats** – Industrial globe valves are available from various manufacturers with a broad variety of seat designs — flat or tapered, and integral or inserted (threaded or welded).

All Edward globe valves employ tapered seats with "area contact" under load to seal over minor imperfections. Many similar valves use "line-contact" seats that seal with less load when new but degrade rapidly if damaged at the seating line.

Except for hydraulic stop valves, all Edward globe valves employ integral (hardfaced) body seats to permit compact design and assure that there can be no leakage "behind" the seat.

• **Disk Guiding** – Globe valve disks may be guided by either the stem or the body. When opened or closed under very high differential pressure, side load due to flow pushes a stem-guided disk eccentric to the seat and makes it difficult to obtain a seal. Under extreme conditions, the stem may bend.

All Edward globe valves employ body guided disks which are held closely concentric with the body seat. Guiding is provided at both the top and bottom of the disk to form a fully body-guided disk piston. The bottom guide ring on the disk, and Edward innovation, minimizes flow behind the disk and minimizes the side load. These features make Edward globe valves well suited to "blowdown" applications in which there is a high differential pressure across the valve when it is partially open.

Since globe valves are not symmetrical with respect to flow, consideration must be given to the direction of flow and differential pressure. It should be noted that the direction of flow when open and differential pressure when closed may not be the same in all applications (e.g., a block valve on a feed line may involve flow into a system when open but may need to prevent leakage out of the system when closed). Users should consider both factors when deciding on the installation direction for a globe valve.

In most globe valve applications, pressure is under the seat when the valve is closed, and the flow is from under to over the seat (termed "flow to open" or "underseat flow"). In installations where the downstream pressure is zero or very low, this arrangement minimizes packing leakage problems. However, handwheel or actuator effort to close the valve is high, because the stem must supply enough load to both overcome the differential pressure load across the seat area and ensure sufficient sealing load on the metal seat-contact surfaces. Since this flow direction is the most common for globe valves, the flow coefficients given in the Flow Performance section of this catalog are for underseat flow.

Globe valves can also be used with overseat flow and pressure ("flow to close"), but such applications require careful consideration. In systems with dirty line fluids, this arrangement could lead to trapping foreign material in locations where it would interfere with opening. With overseat pressure, the effort to close the valve is low, because closure and sealing are pressure-assisted. However, the effort to open the valve at high differential pressure is high, because the stem must overcome the pressure force to lift the disk (in small valves, the stem diameters approaching the seat diameter, this may not be a problem, because the pressure helps to lift the stem). Also, since the flow coefficients given in this catalog are for underseat flow, pressure-drop predictions may not be as accurate (pressure drop may be up to 10% higher with overseat flow).

While not designed as control valves and not recommended for continuous modulation, Edward globe valves are often used successfully for manual or automatic control during limited periods of system operation (start-up, shutdown, etc.). Some manual valves are also used for continuous throttling or "trimming." Inclined-bonnet valves, (e.g., Univalves® and Flite-Flow® valves) have an approximately linear flow characteristic (C_V versus % open).

The Flow Performance section of this catalog covers only flow coefficients for fully open valves, but consult Edward Valves concerning applications involving flow control. It should be understood that severe throttling at high pressure drops involves high energy dissipation, and serious problems (e.g., noise, vibration, cavitation, erosion) can develop if not carefully considered when a system is designed.

Gate Valves

A gate valve employs a closure member (or assembly) that opens and closes by moving perpendicular to the flow stream to engage two seats in the body. There are two basically different types of gate valves – parallel-side and wedge gate – in common use in pressure-piping systems, but there are many variations in design within each type.

As compared to glove valves, all gate valves offer straight-through flow paths which tend to produce less pressure drop than typical globe valves of the same nominal size. A Venturi gate valve with a smaller port than a Regular gate valve may offer a lower first cost as well as a size and weight saving if a minimized pressure drop is not required.

The Flow Performance section of this catalog gives comparable flow coefficients for Edward Equiwedge® gate valves and all Edward globe stop valves. Evaluation of many valve applications has shown that inclined-bonnet globe valves are often competitive with gate valves when all factors are considered.

The stem in a gate valve does not have to overcome the full differential pressure load across the valve seat area to open or close the valve. Instead, it just has to overcome the friction force due to that load. Consequently, for operation at similar differential pressures, a gate valve generally requires less effort for actuation than a globe valve and can employ a smaller actuator when powered operation is required. However, a gate valve requires considerably greater stem travel than a conventional globe or angle valve (slightly greater than an inclined-bonnet globe valve), so a somewhat longer time may be required for action.

The two body seats – the common feature in all ordinary gate valves – can be both an advantage and a disadvantage. Most gate valves are primarily "downstream-sealing," because the closure member is pressure-energized in that direction. However, the upstream seating surfaces may help by limiting leakage if the downstream seat is damaged. Simultaneous sealing at both seats can be hazardous if the center cavity of a closed valve is filled or partially filled with liquid and then subjected to an increase in temperature, causing a corresponding increase in pressure. In moderate cases, this may cause "pressure binding" which can impede or prevent valve opening; in extreme cases, it may cause pressure-boundary failure (e.g., the bonnet could blow off).

Note: *ASME/ANSI B16.34-1988 (paragraph 2.3.3) places the responsibility of the purchaser to assure that the pressure in the valve will not exceed that allowed by the standard.Special operating procedures, such as partially opening a valve during warm-up, may be considered. Special internal design features or external bypass arrangements are required in many applications. Consult Edward Valves regarding Edward Equiwedge® gate valve applications that may be subject to possible center-cavity over-pressurization.*

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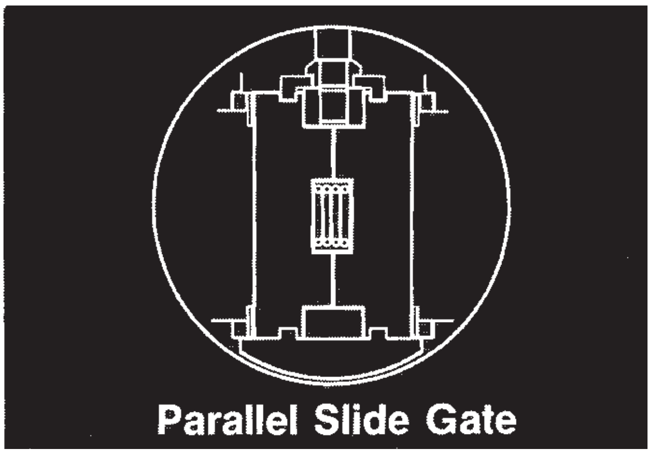
1.1 Stop Valve Applications Guide (con't.)

Some highlights of the various types of gate valves, including the Edward Equiwedge, are discussed below:

• Parallel-Slide Gate Valves
Edward does not offer parallel-slide valves. In these valves, the two seats in the body are in parallel planes, and an assembly including two gates with parallel seating faces moves into or out of engagement with the body seats. The gates are urged into contact with the opposing seats in the closed position by either a spring (or a set of springs) or an internal wedge mechanism.



Since the two gates are relatively independent, the downstream gate is free to align with the downstream seat, and new valves usually seal well so long as the differential pressure across the valve is sufficient to provide adequate seating load. Leakage may be a problem with these valves at low differential pressures (e.g. when filling a system or during low-pressure start-up operation).

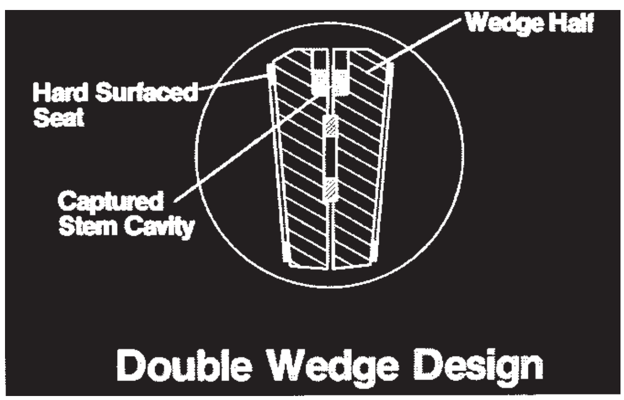
In typical parallel-slide valves, there is continuous sliding contact between the sealing surfaces of the gates and body seats throughout the full stem stroke. Wearing or scoring is possible, particularly when operating with high differential pressures, and this may cause seat sealing to be degraded. This shearing action may be helpful in cleaning loose debris from the seats, however.

• Wedge Gate Valves
A wedge gate valve uses one of the oldest engineering principles to provide mechanical advantage to convert stem load to seat-sealing load. This is particularly important in low-pressure applications where differential pressure alone may not provide sufficient loading on the downstream seat.

Early wedge gate valves for low pressure employed solid wedges, and these are still used in many small high-pressure gate valves. However, as industrial valve requirements moved toward larger sizes and higher pressures and temperatures, a solid wedge designed to provide sufficient strength became too rigid to accommodate the flexibility of the valve body. The seat planes deflect significantly in large, high-pressure valve bodies due to thermal effects and the loads from connecting piping, and a rigid wedge may either leak or bind in the closed position.

Many gate valves have been designed with "flexible" one-piece wedges that have overcome these problems to some degree, but the two halves of the wedge are not truly independent and free to align with the two opposing body seats. It is particularly difficult to provide torsional flexibility in the wedge to accommodate twist in the valve body.

Consequently, the Edward Equiwedge valve was designed with two independent, flexible wedge halves that permit relative rotation and can tilt to accommodate changes in the body-seat angles. The thickness of the wedges was minimized, while maintaining acceptable stresses, to allow deflection to accommodate out-of-flatness in the seat plane. In prototype tests, acceptable sealing was maintained with seats intentionally misaligned 1° in angle and up to 2° in rotation.



The result is a valve that has high-pressure sealing performance comparable to that of a parallel-slide valve but that can also seal exceptionally well at low differential pressures. The independent, flexible wedge halves in Edward Equiwedge gate valves also have commendable resistance to sticking or binding in the closed position. In prototype tests, the valve always opened with a torque less than the design closing torque when exposed to extreme pipe-bending moments and severe thermal transients (heat-up and cool-down).

All wedge gate valves have body guides that must support the wedges when they are not in the fully closed position. The seating surfaces of the wedges and seats are in sliding contact only through a small portion of the opening and closing travel, thus minimizing wear that may degrade seat sealing. Outside that range, the side loads are transferred from the seats to the body guides. Wear or scoring of the body guides does not affect sealing.

In Edward Equiwedge gate valves, the body guides are vertical machined grooves at each side of the valve body which engage tongues on each side of the wedge halves. Precision machining allows transfer of side load from the seats to the body guides within 3% to 5% of valve travel. Testing has proven that this guiding system is rugged and supports the gate assembly effectively, even in "blowdown" services where high differential pressure loads act across the gates when the valve is partially open.

Gate valves of any type are usually not recommended for throttling or modulating flow-control service. The seating surfaces of the gates are subject to impingement when partially open, and some gate valves reportedly exhibit instability (internal vibration) when throttled. Nevertheless, high-velocity flow tests of a prototype Edward Equiwedge gate valve produced no flow-induced vibration, and there are cases where these valves have been used successfully for limited flow-control functions. Consult Edward Valves concerning any proposed throttling or control applications.

continued

1.1 Stop Valve Applications Guide (con't.)

1.1.2 Throttling Characteristics of Edward Stop Valves

As noted in the previous section, Edward stop valves are not normally recommended for continuous modulation, and Edward Valves should be consulted concerning applications involving flow control. This section is intended only to provide general guidelines on flow-control characteristics of typical Edward stop valves. These guidelines may be used for preliminary studies relating to applications involving throttling, but they should not be considered as a substitute for a complete evaluation of the acceptability of a valve for a critical application.

Figure A
Inherent Flow Characteristics

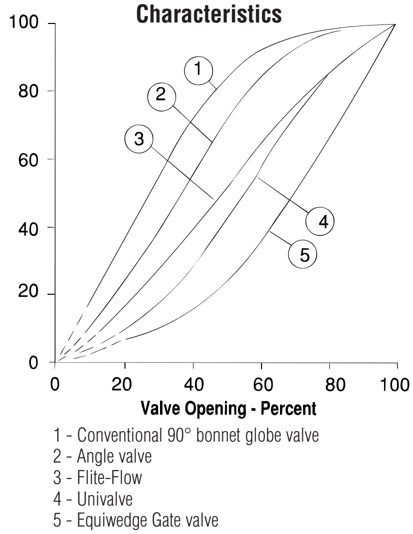


Figure A provides typical **inherent flow-characteristic** curves (percent of full-open flow coefficient versus percent opening) of the most common types of Edward stop valves. It should be understood that these curves are approximate, because there are variations due to size and pressure class that cannot be represented accurately by a single curve for each valve type. Nevertheless, these typical curves can provide some guidance relating to control capabilities of the various valve types.

Note the following subtle differences between the curves in Figure A:

• The conventional 90°-bonnet globe valve provides a relatively steep slope at small openings approaching a "quick-opening" characteristic. While the body-guided disk in Edward globe valves moderates this effect, it makes the flow coefficient very sensitive to small changes in stem position, so it may prove difficult to control low flow rates.

• The angle valve has a characteristic similar to that of a globe valve, but it is slightly closer to linear due to its normally higher full-open flow coefficient. An angle valve has about the same control characteristics as a globe valve of the same size at small openings.

• The cast-steel Flite-Flow® Y-type valve provides a characteristic that is nearly linear over most of its stem-travel range. For control of flow over a broad range, the high flow efficiency of this type of valve may permit use of a smaller valve size for a given allowable pressure drop. The smaller size, combined with the linear characteristic, can give improved control of low flow rates when the valve is throttled.

• The forged-steel Y-type Univalve® provides even better control at very small openings because of its "double throttling" characteristic as the lower disk-guide ring opens the machined port in the body. Other forged-steel valves have this characteristic to some degree.

• The Equiwedge, gate valve has an excellent inherent flow characteristic ("concave upward"), approaching that of an **equal-percentage** control valve. However, this is somewhat misleading. When installed in pipe of the same nominal size as the valve, the pressure drop of a gate valve is so low at large openings (e.g., over 70%) that piping flow resistance usually overshadows that of the valve. The gate valve would provide little control over flow in that range.

While not normally recommended for throttling for the reasons cited in the previous section, the gate valve flow-characteristic curve is attractive from a standpoint of controlling low flow rates without excessive sensitivity. Use of a gate valve for throttling may be considered for some applications.

1.1.3 Stop Valve Actuators and Accessories

Most Edward stop and stop-check valves illustrated in this catalog are shown with handwheels, and the majority of valves are furnished for applications where manual actuation is acceptable. Most larger and higher-pressure globe valves are furnished with standard Impactor handles or handwheels, which provide up to twelve times the stem force of an ordinary handwheel, to provide for adequate seating thrust. Impactogear assemblies on the largest globe valves permit operation using an air wrench. These Edward innovations permit practical manual operation of many valves that would otherwise require gearing or power actuators.

Manual Gear Actuators

When specified, many Edward valves can be supplied with manual actuators with gear reduction in lieu of a handwheel. Such actuators reduce the required rim-pull effort and often permit operation by one person in cases where several people would be required to seat the valve with a handwheel. While manual gear actuators slow down operation, they are often an attractive option for valves that are not operated frequently. Operating pressure and differential pressure should be specified.

Note: *Users sometimes specify that valves be operable at maximum differential pressure with very low rim-pull forces. This may require selection of gearing that may cause two problems: (1) literally thousands of handwheel turns for full-stroke valve operation and/or (2) capability to damage the valve easily with rim-pull forces that are readily applied by many operating personnel. Manual gear actuators with high ratios provide relatively little "feel" to the operator, and it is difficult to tell when a valve is fully open or closed. Good judgment should be exercised in specifying practical rim-pull force requirements.*

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1.1 Stop Valve Applications Guide (con't.)

Power Actuators
Where valves are inaccessible for manual operation or where relatively fast opening or closing is required, most Edward valves can be furnished with power actuators. The most commonly used actuators are electric actuators with torque- and position-control features. Users frequently have individual preferences on actuator brand names and type, so Edward valves are furnished with Limitorque, Rotork, Auma or other actuators to satisfy customer requirements.

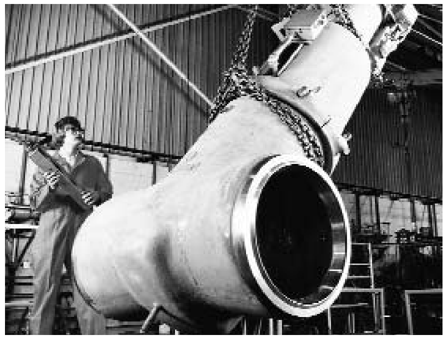
Edward Valves establishes actuator sizes and switch setting based on specific valve-application requirements, using a computer program that matches the valve and actuator operating characteristics to the service-pressure conditions. Unlike most valve manufacturers, Edward Valves makes this selection–not the actuator manufacturer–since we best know the requirements of our valve. However, we must also know the requirements of your application. As a minimum, requests for quotation should specify:

- Operating pressures – under-and over-seat and differential
- Maximum valve operating temperature
- Ambient conditions – temperature, humidity, radiation
- Motor power supply – AC voltage, frequency, and phase or DC voltage (including variance)
- NEMA rating
- Closing/opening time – if important. If not specified, standard nominal stem speed will be 4 inches/minute (100 mm/min) for globe valves and 12 inches/min (305 mm/min) for gate valves.
- Valve-stem plane – vertical (stem up or down) or horizontal
- Special accessories – position indicator, etc.

Any other special requirements should be clearly specified. If there are non-standard manual-override requirements, see the note above relative to rim-pull forces for manual gear actuators.

Stored-Energy Actuators
For critical service applications, special balanced Flite-Flow® valves and Equiwedge® gate valves are furnished with Edward stored-energy actuators that were developed and qualified to meet demanding nuclear power-plant requirements. These linear actuators are commonly installed on Main Steam Isolation Valves and Main Feedwater Isolation Valves (MSIV and MFIV) that must be adjustable to close in 3 to 10 seconds in the event of a line break.

The Edward actuator completed exhaustive qualification testing under elevated temperatures, radiation, seismic loadings and other conditions that realistically simulated the most severe operating conditions



encountered in actual service. In addition, extensive qualification testing was done on an Equiwedge MSIV in combination with an Edward actuator, and over 160 of these combinations are installed in nuclear plants on three continents.

The Edward actuator employs compressed gas-the stored energy of closure of the valve-in a compact, essentially spherical reservoir atop the piston of the valve-actuating cylinder. This integral construction eliminates reliance on external gas-storage tanks or interconnecting piping to connect the stored-energy gas to the power cylinder. Hydraulic fluid is pumped into the cylinder below the piston to open the valve, and regulated release of the fluid to a reservoir provides essential closing-speed control.

1.1.4 By-Passes and Drains

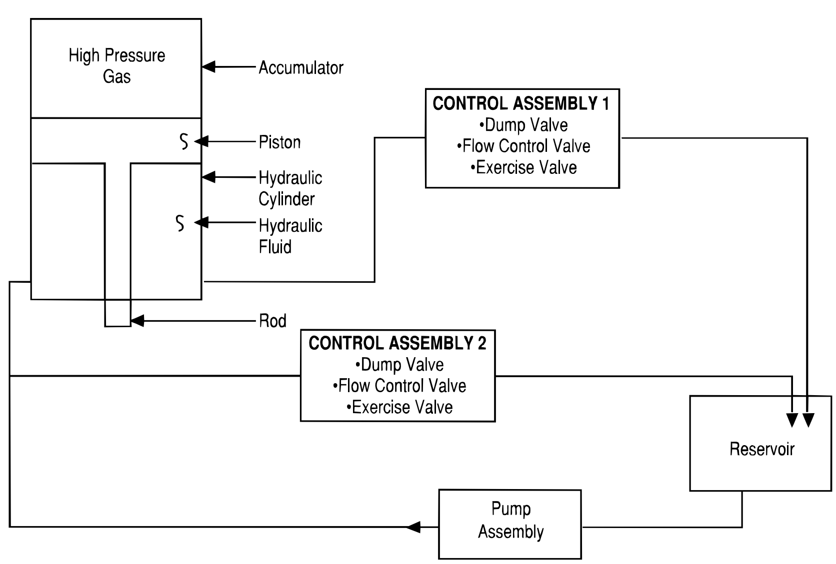
When specified, larger Edward cast-steel valves are furnished with valved by-passes and drains in accordance with ASME-ANSI B16.34 and MSS SP-45. Cast-steel stop valves employ forged-steel Edward globe stop valves, and cast-steel stop-check valves use forged steel Edward stop-check valves as by-pass valves. Sizes and by-pass valve figure numbers are as shown on page F-2.

Drain valves for all main valves are the same as the by-pass valves listed for stop valves. When drains are specified without valves, the standard drain for class 300 and 600 valves is a NPT tapped hole in the valve body, fitted with a pipe plug. For class 900 and higher-pressure valves, the standard drain is a pipe nipple, six inches (152 mm) long, socket-welded to the valve body.

Drain sizes are the same as by-passes. By-pass valves are particularly useful when opened before the main valve to permit controlled warming of the valve and downstream line in services involving steam or other hot fluids. By-passes also can be used to partially or fully balance the differential pressure across the main valve before opening where the downstream line or system is of limited volume. This facilitates opening of a gate valve or a glove valve with overseat pressure.

Large-volume systems may require larger by-passes for balancing in a reasonable time. If this is the case, a special by-pass size should be specified by the purchaser. It should be noted that actuated Edward Equiwedge gate valves do not require by-passes to permit opening if the full differential pressure is specified for actuator sizing. See page F-2 for tables of standard sizes and pressure classes for most applications.

General schematic of stored energy gas-hydraulic actuator.



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1.1.5 Stop Valve Application Chart

TYPE	ADVANTAGES	DISADVANTAGES	APPLICATIONS	LIMITATIONS
Globe 90° Bonnet	• Compact • Easy access to Handwheel or Actuator • Fast response	• High pressure drop • High torque • Heavy in large sizes	• Class 300 – 2500 steam & water • Other gasses and liquids • Usable for throttling	• Not for stem-down installations • Sizes **1/4** thru **24**
Angle	• Same as Globe • Replaces an Elbow • Lower pressure drop than Globe	• High torque • Heavy in large sizes	• Same as Globe	• Same as Globe
Globe Inclined Bonnet	• Lower pressure drop than Globe or Angle • May permit smaller size than Globe	• Same as Angle • Longest end-to-end length • Handwheel or Actuator on an Angle • Long stem travel slows response	• Class 600 – 4500 thru size 4 • Class 300 – 2500 thru size 24 • Otherwise, same as Globe	• Same as Globe
Equiwedge® Gate	• Lowest pressure drop • Lowest torque • May permit smallest size	• Not recommended for throttling • Long stem travel slows response with manual actuation	• Class 600 – 2500 steam & water • Other gasses and liquids • Main steam isolation	• Possibility of pressure binding • Sizes **2-1/2** thru **32**

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1.2 Check Valve Applications Guide

Foreword

Check valves are used in fluid circuits in applications similar to those in which diodes are used in electrical circuits. Reduced to simplest terms, the duty of most check valves is to allow flow in one direction and to prevent flow in the reverse direction. The ideal check would have zero resistance to flow in the normal flow direction and infinite resistance to flow (leakage) in the reverse direction. Of course, the ideal check valve should also be perfectly reliable and should require no maintenance.

There are many different types of check valves, and most do their duty well, giving long, trouble-free service. However, in the real world, no single type of check valve achieves the ideal performance characteristics users sometimes expect. In a very few cases, mismatching of check valves to the needs of fluid circuits has produced serious problems (noise, vibration, severe pressure surges and check-element failures with attendant gross leakage and consequential damage to other equipment). While it is not necessary for every application to be ideal, knowledge of the characteristics of each type of check valve should help system designers and valve users to select the best type and size intelligently. This knowledge should also help in assuring that serious problems are avoided.

Most check valves seen deceptively simple, with only one moving part-a poppet or flapper that appears capable of allowing flow in only one direction. However, this single mechanical part cannot be expected to take the place of a sophisticated control system that senses flow (direction, quantity, rate of change) and provides output to (1) open the valve fully when flow is in one direction and yet (2) close the valve to prevent flow and leakage in the reverse direction. Each type of check valve has features that enable it to perform one or more of its duties well, but each type also has weaknesses. The relative importance of these strengths and weaknesses is highly dependent on the requirements of individual applications.

The goal of this guide is to provide application engineers and users with practical advice on check valve selection and sizing, location in piping systems, preventive maintenance and repairs. Emphasis will be on Edward products, but comparisons will be provided in some cases with other types of check valves.

This guide is based on extensive testing of Edward check valves in sizes from NPS 1/2 through 18 as well as a reasonable sampling of other types. Since complete performance testing of every valve type, size and pressure class is not practical, predictions of the performance of some valves are based on mathematical models. However, the models are based on substantial test data and are believed to be reasonably accurate or conservative. The laboratory test files cover over forty years. Perhaps even more important, the files include feedback from substantial field experience-in fossil and nuclear-fueled power plants, refineries, chemical plants, oil fields and in countless other applications. It is hoped that this test and field experience will help others avoid problems and pitfalls in the application and use of check valves.

Introduction

This guide has been prepared to aid fluid-system designers in sizing and selecting check valves for industrial and power-piping systems. Guidance is also provided on valve orientation (inclination from horizontal, etc.) and on location of check valves with respect to other flow disturbances. In addition, this guide should aid users in planning preventive maintenance programs, performing maintenance and repairs when necessary, and in evaluating and correcting problems.

Emphasis in this guide is on selection and application of forged- and cast-steel Edward products, but comparisons with other types of check valves are given where this can be done based on valid information.

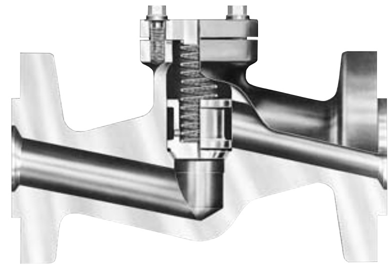
The Flow Performance section of this catalog provides equations and coefficients for the calculation of pressure drop and the flow required to assure full valve opening. In addition, that section provides most of the necessary supplemental data required for routine calculations, such as water and steam density.

This guide also provides caution notes relative to system-related problems to be avoided (such as piping vibration, flow instability, waterhammer). Some of these guidelines are qualitative and could involve further analysis. However, attention to these notes should help to avoid problems.

Finally, this guide addresses check valve maintenance. History indicates that preventive maintenance of check valves is often neglected, and this can lead to serious valve failures which may damage other equipment. The guidelines provided on periodic inspection and preventive maintenance should pay off in terms of reduced overall plant maintenance and repair costs.

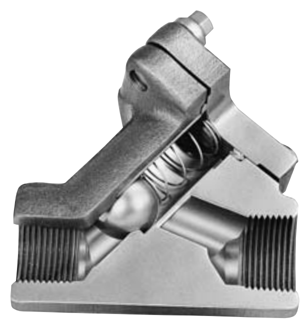
1.2.1 Check Valve Types and Typical Uses

While other types are sometimes encountered in power hydraulics and other specialized applications, four basic types of check valves are commonly used in industrial and power piping applications.

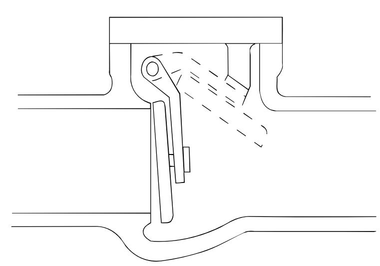


1-Lift Check Valves
The closure element is a poppet or disk that is lifted open by flow and which seats, usually on a mating conical surface in the valve body, under no-flow conditions.

2-Ball Check Valves
A lift check valve in which the closure element is a ball.

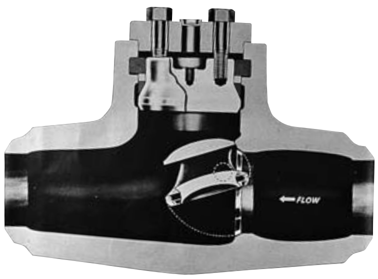


3-Swing Check Valves
The closure element is a pivoted flapper which is swung open by flow and which seats, generally against a mating flat surface in the valve body, under no-flow conditions.



1.2 Check Valve Applications Guide (con't.)

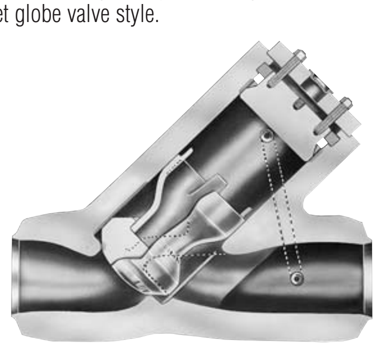
4-Tilting-Disk Check Valve
The closure element is a pivoted disk or flapper, somewhat like that in a swing check valve but with a pivot axis close to the center of the flow stream. It is swung open by flow and seats against a mating conical surface in the valve body under no-flow conditions.



There are many variations among these four basic types of check valves. For example, springs may be included to assist closure and counteract gravitational forces, and accessories may be provided for exercising or position indication. All Edward lift check valves employ body-guided disks with a piston-like extension to provide good guidance and resistance to wear. Accordingly, they are referred to in this guide as *piston-lift check valves*. In addition, Edward manufacturers stop-check valves which are piston-lift check valves that allow positive closure for isolation, just like globe stop valves.

Illustrations of the valve types manufactured by Edward are provided in this catalog, and brief notes on advantages, disadvantages, applications, and limitations are provided in the Check Valve Applications Chart (section 1.2.2). Some further highlights of the features of these valves are provided in the following paragraphs.

Edward Piston-Lift Check Valves
In both small forged-steel and large cast-steel Edward lines, three distinctly different valve body styles appear in the illustrations – inclined-bonnet globe valve style, angle valve style, and 90°-bonnet globe valve style.



With respect to check valve function, these valves are all similar, with only slightly different orientation limits as discussed in the Valve-Installation Guidelines section (1.3). The main difference between these systems is in flow performance:

- Inclined-bonnet piston-lift check valves produce low pressure drop due to flow when fully open. They have flow coefficients comparable to those of tilting-disk check valves and only slightly lower than provided by many swing check valves.

- In most cases, angle piston-lift check valves have lower flow coefficients and thus produce more pressure drop than inclined-bonnet valves, but they are superior to 90°-bonnet valves. Where a piping system requires a bend and a valve, use of an angle piston-lift check valve eliminates the cost and pressure drop of an elbow and the cost of associated piping welds or flanged connections.

- 90°-bonnet piston-lift check valves have the lowest flow coefficients and produce pressure drops comparable to 90°-bonnet globe valves. They are sometimes preferred in systems where pressure drop is not critical or where space requirements dictate a minimum size and easy access to a handwheel or actuator (on a stop-check valve).

Piston-lift check valves are generally the most practical type for small sizes, and they generally provide the best seat tightness. Small forged-steel piston-lift check valves normally include a disk-return spring, but may be ordered without springs. The Flow Performance section of this catalog and section 1.3 below address such valves, both with and without springs. Cast-steel piston-lift check valves have equalizer tubes which connect the volume above the piston with a relatively low-pressure region near the valve outlet. This feature allows a much larger valve opening (and higher flow coefficient) than would be possible otherwise, and it allows the valve to open fully at a relatively low flow.

The body-guided feature of Edward piston-lift check valves is an advantage in most services, because it assures good alignment of the disk with the valve seat and minimizes lateral vibration and wear. However, this feature may lead to sticking problems due to foreign-material entrapment in unusually dirty systems. Another inherent characteristic is that large piston-lift check valves may not respond rapidly to flow reversals and may cause water-hammer problems in systems where the flow reverses quickly [see the Pressure Surge and Waterhammer section (1.4.2)]. Since smaller valves display inherently faster response, historic files have shown no water-hammer problems with small forged-steel check valves.

Edward Stop-Check Valves
Stop-check valves offer the same tight sealing performance as a globe stop valve and at the same time give piston-lift check valve protection in the event of backflow. A stop-check valve is nearly identical to a stop valve, but the valve stem is not connected to the disk. When the stem is in the "open" position, the disk is free to open and close in response to flow, just as in a piston-lift check valve. When serving as a check valve, stop-check valves display the same advantages and disadvantages as discussed above for piston-lift check valves. Small forged-steel stop-check valves, except the Univalve® stop-check valves, employ a disk-return spring, and cast-steel stop-check valves have equalizer tubes that function in the same manner as those on comparable piston-lift check valves.

The stem in the stop-check valve may be driven either by a handwheel or an actuator, and it may be used either to (1) prevent flow in the normal direction when necessary for isolation or (2) supplement line pressure to enhance seat tightness in applications with pressure from the downstream side. Some users automate stop-check valves to give extra system protection against reverse flow and leakage. For example, an actuator may be signaled to close the valve when a pump is shut off; the disk closes quickly by normal check valve action, and the stem follows to seat the valve firmly a short time later.

1.2 Check Valve Applications Edward (con't.)

Edward Ball Check Valves
Ball check valves are offered only in small forged-steel configurations (size 2 and smaller) with inclined-bonnet bodies and ball-return springs. These valves are recommended over piston check valves, for service with viscous fluids or where there is scale or sediment in the system. The bolted-bonnet versions offer flow performance that is generally similar to that of equivalent piston-lift check valves, and they are the preferred ball check valves for most industrial and power-piping applications.

The threaded-bonnet hydraulic ball check valves are used primarily in very high pressure, low-flow applications with viscous fluids. They have lower flow coefficients that have proven acceptable for those services. These valves sometimes exhibit chattering tendencies when handling water, so they are not recommended for low-viscosity fluids.

A unique feature of the ball check valve is that the ball closure element is free to rotate during operation, allowing the ball and seat to wear relatively evenly. This feature, combined with the standard return spring, helps to promote positive seating even with heavy, viscous fluids.

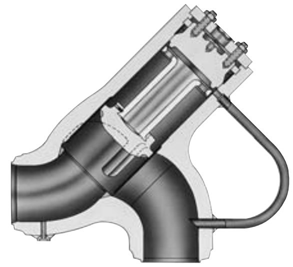
Edward Tilting-Disk Check Valves
Tilting-disk check valves are particularly well-suited to applications where rapid response and freedom from sticking are essential. Fully open valves of this type also exhibit low pressure drop. They have flow coefficients comparable to those of Edward inclined-bonnet piston-lift (Flite-Flow®) check valves and only slightly lower than provided by many swing check valves.

Tilting-disk check valves provide rapid response, because the center of mass of the disk is close to the pivot axis. Just as in a pendulum, this characteristic promotes rapid motion of the disk toward its natural (closed) position whenever the force holding it open is removed. This response can be valuable in applications where relatively rapid flow reversals may occur, such as in pump-discharge service where multiple pumps discharge into a common manifold. In such cases, the flow may reverse quickly, and the rapid response of the tilting-disk check valve minimizes the magnitude of the reverse velocity and the resulting waterhammer pressure surge. This characteristic also minimizes impact stresses on the disk and body seats. However, an extremely rapid flow reversal, as might be produced by an upstream pipe rupture, could cause a problem. See the Pressure Surge and Water Hammer section (1.4.2) for further discussion.

Size-6 and larger tilting-disk check valves have totally enclosed torsion springs in their hinge pins to help initiate the closing motion, but the disk is counterweighted to fully close without the springs. With the free pivoting action of the disk, this type of valve is highly immune to sticking due to debris in the system.

Tilting-disk check valve are superficially similar to swing check valves in that both operate on a pivoting-disk principle. However, the pivot axis in a swing check valve is much farther from the disk's center of mass, and this increases the "pendulum period" and hence the time required for closure in services with flow reversal. In addition, the one-piece disk in the tilting-disk check valve avoids the necessity of internal fasteners and locking devices, which are required to secure disks to pivot arms in most swing check valves. However, like swing check valves, tilting-disk check valves have hinge pins and bearings that are subject to wear due to disk flutter if the valve is not fully open and/or there are flow disturbances or instabilities. Such wear may product eccentricity of the disk and seat when the valve closes, leading to a degradation of seat tightness (particularly at low differential pressures). Applications involving severely unstable flow or prolonged service without preventive maintenance can lead to failures in which the disk separates completely from the hinge pins and will not close. Other sections of this guide address the flow conditions which may lead to problems as well as maintenance recommendations.

Edward Elbow-Down Check and Stop-Check Valves
Elbow-down piston-lift check and stop-check valves are similar to Flite-Flow valves except that the valve outlet is in the form of an elbow to direct the flow downward. These valves were designed specifically for applications in controlled-circulation power plants, and they have special clearances and other design features. Because of these special features, the sizing and pressure-drop calculation methods given



in the Flow Performance section of this catalog do not apply. However, special elbow-down valves can be furnished with conventional check valve design features for applications where this valve-body geometry is desirable.

Edward Combinations of Check and Stop-Check Valves
As noted in the Foreword to this section (1.2), no single type of check valve achieves ideal performance characteristics. The advantages and disadvantages noted in the Check Valve Applications Chart (section 1.2.2) and other information in this catalog should assist in selection of the best valve size and type for any specific application. However, the selection of any single valve may require undesirable compromises.

Some system designers and users specify two check valves in series for critical applications, and this does give some insurance that at least one valve will close even if the other valve fails. However, if two identical valves are used, a system characteristic that is troublesome to one valve could produce problems with both. In such cases, use of two valves does not assure double safety or double life. Sometimes it is worth considering the selection of two different types of check valve, each with advantages to offset disadvantages of the other.

One specific check valve combination has been used in applications of Edward valves to provide advantages that no single valve can offer. A tilting-disk check valve in series with a piston-lift check valve offers minimum waterhammer and freedom from sticking (from the tilting-disk) and good seat tightness (from the piston-lift check). The disadvantage is added pressure drop and cost, but the pressure-drop penalty is minor if the Flite-Flow inclined-bonnet piston-lift check valve is used. Even the cost penalty may be offset if a stop-check valve is used, because it may be able to take the place of a stop valve that would be required otherwise for isolation.

1.2.2 Check Valve Applications Chart

TYPE	ADVANTAGES	DISADVANTAGES	APPLICATIONS	LIMITATIONS
Piston Lift Check	• Very low pressure drop in inclined bonnet valves. • Relatively low pressure drop in angle valves. • Larger valves incorporate an external equalizer. • Minimum chatter due to flow disturbances. • Good seat tightness. • Forged steel valves with spring can be mounted in any orientation.	• Relatively high pressure drop in 90° bonnet valves. • Subject to “sticking” in very dirty systems.	• Class 300–4500 service. • High temperature steam and water. • Refining, petrochemical, chemical, etc. • Oilfield production. • Can be used in series with Tilting Disk Check to provide maximum line protection (advantages of both types).	• Sizes **1/4** thru **24**. • For orientation limits see VALVE INSTALLATION GUIDELINES. • For flow limits see Flow Performance section of this catalog.
Ball Check	• Wear on body seat and check element evenly distributed. • Long service life. • Forged steel valves with spring can be mounted in any orientation. • Available with either integral or threaded seat for hydraulic valve. • Low cost.	• High pressure drop. • Available only in small sizes.	• Class 600 and Series 1500 service. • Water, steam, refining, petrochemical, chemical, etc. • Service where scale and sediment exist. • Viscous fluids.	• Sizes **1/4** thru **2**. • For orientation limits see VALVE INSTALLATION GUIDELINES. • Not recommended for gas service at low flow rates. • For flow limits see Flow Performance section of this catalog.
Tilting Disk Check	• Very low pressure drop. • Straight through body design. • Very fast closing. • Minimizes disk slamming and waterhammer pressure surges. • Will not “stick” in dirty systems.	• Not recommended for service with rapidly fluctuating flow. • Seat tightness may deteriorate at low differential pressure.	• Class 600–4500 service. • High temperature steam and water. • Refining, petrochemical, chemical, etc. • Oilfield production. • Can be used in series with Piston Lift Check or Stop-Check to provide maximum line protection (advantages of both types).	• Sizes **2-1/2** thru **24**. • For orientation limits see VALVE INSTALLATION GUIDELINES. • For flow limits see Flow Performance section of this catalog.
Stop-Check	• See Piston Lift Check above. • Can be used for Stop valve service. • Stem can be lowered onto disk to prevent chatter at low flow. • Stem force can overcome “sticking.”	• See Piston Lift Check valve above.	• See Piston Lift Check above.	• See Piston Lift Check above.

1.3 Check and Stop-Check Valve Installation Guidelines

Figure 1
45° Inclined Bonnet Piston Lift Check Valves
Maximum Check Valve Orientation Limits

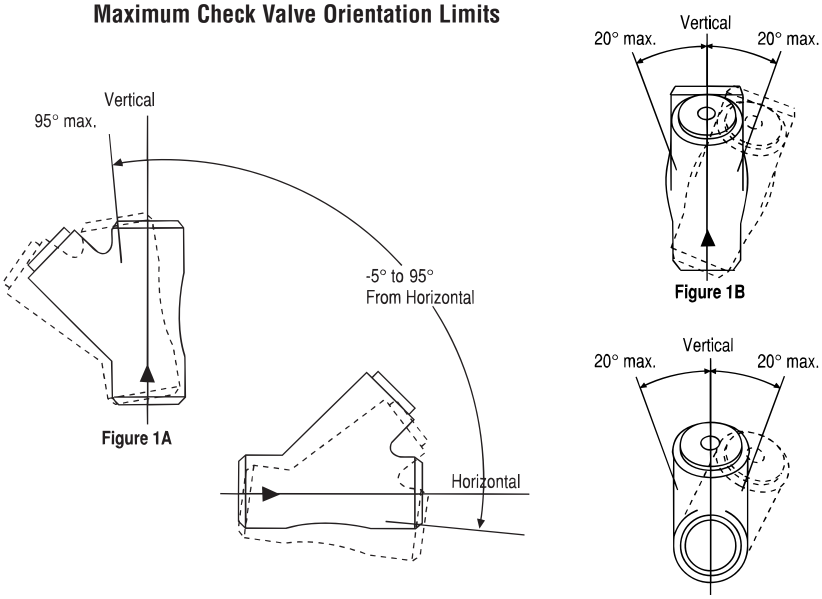


Figure 1A

Figure 1B

Figure 1C

Figure 2
90° Bonnet Piston Lift Check Valves
Maximum Valve Orientation Limits

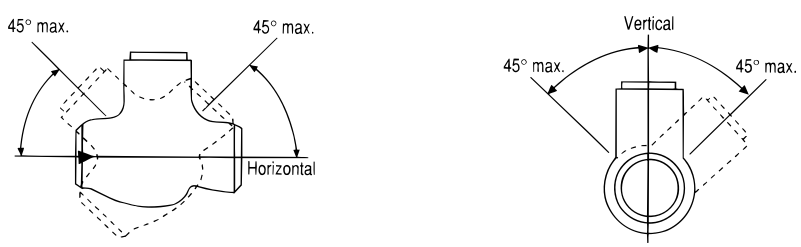


Figure 2A

Figure 2B

Note: *For piston lift check valves, any installation resulting in combined out of position orientation, such as a valve in an inclined line with a rollover angle as well, should limit the angle of the bonnet to the following:*
- *45° from vertical for angle and 90°bonnet valves.*
- *50° from vertical for inclined bonnet valves.*

Unlike stop valves, which can be installed in any position with little or no effect on performance, most check and stop-check valves have limitations as to their installed orientation. Although the normal installation is in a horizontal or vertical line (depending on valve type), check and stop-check valves can be installed in other orientations. It should be noted, however, that valves installed in other than the normal positions may exhibit a degradation of performance, service life and resistance to sticking, depending on the flow conditions and cleanliness of the line fluid. For maximum reliability, it is recommended that piston-lift check valves and stop-check valves be installed with flow axis horizontal (vertical inlet and horizontal outlet for angle valves) with the bonnet above the valve in a vertical plane. Following are maximum out-of-position orientations that may be used for less critical applications and which should never be exceeded.

- All Edward forged-steel check and stop-check valves (except Univalve® stop-check valves) are normally furnished with spring-loaded disks and may be installed in any position. The spring-loaded disk enables positive closure regardless of valve position. However, installed positions in which dirt or scale can accumulate in the valve neck should be avoided. An example of this would be an inclined-bonnet valve installed in a vertical pipeline with downward flow. If forged-steel valves are ordered without springs, the limitations below should be observed.

- Edward cast-steel Flite-Flow®, forged-steel Univalve, and inclined-bonnet check and stop-check valves without springs, when installed in vertical or near vertical lines, should be oriented such that the fluid flow is upward and the angle of incline of the line is not more than 5° past the vertical in the direction of the bonnet. When installed in horizontal or near horizontal lines, the valve bonnet should be up and the angle of incline of the line should be not more than 5° below the horizontal. See Figure 1A. Also, the roll angle of the valve bonnet should not be more than 20° from side to side for either vertical or horizontal installations. See Figures 1B and 1C. Consult your Edward Valves representative concerning installation limits of bolted-bonnet forged-steel check valves without springs.

- Edward cast-steel and forge-steel 90°-bonnet check and stop-check valves without springs should be installed with the bonnet up, and the angle of incline of the line should not be more than 45° form the horizontal. Also, the roll angle of the valve bonnet should not be more than 45° from side to side. See Figures 2A and 2B.

- Edward cast-steel and forged-steel angle check and stop-check valves without springs should be oriented such that the incoming flow is upward, and the angle of incline of the line should not be more than 45° in either direction. See Figure 3A and 3B.

- Edward tilting-disk check valves may be installed in horizontal lines and vertical lines and at any incline angle in between. When the incline angle is not horizontal, flow should always be up. The roll angle of the valve should not be more than 30° from side to side. See Figures 4A and 4B. Also, when installed in other than vertical lines, the bonnet should always be oriented up.

In each case described above, the limitations given for line inclination and bonnet roll angle should not be combined.

It should be understood that the information given in the section of this catalog entitled Flow Performance is based on traditional horizontal orientations. For other orientations, the pressure drop and flow required for full lift may be affected. In addition, seat tightness, particularly at low differential pressures, may be adversely affected.

Orientation restrictions may also exist for power-actuated stop-check valves. Most linear valve actuators are designed to be mounted upright and nearly vertical, although they can usually be modified for mounting in any position. When selecting a stop-check valve and power actuator, be sure to specify the mounting position desired if not vertical and upright.

1.3 Check and Stop-Check Valve Installation Guidelines (con't.)

Figure 3
Angle Piston Lift Check Valves Orientation Limits

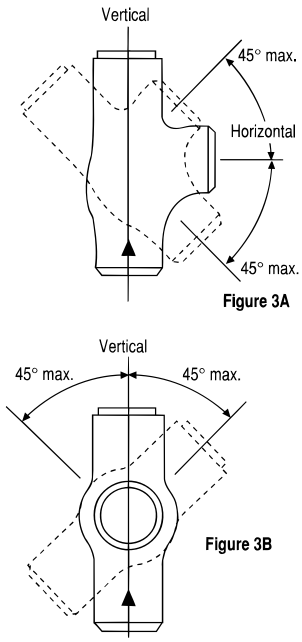
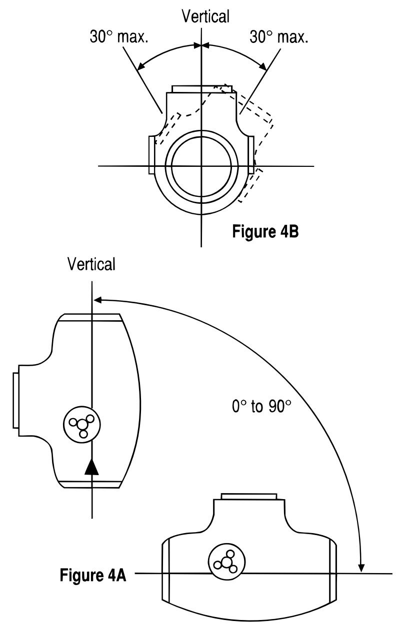


Figure 4
Tilting Disk Check Valves Orientation Limits



1.3.1 Adjacent Flow Disturbances

Check valves, like other valve types, are generally tested for performance and flow capacity in long, straight-pipe runs. Flow coefficients obtained from these tests are then used to predict the flow rate or pressure drop that will be experienced in actual applications. The ideal installation of a check valve in a plant would be in a long run of straight pipe so that performance would correspond to the test conditions. Since space limitations involved with many installations preclude such ideal straight-pipe runs, the effects of adjacent pipe fittings, control valves, pumps and other flow disturbances must be considered.

Figure 6
Non-uniform velocity profile at blower or pump discharge can affect stability.

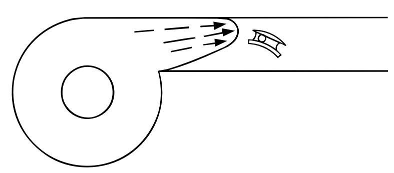
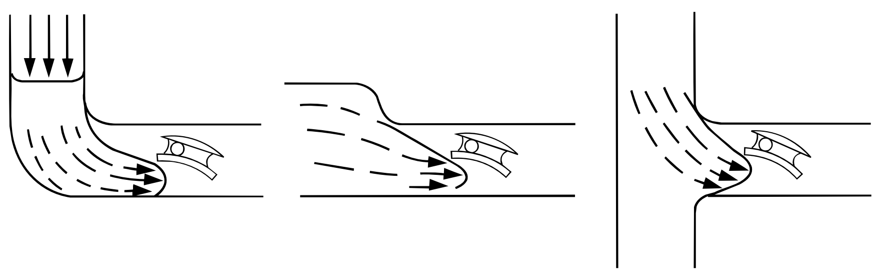
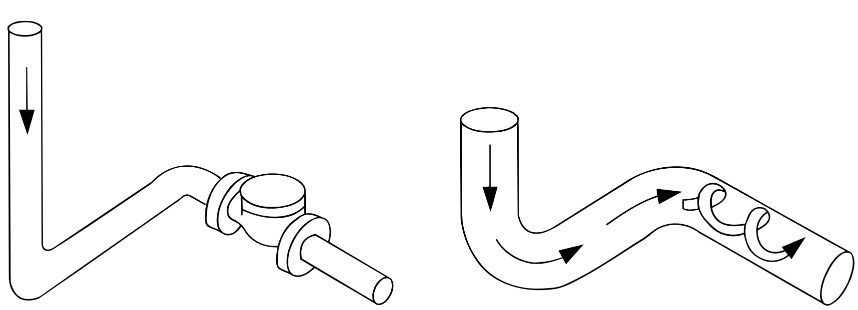


Figure 5
Pipe fittings near valves may produce instability because of velocity profile distortion



Previously published data have indicated that flow disturbances, particularly upstream disturbances, may significantly affect check valve performance. It has been reported that valve flow capacity may be significantly reduced as compared to that measured in straight-pipe tests, and there have been strong suggestions that such disturbances aggravate check valve flutter and vibration. Since these conditions could degrade valve performance and contribute to rapid wear and premature valve failure, they are important factors in evaluating check valve applications. Figure 5 illustrates how upstream pipe fittings may alter the flow profile entering a check valve, crowding it to one side or the other. A similar distortion occurs in a valve located near the discharge of a centrifugal pump or blower, as shown in Figure 6. Elbows in two planes cause a flow stream to swirl, which might produce unusual effects on a check valve installed as shown in Figure 7.

Figure 7
Elbows in two places cause swirl which can promote instability.



1.3 Check and Stop-Check Valve Installation Guidelines (con't.)

Since there was no known way to predict the effects of flow disturbances on check valves by mathematical models, Edward conducted extensive testing of size 2, 4, 8 and 10 check valves in straight-pipe runs and in piping with upstream flow disturbances. Figures 8 and 9 illustrate typical flow-test setups.

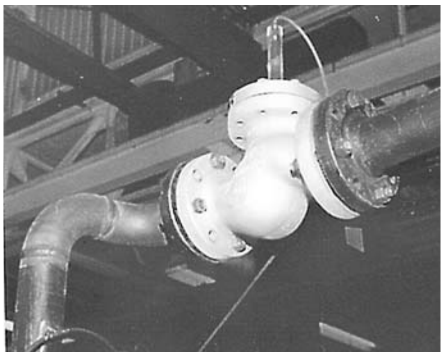


Figure 8
Size 4 Class 600 90° bonnet piston lift check valve with two upstream elbows (out of plane). This arrangement produces swirl as shown in Figure 7.

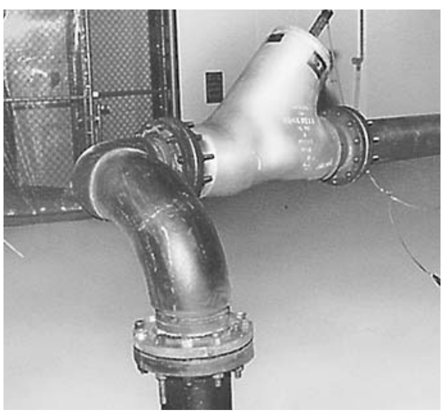


Figure 9
Size 10 Class 1500 Flite-Flow® inclined bonnet piston lift check valve with two upstream elbows. Test loop capacity permitted tests with line velocity over 20 ft./sec. (6 m/sec.).

In most tests, room temperature water was the flow medium, but limited straight-pipe testing was performed with air. The valves tested included Edward piston-lift check (inclined-bonnet, angle and 90°-bonnet), tilting-disk check valves and a size-4 swing check valve manufactured by another company. The tests were designed to evaluate the effects of flow disturbances on (1) valve stability, particularly when partially open; (2) flow rate required to open the valve fully; and (3) the flow coefficient (C_V) of the valve. The flow disturbances evaluated included single and double (out of plane) 90° elbows in various orientations immediately upstream of the check valves. In addition, the effects of a throttled, upstream control valve were simulated with an offset-disk butterfly valve (at various throttle positions) mounted immediately upstream, as well as at five and eleven pipe diameters upstream, of the check valves.

With few exceptions, tests with 10 or more diameters of straight pipe upstream of check valves produced little cause for concern. In water flow tests, visual position indicators usually showed only minor disk "wobble" or very small open-close flutter (e.g. less than 1° total rotation of a tilting disk), even at very low flows and small valve openings. The only conditions that produced severe instability were those involving air flow at very low pressures (below 50 psi or 3.4 bar) and valve openings less than 20%. Such conditions produced significant cyclic motion, with disks bouncing on and off the body seats. In view or the many uncertainties in applying laboratory test results to service conditions, it is considered prudent to avoid operating conditions which produce check valve openings of less than 25%, even in ideal straight-pipe applications.

Highlights of the results of the Edward tests with flow disturbances are given in Table A on page G17. The test program clearly showed that upstream flow disturbances do affect check valve performance, but the effect is not always predictable. The magnitude of the effect can vary, depending on the type and even the size of the valve. In some cases, even the direction of the effect (improvement or degradation) varies from valve to valve. Nevertheless, some general observations on the results of these tests are:

- Single and double upstream elbows produced less severe effects on check valve performance than had been expected, and some valves displayed no discernible effects. For example, Edward angle piston-lift check valves exhibited the same stability, lift and flow coefficients (C_V) with upstream elbows as with straight pipe. In tests of other types of valves, upstream elbows produced both beneficial and adverse effect to various degrees.

- In each case where a check valve was tested with a throttled butterfly valve immediately upstream, there were significant effects on performance. The effects included increased disk flutter and reduced valve opening at a given flow, as compared to straight-pipe performance. In some cases, full check valve opening could not be achieved at any flow within the capabilities of the test loop.

Even where full opening was obtained, some valves continued to flutter on and off their stops. These effects were worst when the butterfly valve was most severely throttled (smallest opening and highest pressure drop). In the worst cases, the butterfly valve exhibited audible cavitation, but it is not clear whether the adverse effects resulted from simple flow distortion or the two-phase flow stream from the cavitating butterfly valve.

In similar tests with the butterfly valve moved 5 diameters upstream of the check valve (but with similar throttling), the adverse performance effects were decreased significantly but not eliminated. When the butterfly valve was moved 11 diameters upstream of the check valve, normal check valve performance was restored.

The results of these tests were enlightening, but they must be combined with observations based on field experience. For example, while upstream elbows produced less severe effects than expected, there were still adverse effects on some valves. It is difficult to extrapolate a laboratory test to years of service in a plant installation, but Edward se vice files include an interesting and relevant incident. Two size-12 tilting-disk check valves in one plant had hinge-pin failures over a time period of several months after 25 years of service. While this incident might best be cited as a case for more inspection and preventive maintenance, the details of the installation were investigated. It was determined that the flow rates were in a range that should have assured full disk opening, but the valves were installed close to upstream elbows.

Users of this catalog may wish to refer to EPRI Report No. NP 5479 (see the Sources for Additional Information section of this catalog) for further data on the performance of swing check valves in tests similar to those conducted by Edward. The size-4 swing check valve used in the Edward test program had a stop positioned to restrict the disk-opening angle to about 38°. This valve opened fully at a relatively low flow and exhibited reasonably stable performance. The tests sponsored by EPRI showed that other swing check valves (with less restrictive stops) exhibited larger amplitudes of flutter than were observed in comparable Edward tests.

1.3 Check and Stop-Check Valve Installation Guidelines (con't.)

The following guidelines are based on Edward tests and field experience, combined with other published information:

• If possible, check valves near flow disturbances should be sized to be fully open, preferably by a good margin, even at the lowest sustained flow rate anticipated for each application. The Flow Performance section of this catalog provides methods for sizing Edward check valves for new installations or for evaluating existing applications. When flow-induced forces load a valve closure element firmly against a stop, it is less likely to flutter and suffer from rapid wear.

Full opening does not guarantee freedom from problems if the margin is not sufficient to provide a firm load against the stop. Equalizers on Edward cast-steel piston-lift check and stop-check valves enhance this margin and provide good stop loading, but flow disturbances may cause other valve disks to bounce on and off their stops. This "tapping" phenomenon may cause faster wear than flutter about a partially open position. For this reason, the minimum sustained flow rate through a tilting-disk check valve near flow disturbances should be about 20% greater than the flow rate required to just achieve full opening.

If it is not possible to assure full opening of a check valve at minimum flow conditions, at least 25% opening should be assured. Valves operating at partial opening for significant periods of time should be monitored regularly to determine if there is insta bility or wear.

• In view of uncertainties associated with long-term effects of flow disturbances, it is recommended that a minimum of 10 diameters of straight pipe be provided between the inlet of a check valve and any upstream flow disturbance (fittings, pumps, control valves, etc.), particularly if calculations indicate that the check valve will not be fully open for a substantial portion of the valve service life. There should be a minimum of 1 to 2 diameters of pipe between the check valve and the nearest downstream flow disturbance.

• In the specific case of upstream elbows, reasonably successful performance should be attainable with 5 diameters of straight pipe between an upstream elbow and a check valve if the valve will not be partially open for a significant portion of its service life. Tests described in EPRI Report No. NP 5479 indicate that elbows installed 5 diameters or more upstream had a negligible effect on swing check valves, and this is expected to be true for other check valve types. Even less straight pipe may be satisfactory, but such close spacing should be reserved for applications with very tight space constraints. More frequent inspection and preventive maintenance should be planned for valves in such installations.

• In the specific case of throttled upstream control valves, the minimum requirement of 10 upstream pipe diameters should be adhered to rigidly. Calculations indicating full valve opening based on straight-pipe tests cannot be trusted to prevent problems, because severe flow disturbances may prevent full opening. Even greater lengths of straight pipe should be considered if the control valve operates with very high pressure drop or significant cavitation.

• Users with existing check valve installations that do not meet these guidelines should plan more frequent inspection and preventive maintenance for such valves. If a check valve is installed close to an upstream control valve that operates with a high pressure drop, considerations should be given to a change in piping or valve arrangements.

Table A - Effects of Upstream Flow Disturbances on Check Valve Performance

VALVE SIZE & TYPE	SINGLE ELBOW[1] AT VALVE INLET	DOUBLE ELBOWS (OUT OF PLANE) AT VALVE INLET	THROTTLED BUTTERFLY VALVE		
			AT VALVE INLET	5 DIAM. UPSTREAM	11 DIAM. UPSTREAM
Size 2, Inclined-Bonnet Piston-Lift Check	Higher Lift for Same Flow; Disk Flutter at Lower Lifts[2]	Higher Lift for Same Flow	NA	NA	NA
Size 4, Angle Piston-Lift Check	No Effect	No Effect	NA	NA	NA
Size 4, 90°-Bonnet Piston-Lift Check	Same, Lower or Higher Flow for Full Lift	No Effect	Disk Flutter and Chatter: Failure to Achieve Full Open	NA	NA
Size 4, Swing Check	Smaller Opening for Same Flow	Smaller Opening for Same Flow	Larger Opening for Same Flow; Disk Flutter	NA	NA
Size 8, Angle Piston-Lift Check	No Effect	NA	NA	NA	NA
Size 8, 90°-Bonnet Piston-Lift Check	Disk Flutter at Partial Lift	NA	NA	NA	NA
Size 10, Inclined-Bonnet Piston-Lift Check	Same or Lower Lift for Same Flow; Slight Disk Wobble	No Effect	Failure to Achieve Full Open; Disk Flutter and Chatter	Failure to Achieve Full Open	No Effect
Size 10, Tilting-Disk Check	No Effect	Minor Flutter	Same, Lower or Higher Lift for Same Flow; Disk Flutter and Chatter	Minor Flutter	No Effect

[1]Tests were conducted with single 90° elbows in the horizontal plane and in the vertical plane (with flow both from above and below).
[2]One size-2 valve exhibited flutter at lower lifts; another was stable.

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1.3 Check and Stop-Check Valve Installation Guidelines (con't.)

1.3.2 Other Problem Sources

In addition to the fundamentals of check valve selection, sizing and installation, several other potential sources of check valve problems should be considered in applications engineering or, if necessary, in solving problems with existing installations:

• Piping-System Vibration
In other sections of this guide, it has been noted that check valve damage or performance problems may result from flow-induced flutter or vibration of the closure element. Very similar damage may result from piping-system vibration. Such vibration may originate at pumps, cavitating control valves or other equipment. Check and stop-check valves are susceptible to vibration damage, because the check element is "free floating" when partially open, with only the forces due to fluid flow to balance the moving weight. Impact damage and internal wear may result if the valve body vibrates while internal parts attempt to remain stationary. This condition may be avoided by adequately supporting the piping system near the check valve or by damping vibration at its source. Of course, it is helpful to assure that the check element opens fully, because flow forces at the disk-stop help to inhibit relative motion.

• Debris in Line Fluid
Debris in the flow stream can cause damage and performance problems in check and stop-check valves. Debris entrapped between the disk and seat may prevent full closure and lead directly to seat leakage. If hard particles or chips are in the debris, they may damage the seating surfaces and contribute to seat leakage even after they are flushed away. Debris caught between the disk and the body bore of a piston-lift check valve can cause the disk to jam and prevent full opening or closing. To insure best check valve performance and seat tightness, line fluids should be kept as clean as practical. As noted before, tilting-disk check valves are particularly resistant to sticking or jamming, but they are no more resistant to seat damage than other types.

• Unsteady (Pulsating) Flow
An unsteady flow rate can lead to rapid check valve damage, particularly if the minimum flow during a cycle is not sufficient to hold the valve fully open. The valve may be damaged just because it does what a check valve is designed to do – open and close in response to changes in flow. As an example, a check valve installed too close to the outlet of a positive displacement pump may attempt to respond to the discharge of each cylinder. If the mean flow during a cycle is low, the disk may bounce off the seat repeatedly in a chattering action. If the mean flow is higher, the disk may bounce on and off the full-open stop. Such pulsating flows may be difficult to predict. For example, a steam leak past the seat of an upstream stop valve may produce a "percolating" action in a line filled with condensate and cause a check valve to cycle. Such problems may only be discovered by preventive maintenance inspections.

• Vapor Pockets in Liquid Piping Systems
Unusual phenomena are sometimes observed in piping systems containing hot water that partially vaporizes downstream of a closed check valve. Vapor pockets at high points may collapse suddenly when the check valve opens (due to the start-up of a pump, for example). This collapse may be remote from the check valve and have no effect on the check valve performance. However, if a vapor pocket exists in the upper part of a piston-lift check or stop-check valve body (above the disk), the collapse may generate unbalanced forces in the direction of disk opening. Since the vapor offers little fluid resistance, rapid acceleration of the disk toward the fully open position may occur. In extreme cases, the disk or bonnet stops may be damaged due to impact. Such thermodynamic quirks are difficult to anticipate when designing a piping system and are sometimes as difficult to diagnose if they occur in an existing installation. Changes in piping arrangements or operating procedures may be necessary if severe problems occur. It is possible that similar problems may occur during low-pressure start-up operations in unvented liquid-piping systems.

1.4 Check Valve Performance

1.4.1 Check Valve Seat Tightness

Edward check valves are factory-tested with water in accordance with MSS SP-61 (Manufacturers Standardization Society of the Valve and Fittings Industry, Inc.) at an overseat pressure of 1.1 times the pressure ratings of the valve. While check valves are allowed leakage rates up to 40 ml/hr per unit of nominal valve size by MSS SP-61, Edward allows no more than 5% of this leakage for cast-steel valves and no visible leakage for forged-steel valves. Tilting-disk and forged-steel check valves are then tested again at a reduced pressure with allowable leakage rates which are less than the MSS SP-61 requirements.

Closed check valve closure elements (disk, ball, flapper, etc.) are acted on by a combination of forces produced by gravity, springs (where applicable) and reversed differential pressure. While gravity and spring forces help to position the closure element into the substantially closed position, metal-to-metal seating check valves typically rely on pressure forces to produce the seating loads necessary for good seat tightness.

Some metal-seated check valves do not produce good seat tightness at low differential pressures, particularly when the pressure increases from zero. A threshold level of differential pressure is required to produce uniform metal-to metal contact and restrict leakage to a reasonable rate. An even higher level is required to assure that a valve meets leakage-rate criteria like those in MSS SP-61. Unfortunately, these levels of differential pressure are difficult to predict; they vary with valve type, condition and orientation (and with cleanliness of line fluid).

Tests of new valves in horizontal lines show that cast-steel inclined-bonnet and 90°-bonnet piston-lift check and tilting-disk check valves seal off reasonably well at under 50 psi (3.4 bar) when differential pressure increases from zero. Small forged-steel ball and piston-lift check valves are less consistent, sometimes seating at less than 50 psi (3.4 bar) and sometimes requiring 250 psi (17 bar) or more. This "seating" action often occurs suddenly when the pressure forces shift the closure element into good metal-to metal contact with the body seat, and leakage generally continues to decrease as the pressure is increased. Once seated, most valves seal well if pressure is reduced below the threshold required for initial seating, but the seat tightness with reducing pressure is also difficult to predict.

Some of the Edward check valves described in this catalog have been manufactured with "soft seats" to provide improved seat tightness at low differential pressures. This design feature includes an elastomeric or plastic sealing member on the valve closure element to supplement the basic metal-to-metal seating function. Since the design and material selection for these sealing members are very sensitive to pressure, temperature and compatibility with the line fluid, there are no standard, general-purpose, soft-seated valves. Consult Edward Valves for further information about specific applications.

Foreign material in the flow medium is a major source of leakage problems in many valves. Because of the limited seating forces in check valves, dirt has a far greater effect on the tightness of these valves than other types. Attention to cleanliness of the fluid is necessary where good check valve seat tightness is desired.

Incorrect sizing or misapplication of a check valve can also lead to leakage problems. Chattering of the closure element on its seat due to insufficient flow or pressure can cause damage to the seat or closure element and result in leakage.

In applications where check valve leakage is a problem, a stop-check valve may offer the solution. Stem load from a handwheel or actuator can provide the necessary seating force independent of pressure. Of course, the stem must be returned to the "open" position to allow flow in the normal direction. Consult Edward Valves about applications that are usually sensitive to leakage.

A complete treatment of the subject of pressure surge and waterhammer is beyond the scope of this catalog, but some discussion is provided so that application engineers may appreciate the significance of the problem as it relates to check valves.

1.4.2 Pressure Surge and Waterhammer

One part of the problem is that the terminology or jargon is not consistently used. For example, "waterhammer" or "steam hammer" is sometimes used to describe the implosion which occurs when water enters a hot, low pressure region and causes a steam void to collapse. This has occurred in systems with a failed check valve, where the water came back from a large reverse flow through the check valve. However, the more common "waterhammer" problem associated with check valves occurs as a result of the check valve closing and suddenly terminating a significant reversed flow velocity. This problem is generally associated with valves handling water or other liquids. A similar pressure surge phenomenon may be encountered with steam or gas, but it is generally much less serious with a compressible flow medium.

Waterhammer is a pressure surge produced by the deceleration of a liquid column, and it involves pressure waves that travel at close to the velocity of sound through the fluid. It is commonly illustrated in texts by an example involving rapid closure or a valve in a long pipe. For such a case, it can be shown that instantaneous closure of a valve in a room-temperature water line will produce an increase in pressure of about 50 psi (3.4 bar) above the steady-state pressure for every 1 ft/sec (0.30 m/sec) decrease in water velocity. Even if the valve does not close instantaneously, the same pressure increase would develop if the upstream pipe is long enough to prevent reflected pressure waves from reaching the valve before it closes. The waves of increasing pressure that are generated by the closing valve "reflect" from a constant-pressure reservoir or vessel, if present in the system, and return to the valve as inverted waves that decrease pressure. A solution to the "textbook problem" is to slow down the valve closure so that the reflected pressure waves attenuate the surge. However, this is not necessarily the best approach in the case of a check valve.

In a check valve, the fluid velocity is forward before the valve starts to close, but it reduces due to some system action (e.g., a pump is shut off). If the velocity reverses before the valve closes, a waterhammer surge will be produced by a conventional check valve that is nearly proportional to the magnitude of the maximum reversed velocity. Figure 10 provides curves illustrating flow transients associated with different types of systems and flow interruptions. The graphs illustrate velocity in the pipe, forward and reverse, versus time on arbitrary scales. The following discussions describe each of the curves:

- **Curve A** illustrates flow coast-down in a simple circulating loop, such as a cooling system, following switch-off of pump power. The momentum of the pump impeller and the fluid keeps the fluid going forward until it is decelerated and finally stopped by friction. There would be no need for a check valve to prevent reverse flow in this system, but one might be included to permit pump maintenance without draining other equipment. In normal operation of this system, the check valve could produce no waterhammer.

- **Curve B** illustrates an application with a pump feeding a high-pressure system with a fairly large volume. It might represent a boiler feed system of a pump feeding a high reservoir. In this case, assuming similar momentum in the pump and fluid, forward flow continues for a while after the pump is switched off, but the downstream pressure decelerates the flow more rapidly and then reverses its direction. Without a check valve, the reverse flow would increase and stabilize at some value, unless the downstream system pressure declined. In the illustration, the magnitude of the maximum reverse velocity is drawn less than the initial forward velocity, but it might be higher in some systems.

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1.4 Check Valve Performance (con't.)

Figure 10 - Flow Reversal Transients

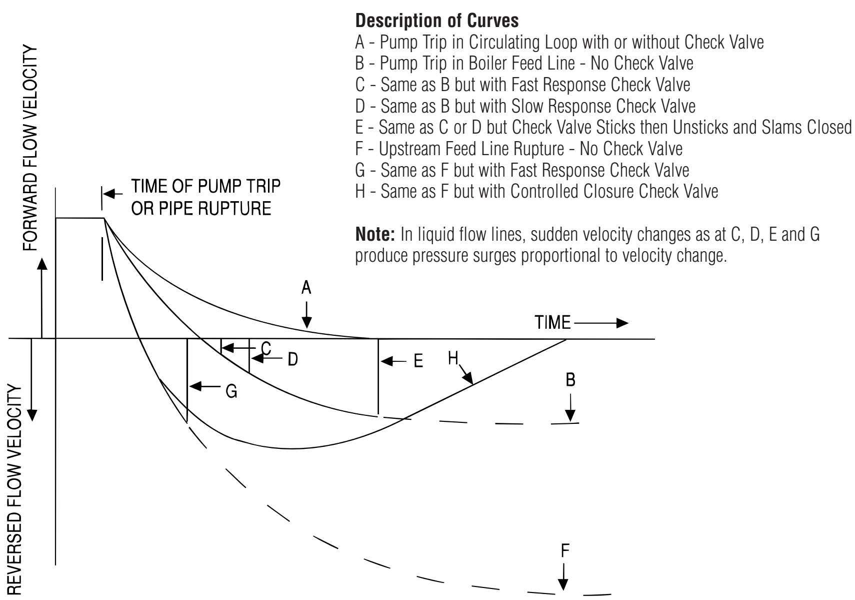
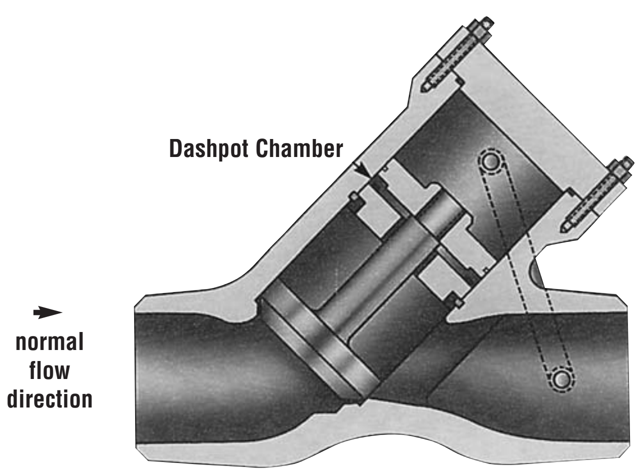


Figure 11 - Controlled Closure Check Valve (CCCV)



• **Curve C** illustrates what would happen in the system described for Curve B with a fast-response check valve (e.g., a tilting-disk type) installed. As discussed in the Foreword to this guide, an "ideal" check valve would allow no reverse flow and would close exactly at the time the velocity curve passes through zero; there would be no waterhammer. A "real" check valve starts closing while the flow is still forward, but it lags the velocity curve. With fast response, it closes before a high reverse velocity develops, thus minimizing the waterhammer surge.

• **Curve D** illustrates the same system with a check valve that responds just a bit slower. It shows that just a small increase in check valve lag may allow a large increase in reverse velocity (and a corresponding increase in waterhammer surge pressure).

• **Curve E** illustrates an accidental situation that might develop with a severely worn valve or a dirty system. If a check valve in the system described above should stick open, it might allow the reverse velocity to build up so as to approach that which would occur without a check valve. If the reverse flow forces should then overcome the forces that caused the sticking, the resulting valve stem could cause a damaging waterhammer surge.

• **Curve F** illustrates what might happen in the system described for Curve B if there were a major pipe rupture just upstream of the check valve. With free discharge through the open end, the flow would decelerate much more rapidly and, without a check valve, reach a much higher reverse velocity.

• **Curve G** shows the response of the system in Curve F if even a fast-response conventional check valve were to be used. With a flow deceleration this rapid, even a small lag may result in a very high reverse velocity to be arrested and a correspondingly high waterhammer surge.

Fortunately, it is not necessary to design every piping system with a check valve to cope with a pipe rupture. However, this requirement has emerged in some power-plant feedwater piping systems. Edward analyses and tests have shown that even the most rapid-responding conventional check valve could produce unacceptable waterhammer surges. This led to the development of the special controlled-closure check valve (CCCV–see Figure 11). Since high reverse velocities are inevitable, the CCCV solves the problem the way the "textbook problem" discussed above is solved–by closing slowly. The CCCV is a piston-lift check valve, but it has an internal dashpot which slows the closing speed of the valve. Closing speed depends on the rate at which water is squeezed out of the dashpot chamber, through flow paths that are sized for each application.

• **Curve H** illustrates the velocity variation in the pipe-rupture situation described for Curve F, but with a CCCV in the line. In this case, the maximum reverse velocity might even be higher than in Curve G, but it is decelerated back to zero slowly, allowing reflected reducing-pressure waves to minimize the resulting waterhammer surge. Figure 12 provides a comparison between a conventional check valve and a CCCV for a specific pipe-rupture situation. Note that the conventional check valve closes in 0.07 seconds as compared to 1.0 seconds for CCCV. As a result, the conventional check valve produced a surge of 3000 psi (207 bar) while the CCCV limits the surge to 600 psi (41 bar). These characteristics have been demonstrated in tests and can be duplicated in computer-based dynamic analysis simulations of specific valves and systems.

While the CCCV solves a special problem, even this sophisticated product does not fulfill the definition of an ideal check valve. By closing slowly, it allows significant reverse blow before it seats. This characteristic might be undesirable in common pump-discharge applications, because the reverse flow might have adverse effects on pumps or other equipment. Studies of systems designs sometimes show that fast-response check valves, such as the tilting-disk type, should be retained at pump discharge points where an upstream pipe rupture is unlikely, with CCCVs applied at locations where an upstream pipe rupture could cause serious consequences (e.g., in feedwater lines inside the containment vessel of a nuclear power plant).

1.4 Check Valve Performance (con't.)

Figure 12
Example Comparison of Closure Time and Surge Pressure
Conventional vs Controlled Closure Check Valves

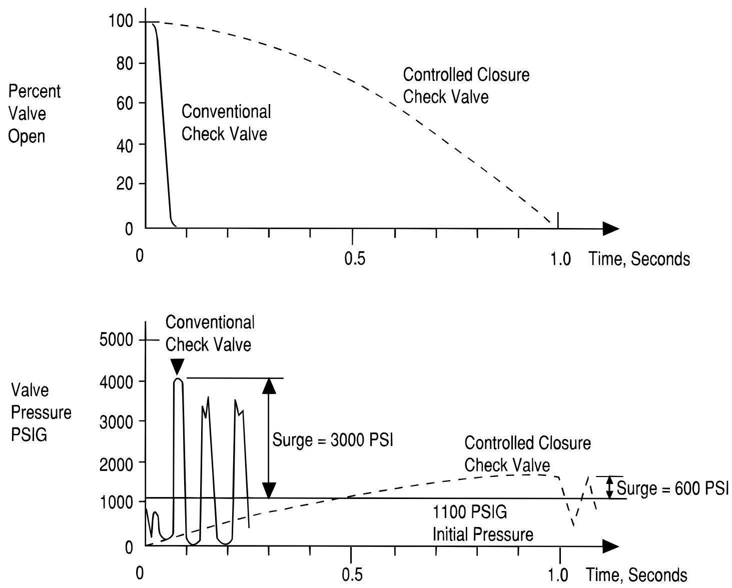
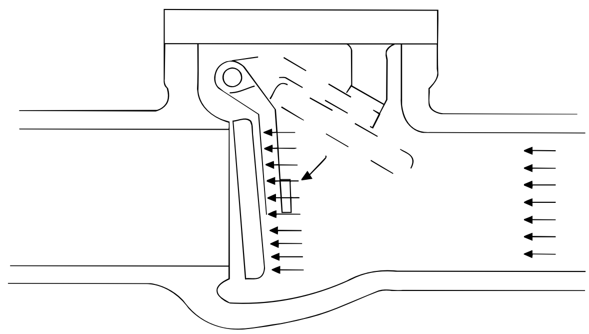


Figure 13
Reverse Flow in Conventional Swing Check Valve - Just Before Closing



In Curves C, D, E, and G of Figure 10, it may be noted that the final terminations of reverse velocity are shown as substantially vertical lines. This does not imply that the valve closes instantaneously. However, tests of conventional check valves show that the reverse velocity in the pipe containing the valve does terminate almost instantaneously. This apparent contradiction may be understood by referring to Figure 13, which illustrates a check valve approaching the closed position with reverse flow (while the illustration depicts a swing check valve, the flow condition discussed here would be much the same with a poppet or disk in a conventional lift check or piston-lift check valve).

The key observation from Figure 13 is that a column of fluid follows the closure element at roughly the same velocity that the closure element has as it approaches its seating surface in the valve body. While the valve may start to close while the flow velocity is still forward (see Figure 10), an undamped check valve has little effect on pipe flow during closure, and the disk velocity is about the same as the reverse flow velocity in the pipe at the instant just before closure. Since the disk is stopped substantially instantaneously when it makes metal-to-metal contact with the body seat, the reverse flow velocity in the pipe must also be arrested instantaneously. Because of this characteristic, the surge produced by the slam of a conventional check valve cannot be attenuated significantly by reflected reducing-pressure waves, and the surge tends to be relatively insensitive to system pipe lengths.

In some check valve applications, problems have been observed due to a phenomenon that is related to waterhammer but not as widely recognized. When a high-pressure wave is produced on the downstream side of a check valve at closure, a reverse low-pressure wave is produced on the upstream side. If this low-pressure wave reduces the fluid pressure to below the saturation pressure of the fluid, a vapor pocket can form. This can be compared to a tensile failure of the flow stream, and it is sometimes referred to as *column separation* or *column rupture*. This vapor pocket is unstable and will collapse quickly, with an implosion that produces a high-pressure "spike." It is possible for this pressure surge to exceed the one initially produced on the downstream of the check valve. Instrumented laboratory tests have shown that the upstream pressure spike sometimes causes the disk to reopen slightly and "bounce" off its seat once or twice. In very rare occasions, sometimes involving systems with multiple check valves, this characteristic has been known to amplify, leading to damaging pipe vibrations.

In summary, waterhammer can produce complex problems in check valve applications. Numerical solutions to these problems require sophisticated computer-based dynamic analyses of both the check valve and the fluid in the piping system. This catalog does not provide the methods for making such analyses; instead, the information in this section is intended to assist fluid-system designers in avoiding the problem.

Users who already have check valves in liquid flow lines that emit loud "slams" when they close should be aware that the noise is probably associated with pressure surges that could lead to fatigue problems in the valve, piping or other components. Where the existing check valve is a piston-lift check or stop-check valve, the solution could be to add a tilting-disk check valve in series with the existing check valve to gain the advantages of both valve types. Where the existing valve is a swing check valve, replacement by a tilting-disk check valve might be considered. See the section of this catalog entitled Check Valve Types and Typical Uses (1.2.1) for a discussion of the strengths and weaknesses of the various valve types.

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1.4 Check Valve Performance (con't.)

1.4.3 Check Valve Accessories and Special Features

Edward Check valves can be provided with various accessories which are used to induce check-element motion (exercise) or indicate check-element position. Some of the features available are as follows:

- Visual disk-position indicator for tilting-disk check valve
- Electrical open/close position indicator for tilting-disk or cast-steel piston-lift check valve
- Manual or pneumatic actuator to partially open tilting-disk check valve under zero differential pressure
- CCCVs can be furnished with an injection port which allows the valve disk to be exercised by injecting water into the dashpot chamber when the valve is under a zero differential pressure.

1.4.4 Check/Stop-Check Valve Periodic Inspection and Preventive Maintenance

Periodic inspection and preventive maintenance of check and stop-check valves should be performed to insure that the valves are operating properly. Bonnet-joint leakage and packing leakage on stop-check valves are easy to detect. Seat leakage of a check or stop-check valve might be indicated by one of the following: a definite pressure loss on the high-pressure side of the valve; continued flow through an inspection drain on the low-pressure side; or, in hot water or steam lines, a downstream pipe that remains hot beyond the usual length of time after valve closure. Leakage of steam through a valve which is badly steam-cut has a whistling or sonorous sound. If the valve is only slightly steam-cut, however, leakage is identified by subdued gurgling or weak popping sounds. These sounds can often be heard through a stethoscope.

Excessive vibration, noise or humming coming from within a piston-lift check or stop-check valve indicates the possibility that the disk-piston assembly is wedged inside the body. Such sticking may be caused by uneven body-guide rib wear on the downstream side. Sticking rarely occurs with tilting-disk check valves.

"Tapping," "thumping" or "rattling" noises detected from or near a check valve may indicate disk instability or cavitation. Instability could lead to rapid wear and possible valve failure. Audible cavitation is also detrimental. It may produce damage to the valve or the downstream piping. While the noise symptoms may be transmitted through the pipe from other equipment, prompt investigation is required if the check valve's performance is critical to plant reliability.

No specific inspection/preventive maintenance schedule can be given to cover all check valves. It is suggested that small valves be sampled by size and type (there may be hundreds in a large installation). Schedules for audit of larger valves should consider the criticality of the valve service. It is wise to open some critical valves for internal inspection at intervals even if no suspicious noises are detected.

Where check valves are installed close to pumps, control valves, pipe fittings or other flow disturbances, they should have more frequent inspection [see the section of this catalog entitled Adjacent Flow Disturbances (1.3.1)]. In addition, attention should be given to valves in installations with significant pipe vibration.

Users of this guide may wish to consider non-intrusive check valve monitoring methods as a supplement to periodic visual inspection and measurement of check valve internals. Noise and vibration, acoustic emission, ultrasonic and radiographic methods have been studied and demonstrated. EPRI Report No. NP 5479 provides an evaluation of the state of the art, but users are advised to obtain the most current information available on these emerging technologies.

If problems are found through any of the inspections discussed above, refer to section J: Maintenance.

2. Flow Performance

2.1 Choose the Best Valve Size for Your Service Conditions

The most economical valve is the valve correctly sized for the service flow conditions. Too small a valve will have a high pressure drop and will incur expensive energy costs in service. Too large a valve wastes money at the time of purchase, and it may require excessive effort or an excessively large and expensive actuator for operation.

Piping-system designers sometimes optimize the size of valves and piping systems to *minimize the sum of investment costs and the present value of pumping power costs*. While this may not be practical for selection of every valve, it is a goal that should be kept in mind. This catalog provides information necessary to evaluate the various types and sizes of Edward valves for stop (isolation), stop-check and check valve applications.

In the case of stop-check and check valves, another consideration is that an oversized valve may not open completely. Obviously, if a valve is not fully open, the pressure drop will be increased. Also, if the disk operates too close to the seat, unsteady flow may cause flutter that may damage valve seats, disks or guides.

System designers should also address "turndown" if service conditions involve a broad range of flow rates (e.g., high flow in normal operation but low flow during start-up and standby conditions). For these reasons, selection of check valves requires extra steps and care in calculations.

This section includes equations for the calculation of pressure drop, required flow coefficient, flow rate or inlet flow velocity. Procedures are also provided to check and correct for cavitation and flow choking. The equations in this section assume that the fluid is a liquid, a gas or steam. Two-component flow (e.g. slurries, oil-gas mixtures) is not covered by the equations. Consult Edward Valves for assistance in evaluating such applications.

Tables in this section contain performance data for all Edward stop, stop-check and check valves. Flow coefficients and cavitation/choked-flow coefficients are given for all fully open Edward valves. In addition, for check and stop-check valves, the tables provide minimum pressure drop for full lift, crack-open pressure drop, and a novel "sizing parameter" that is helpful in selecting the proper valve size for each application.

***Caution:** Pressure drop, flow rate and check valve lift estimates provided by Edward calculation methods are "best estimate" valves. Calculations are based on standard equations of the Instrument Society of America (ISA), flow rate and fluid data provided by the user, and valve flow coefficients provided by Edward Valves.*

Flow rate and fluid data are often design or best-estimate values. Actual values may differ from original estimates. Flow and check valve lift coefficients are based on laboratory testing. Valves of each specific type are tested, and results are extended to sizes not tested using model theory. This approach is fundamentally correct, but there is some uncertainty because of geometric variations between valves.

These uncertainties prevent a guarantee with respect to valve pressure drop, flow rate and lift performance, but we expect results of calculations using Edward Valves methods to be at least as accurate as comparable calculations involving flow and pressure drop of other piping system components.

2.1.1 Pressure Drop, Sizing and Flow Rate Calculations – Fully Open Valves – All Types

This section is divided into two parts. The Basic Calculations section (2.1) covers most applications where pressure drops are not excessive. This is generally the case in most Edward valve applications, and the simple equations in this section are usually sufficient for most problems.

When the pressure drop across a valve is large compared to the inlet pressure, refer to the Corrections Required with Large Pressure Drops section (2.2). Various fluid effects must be considered to avoid errors due to choked flow of steam or gas – or flashing or cavitation of liquids. While use of these more detailed calculations is not usually required, it is recommended that the simple checks in that section *always* be made to determine if correction of the results of the Basic Calculations is necessary. With experience, these checks can often be made at a glance.

2.2 Basic Calculations

The following equations apply to FULLY OPEN gate and globe valves of all types. They also apply to stop-check and check valves if the flow is sufficient to open the disk completely. *The Check Valve Sizing section (2.3) must be used to determine if a check valve is fully open and to make corrections if it is not.*

The following simple methods may be used to calculate pressure drop, required flow coefficient, flow rate or inlet flow velocity for fully open Edward valves in the majority of applications. Always check Basic Calculations against the $\Delta P/p_1$ criteria in Figure 14 to see if corrections are required. This check is automatically made when using the Proprietary Valve Sizing Computer Program available from Edward Valves.

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***Note:** In **preliminary calculations** using the following equations, a piping geometry factor, F_p = 1.0, may be used, assuming that the valve size is the same as the nominal pipe size. However, if an application involves installing a valve in a larger-sized piping system (or piping with a lower pressure rating than the valve, which will have a larger inside diameter), determine F_p from the Pipe Reducer Coefficients section when final calculations are made.*

Equations and calculations outlined in this manual are available in a proprietary Edward Valves computer program. Consult your Edward Valves sales representative for more information.

2. Flow Performance (con't.)

2.2.1 Pressure Drop

KNOWN: Flow rate (w or q)
Fluid specific gravity (G) or Density (ρ)
For water, steam or air, see Figures 22-24

FIND: Valve flow coefficient (C_V) from appropriate table

CALCULATE: Pressure drop (ΔP)

When flow rate and fluid properties are known, determine required coefficients for a specific valve and calculate the pressure drop from the appropriate equation (see Nomenclature table for definition of terms and symbols):

$$\Delta P = G\left(\frac{q}{F_P C_V}\right)^2 \quad \text{(U.S.)(1a)}$$

$$\Delta P = G\left(\frac{q}{0.865 F_P C_V}\right)^2 \quad \text{(metric)(1b)}$$

$$\Delta P = \frac{1}{\rho}\left(\frac{w}{63.3 F_P C_V}\right)^2 \quad \text{(U.S.)(1c)}$$

$$\Delta P = \frac{1}{\rho}\left(\frac{w}{27.3 F_P C_V}\right)^2 \quad \text{(metric)(1d)}$$

If the resulting pressure drop is higher than desired, try a larger valve or a different type with a higher C_V. If the pressure drop is lower than necessary for the application, a smaller and more economical valve may be tried.

2.2.2 Required Flow Coefficient

KNOWN: Flow rate (w or q)
Allowable pressure drop (ΔP)
Fluid specific gravity (G) or Density (ρ)
For water, steam or air, see Figures 22-24

CALCULATE: Minimum required valve flow coefficient (C_V)

When the flow, fluid properties and an allowable pressure drop are known, calculate the required valve flow coefficient from the appropriate equation:

$$C_V = \frac{q}{F_p}\sqrt{\frac{G}{\Delta P}} \quad \text{(U.S.)(2a)}$$

$$C_V = \frac{q}{0.865 F_p}\sqrt{\frac{G}{\Delta P}} \quad \text{(metric)(2b)}$$

$$C_V = \frac{w}{63.3 F_p \sqrt{\Delta P \rho}} \quad \text{(U.S.)(2c)}$$

$$C_V = \frac{w}{27.3 F_p \sqrt{\Delta P \rho}} \quad \text{(metric)(2d)}$$

Results of these calculations may be used to select a valve with a valve flow coefficient that meets the required flow and pressure-drop criteria. Of course, valve selection also required prior determination of the right valve type and pressure class, using other sections of this catalog. The tabulated C_V of the selected valve should then be used in the appropriate pressure drop or flow-rate equation to evaluate actual valve performance. At this stage, the checks described in section 2.2 should be made to correct for effects of large pressure drops if required.

As discussed below under flow-rate calculations, the flow-coefficient equations assume that the allowable pressure drop is available for the valve. Piping pressure drop should be addressed separately.

Caution: *In applications of stop-check or check valves, the results of these equations will apply only if the valve is fully open.* ***Always*** *use the methods given in the Check Valve Sizing section (2.3) to assure that the valve will be fully open or to make appropriate corrections.*

Nomenclature
(Metric units in parentheses)

C_V = valve flow coefficient

d = valve inlet diameter, inches (mm)

F_L = liquid pressure recovery coefficient, dimensionless

F_p = piping geometry factor, dimensionless

G = liquid specific gravity, dimensionless

G_V = gas compressibility coefficient, dimensionless

k = ratio of specific heats, dimensionless

K_i = incipient cavitation coefficient, dimensionless

ΔP = valve pressure drop, psi (bar)

ΔP_{CO} = valve crack-open pressure drop, psi (bar)

ΔP_{FL} = minimum valve pressure drop for full lift-psi (bar)

p_1 = valve inlet pressure, psia (bar, abs)

p_V = liquid vapor pressure at valve inlet temperature-psia (bar, abs)

q = volumetric flow rate, U.S. gpm (m^3/hr)

R_F = ratio of sizing parameter to sizing parameter for full lift

R_p = ratio of valve pressure drop to minimum pressure drop for full lift

R_1 = pressure drop ratio (gas or steam)

R_2 = pressure drop ratio (liquids)

SP = valve sizing parameter

SP_{FL} = valve sizing parameter for full lift

V = fluid velocity at valve inlet, ft/sec (m/sec)

w = weight flow rate-lb/hr (kg/hr)

x_T = terminal value of $\Delta P/p_1$ for choked gas or steam flow, dimensionless

Y = gas expansion factor, dimensionless

ρ = weight density of fluid at valve inlet conditions, lb/ft^3 (kg/m^3)

Conversion factors are provided in the Conversion Factors section at the end of this catalog.

2. Flow Performance (con't.)

2.2.3 Flow Rate

KNOWN: Pressure drop (ΔP)
Fluid specific gravity (G) or
Density (ρ)
For water, steam or air, see Figures 22-24

FIND: Valve flow coefficient (C_V) from appropriate table

CALCULATE: Flow rate (w or q)

When the fluid properties and an allowable pressure drop are known, determine required coefficients for a specific valve and calculate the flow rate from the appropriate equation:

$$q = F_p C_V \sqrt{\frac{\Delta P}{G}} \quad \text{(U.S.)(3a)}$$

$$q = 0.865 F_p C_V \sqrt{\frac{\Delta P}{G}} \quad \text{(metric)(3b)}$$

$$w = 63.3 F_p C_V \sqrt{\Delta P \rho} \quad \text{(U.S.)(3c)}$$

$$w = 27.3 F_p C_V \sqrt{\Delta P \rho} \quad \text{(U.S.)(3c)}$$

2.2.4 Inlet Flow Velocity

KNOWN: Flow rate (w or q)
Fluid specific gravity (G) or
Density (ρ)
For water, steam or air, see Figures 22-24

FIND: Valve inlet diameter (d) from appropriate table

CALCULATE: Fluid velocity at valve inlet (V)

While not normally required for valve sizing and selection, the fluid velocity at the valve inlet may be calculated from the appropriate equation:

$$V = \frac{0.409q}{d^2} \quad \text{(U.S.)(4a)}$$

$$V = \frac{354q}{d^2} \quad \text{(metric)(4b)}$$

$$V = \frac{0.0509w}{\rho d^2} \quad \text{(U.S.)(4c)}$$

$$V = \frac{354w}{\rho d^2} \quad \text{(metric)(4d)}$$

These valve flow-rate calculations are used less frequently than pressure drop and flow-coefficient calculations, but they are useful in some cases.

Caution: *These equations assume that the pressure drop used for the calculation is available for the valve. In many piping systems with Edward valves, flow is limited by pressure drop in pipe and fittings, so these equations should not be used as a substitute for piping calculations.*

Use of these flow-rate equations for stop-check and check valves is not recommended unless the allowable pressure drop is relatively high (e.g., over about 10 psi or 0.7 bar). At lower values of ΔP, two or more different flow rates might exist, depending on whether or not the disk is fully open. Flow would vary depending on whether the pressure drop increased or decreased to reach the specified value.

Note: *If a specific pipe inside diameter is known, that diameter may be used as the "d" value in the equation above to calculate the fluid velocity in the upstream pipe.*

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2.3 Corrections Required With Large Pressure Drops

While most Edward valves are used in relatively high-pressure systems and are usually sized to produce low pressure drop at normal flow rates, care is necessary to avoid errors (which may be serious in some cases) due to flow "choking" (or near-choking). Problems arise most often at off-design flow conditions that exist only during plant start-up, shutdown, or standby operation.

Since steam and gas are compressible fluids, choking (or near-choking) may occur due to fluid expansion which causes the fluid velocity to approach or reach the speed of sound in reduced-area regions. While liquids are normally considered to be incompressible fluids, choking may also occur with liquid flow due to cavitation or flashing. In each case, simple calculations can be made to determine if a problem exists. Relatively simple calculations are required to correct for these effects. In some cases, these calculations may require a change in the size of type of valve required for a specific application.

The flow parameters K_i, F_L and x_T in the valve data tables assume that the valve is installed in pipe of the same nominal size. This is a fairly good assumption for preliminary calculations, but refer to the Pipe Reducer Coefficients section if there is a mismatch between valve and pipe diameters (also see instructions relative to F_p calculations in section 2.1) and make the appropriate corrections when final calculations are made.

Note: Because large pressure drop problems are not encountered frequently, equations are presented in terms of weight flow rate (w) and density (ρ) only. See the Conversion of Measurement Units section for converting other units of flow rate to weight flow rate.

2.3.1 Gas and Steam Flow

2.3.1.1 Pressure Drop – To determine if corrections are needed for compressible flow effects, use the data from the Basic Calculations to determine the ratio of the calculated pressure drop to the absolute upstream pressure:

$$R_1 = \frac{\Delta P}{p_1} \qquad (5)$$

If the ratio R_1 is less than the values in Figure 14, the results of the Basic Calculations will usually be sufficiently accurate, and further calculations are unnecessary.

Figure 14

MAXIMUM $\Delta P/p_1$ FOR USE OF BASIC CALCULATIONS WITHOUT CORRECTION	
Valve Type	Max. $\Delta P/p_1$
Gate	0.01
Inclined-Bonnet Globe Angle Tilting-Disk Check	0.02
90°-Bonnet Globe	0.05

If the pressure-drop ratio R_1 exceeds that tabulated for the valve type under evaluation, the procedure described below should be used to check and correct for possible flow choking or near-choking.

(1) Calculate the gas compressibility coefficient:

$$G_y = \frac{0.467}{kx_T}\left(\frac{\Delta P}{p_1}\right) \qquad \text{(U.S. or metric)(6)}$$

Note: *The ΔP in this equation is the uncorrected value from the Basic Calculations. Values of x_T are given in valve data tables, and values of k are given in Figure 21.*

(2) The next step depends on the value of G_y determined in equation (6):

- If $G_y < 0.148$, *the flow is not fully choked.* Read the value of Y from Figure 15 and calculate the corrected pressure drop:

$$\Delta P_C = \frac{\Delta P}{Y^2} \qquad \text{(U.S. or metric)(7)}$$

- If $G_y \geq 0.148$, *the flow is choked.* The desired flow cannot be achieved at the specified upstream pressure and will be limited to the choked flow rate given by:

$$w_{choked} = 35.67\ F_p\ C_V \sqrt{kx_T p_1 \rho} \qquad \text{(U.S.)(8a)}$$

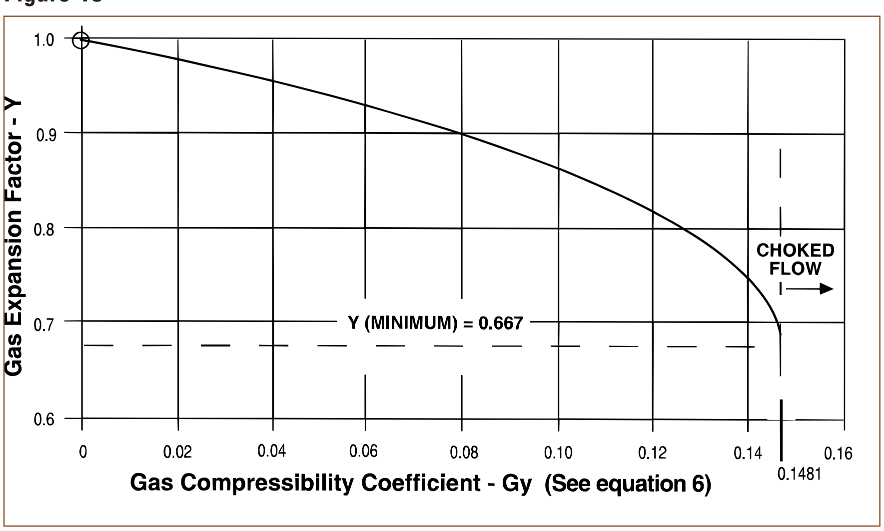
$$w_{choked} = 15.4\ F_p\ C_V \sqrt{kx_T p_1 \rho} \qquad \text{(metric)(8b)}$$

- When flow is choked, the actual pressure drop cannot be calculated using valve flow calculations alone. It can be any valve greater than the following minimum value for choked flow:

$$\Delta P_{min.\ choked} \geq 0.714 kx_T p_1 \qquad \text{(U.S. or metric)(9)}$$

- The only way to determine the pressure downstream of a valve with choked flow is to calculate the pressure required to force the choked flow rate through the downstream piping. This may be done with piping calculations (not covered by this catalog).

Figure 15



2.3 Corrections Required With Large Pressure Drops (con't.)

2.3.1.2 Flow Rate – When calculating the flow rate through a valve, the actual pressure drop is known, but the flow may be reduced by choking or near-choking.

To check for high pressure-drop effects, calculate R_1, the ratio of pressure drop to absolute upstream pressure (see equation 5 above) noting that the pressure drop in this case is the known value.

(1) Flow rates determined using the Basic Calculations are sufficiently accurate if R_1 is less that about *twice* the value tabulated in Figure 14 for the applicable valve type (higher because actual pressure drop is used in the ratio). In this case, no correction is necessary.

(2) When corrections for higher values of R_1 are required, calculate the gas expansion factor directly from:

$$Y = 1 - 0.467\left(\frac{\Delta P/p_1}{kx_T}\right) \quad \text{(U.S. or metric)(10)}$$

(3) The calculation method to determine the flow rate depends on the calculated value of Y from equation (10):

• If Y is greater than 0.667 (but less that 1), the flow is not fully choked. Calculate the corrected flow rate as follows:

$$w_C = Yw \quad \text{(U.S. or metric)(11)}$$

• If Y is equal to or less than 0.667, the valve flow is choked, and the results of the Basic Calculations are invalid. The actual flow rate may be calculated from the equation for w_{choked} [(8a) or (8b)] above.

> ***Caution:*** *Choked or near-choked flow conditions may produce significant flow-induced noise and vibration. Prolonged operation with flow rates in this region may also cause erosion damage within a valve or in downstream piping, particularly if the flow condition involve "wet" steam. Edward valves tolerate these conditions well in services involving limited time periods during plant start-up, shutdown, etc., but consult Edward Valves about applications involving long exposure to such conditions.*

2.3.2 Liquid Flow – Cavitation and Flashing

The fluid pressure in high-velocity regions within a valve may be much lower than either the upstream pressure of the downstream pressure. If the pressure within a valve falls below the vapor pressure (p_v) of the liquid, vapor bubbles or cavities may form in the flow stream. Cavitation, flashing and choking may occur. Use the equations and procedures in this section to evaluate these phenomena.

Cavitation and flashing are closely related, and they may be evaluated by calculating a pressure-drop ratio that is slightly different from that used for gas or steam:

$$R_1 = \frac{\Delta P}{(p_1 - p_v)} \quad (12)$$

To evaluate a particular valve and application, find values of K_i and F_L from the appropriate valve-data table, find p_v values for common liquids given in Figure 25, calculate R_2, and perform the following checks:

(1) Cavitation – the sudden and sometimes violent coalescence of the cavities back to the liquid state – occurs when the downstream pressure (within the valve or in the downstream pipe) recovers to above the vapor pressure.

• If $R_2 < K_i$, there should be no significant cavitation or effect on flow or pressure drop. Results of the Basic Calculations require no correction.

• If $R_2 > K_i$, cavitation begins. If the ratio is only slightly greater than K_i, it may be detected as an intermittent "ticking" noise near the valve outlet, although pipe insulation may muffle this sound. This stage of cavitation is usually related to tiny vapor cavities that form near the center of vortices in the flow stream, and it generally produces neither damage nor effects on flow characteristics. However, as the pressure drop ratio R_2 increases, the noise progresses to a "shh," then a "roar."

• If $R_2 > (K_i + F_L^2)/2$, approximately, larger vapor cavities develop and the risk of cavitation damage (pitting) in the valve or downstream pipe may be a concern if this flow condition is sustained for significant periods of time. *Noise* may also pose a problem. Still, at this stage, there is usually no significant effect on valve flow characteristics. Results of the Basic Calculations require no correction.

As the pressure-drop ration increases beyond this point, some valves suffer slight reductions in their C_V values, but there is no practical way of correcting pressure drop or flow calculations for this effect. Vibration and noise increase, ultimately sounding like "rocks and gravel" bouncing in the pipe at about the point where flow becomes choked.

(2) Flashing – the persistence of vapor cavities downstream of the valve – occurs when the pressure downstream of the valve remains below the vapor pressure.

• If $R_2 > 1$, *flashing* occurs, and the flow is choked due to vapor cavities in the flow stream.

(3) Liquid choking – A slightly different ratio may be used to predict the minimum pressure drop at choked flow conditions. Choking occurs due to vapor cavities near the minimum-area region in the flow stream when:

$$\frac{\Delta P}{(p_1 - 0.7p_v)} \geq F_L^2 \quad (13)$$

Thus, the minimum pressure drop which will produce choked liquid flow is given by:

$$\Delta P \geq F_L^2(p_1 - 0.7p_V) \quad (14)$$

Note that flow may be choked by either severe cavitation or flashing.

2.3.2.1 Predicting Choked Flow Rate – If the result of a Basic Calculation to determine *pressure drop* exceeds the value determined from equation (13), the Basic Calculation is invalid. The flow used for input cannot be obtained at the specified upstream pressure and temperature. In such a case, or if it is necessary to calculate liquid flow rate through a valve with high pressure drop, the choked flow rate at specified conditions may be calculated from:

$$w_{choked} = 63.3F_pC_VF_L\sqrt{\rho(p_1 - 0.7p_V)} \quad \text{(U.S.)(15a)}$$

$$w_{choked} = 27.3F_pC_VF_L\sqrt{\rho(p_1 - 0.7p_V)} \quad \text{(metric)(15b)}$$

When flow is choked due to either cavitating or flashing flow, the actual pressure drop cannot be determined from valve calculations. It may be any value greater than the minimum value for choked flow [equation (14)]. As in the case of choked gas or steam flow, the pressure downstream of a valve must be determined by calculating the pressure required to force the choked flow through the downstream piping. This may be done with piping calculations (not covered by this catalog).

• If the pressure drop from a Basic Calculation was used to determine *flow rate*, and the pressure drop exceeds the pressure drop of choked flow, the result is invalid. The corrected flow rate may be calculated from equation (15a) or (15b) above.

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2.4 Check Valve Sizing

The most important difference between check (including stop-check) valves and stop valves, from a flow performance standpoint, is that the check valve disk is opened only by dynamic forces due to fluid flow. The preceding calculation methods for flow and pressure drop are valid only if it can be shown that the valve is fully open.

The primary purpose of this section is to provide methods to predict check valve disk opening and to make corrections to pressure-drop calculations if the valve is not fully open. These methods are particularly applicable to sizing valves for new installations, but they are also useful for evaluation of performance of existing valves.

In selecting a stop-check or check valve for a new installation, the first steps require selecting a proper type and pressure class. The Stop and Check Valve Applications Guide section of this catalog should be reviewed carefully when the type is selected, noting advantages and disadvantages of each type and considering how they relate to the requirements of the installation. Other sections of this catalog provide pressure ratings to permit selection of the required pressure class.

2.4.1 Sizing Parameter

The first step in evaluating a stop-check or check valve application is to determine the Sizing Parameter based on the system flow rate and fluid properties:

$$SP = \frac{w}{\sqrt{\rho}} \qquad \text{(U.S. or metric)(16)}$$

Tables in this section provide a Sizing Parameter for full lift (Sp_{FL}) for each Edward stop-check and check valve. The amount of opening of any check valve and its effect on pressure drop can be checked simply as follows:

• *If $Sp_{FL} < SP$, the valve is fully open.* Pressure drop may be calculated using the equations given previously for fully open valves (including corrections for large pressure drops if required).

• *IF $SP_{FL} > SP$, the valve is not fully open.* A smaller size valve or another type should be selected if possible to assure full opening. If that is not feasible, three additional steps are required to evaluate the opening and pressure drop of the valve under the specified service conditions.

> ***Note:*** *EPRI Report No. NP 5479 (Application Guideline 2.1) uses a "C" factor to calculate the minimum flow velocity required to fully open a check valve. The sizing procedures in this catalog do not employ the "C" factor, but values are given in the valve data tables for readers who prefer to use the EPRI methods. Since the EPRI methods are based on velocity, a flow area is required as a basis. Valve Inlet Diameters presented in data tables are the basis for correlation between flow rate and velocity.*

2.4.2 Calculations for Check Valves Less Than Fully Open

If the preceding evaluation revealed an incompletely open check valve, perform the following additional calculations:

Calculate the flow-rate ratio:

$$R_F = \frac{SP}{SP_{FL}} \qquad \text{(U.S. or metric)(17)}$$

Determine the disk operating position:

Using the R_F value calculated above, determine the valve operating position from Figure 16 (forged-steel valves) or Figures 17-20 (cast-steel valves). Performance curve numbers for individual cast-steel stop-check and check valves are given in the tabulations with other coefficients. Evaluate the acceptability of the operating position based on recommendations in the Check Valve Applications Guide and in the specific sizing guidelines below.

Calculate the pressure drop:

Again using the R_F value calculated above, determine the pressure drop ratio R_P from Figures 16-20, and calculate the valve pressure drop at the partially open position:

$$\Delta P = R_P \Delta P_{FL} \qquad \text{(U.S. or metric)(18)}$$

Values for ΔP_{FL} for all stop-check and check valves are given in Valve Tables 1 to 5 and 10 to 15 with other coefficients.

> ***Note:*** *The values of the various valve coefficients given in the tabulations are based on testing of a substantial number of valves. Most are applicable to any line fluid, but those involving check valve lift are influenced by buoyancy. Tabulated values are based on reference test conditions with room-temperature water. SP_{FL} and ΔP_{FL} are slightly higher in applications involving lower-density line fluids. Considering the expected accuracy of these calculations, the following corrections may be considered:*
>
> • *For water at any temperature and other common liquids – No correction required.*
>
> • *For steam, air and other common gases at normal operating pressures and temperatures – Increase SP_{FL} by 7% and increase ΔP_{FL} by 14%.*

2.4.3 Sizing Guidelines

Considering the recommendations in the Check Valve Applications Guide section of this catalog and the calculation methods described above, the following specific steps are recommended for sizing check valves for optimum performance and service life (it is assumed that the check valve type and pressure class have already been selected before starting this procedure):

(1) Constant flow rate – If the application involves a substantially *constant flow rate* during all operating conditions, the check valve should be sized to be fully open. This may be accomplished by the following procedure:

• Calculate the check valve sizing parameter (SP) for the application from equation (15). Values of density for water, steam, and air are available in Figures 22-24.

If the flow rate is not given in lb/hr (or kg/hr), refer to the Conversion of Measurement Units section of this catalog to make the necessary calculation.

• Select the valve size with *the next smaller* SP_{FL} value from valve data tables (Tables 1-5 for forged-steel valves and Tables 10-15 for cast-steel valves). Make note of the C_V, ΔP_{CO}, ΔP_{FL}, K_i, F_L and x_T values for use in later calculations.

> ***Note:*** *Preferably, there should be a good margin between SP and SP_{FL} to be sure the valve will be fully open. In the specific case of tilting-disk check valves, it is recommended that SP_{FL} be less than 0.83 (SP) to be sure that the disk is fully loaded against its stop (particularly if it is close to a flow disturbance).*

• Calculate the pressure drop using the Basic Calculation method in equation (1) and the Cxx value of the valve size selected above. Make the simple checks described above in section 2.2 (Corrections Required With Large Pressure Drops), and make appropriate corrections in necessary (this is rarely needed for a valve sized for constant flow rate, but the check is desirable).

• Evaluate the pressure drop. If it is too high, a larger size or another check valve type should be tried. If it is lower than necessary for the application, a smaller and more economical valve (with a lower SP_{FL}) may be evaluated with assurance that it would also be fully open.

2.4 Check Valve Sizing

• Evaluate the crack-open pressure drop (ΔP_{CO}) to be certain that the system head available at the initiation of flow will initiate valve opening. Note that, for some valves, the crack-open pressure drop exceeds the pressure drop for full lift. Preceding calculations might indicate no problem, but it is possible that a valve might not open at all in a low-head application (e.g., gravity flow).

(2) Variable flow rate – If the application involves check valve operation over a *range of flow rates*, additional calculations are necessary to assure satisfactory, stable performance at the lowest flow rate without causing excessive pressure drop at the maximum flow condition. This required careful evaluation of specific system operating conditions (e.g., are the minimum and maximum flow rates *normal* operating conditions or *infrequent* conditions that occur only during start-up or emergency conditions?).

The following options should be considered in selecting the best stop-check or check valve size for variable flow applications:

• The best method, if practical, is to size the valve to be fully open at the minimum flow condition. This may be done by following the first two steps listed above for the constant flow-rate case, but using the *minimum flow rate* in the sizing parameter (SP) calculation.

The only difference is that the pressure-drop calculations and evaluations in the third and fourth steps must be repeated at *normal and maximum flow rates*. If the selected valve size is fully open at the minimum flow rate and has an acceptable pressure drop at the maximum flow condition, it should give good overall performance.

• Sometimes a change in valve type provides the best cost-effective solution for variable-flow applications (e.g. use a smaller Flite-Flow® stop-check or check valve instead of a 90°-bonnet type to provide full lift at the minimum flow condition, but a high C_V for low pressure drop at maximum flow).

• Operation at less than full lift may have to be considered.

(3) Operation at less than full lift – "High Turndown" applications sometimes exist on boilers and other process systems that must swing through periodic flow changes from start-up, to standby, to maximum, and back again. In such cases, calculations may not reveal any single valve that will offer a satisfactory compromise assuring full lift and an acceptable pressure drop at both minimum and maximum flow conditions.

It may be acceptable to permit a check valve to operate at less than fully open at the minimum flow condition if such operation is infrequent or not expected to be sustained continuously for long periods. A valve may be sized by following the methods above using the *lowest expected normal sustained flow rate* in the sizing parameter (SP) calculation. Pressure drop at normal and maximum flow rates should then be calculated and evaluated.

The acceptability of valve operation at the *minimum* flow condition should be evaluated as follows:

• Calculate the sizing parameter (SP) at the minimum flow rate and the flow-rate ratio R_F from equation (17). The valve operating position (% open) should be determined from the proper performance curve (Figures 16-20).

> ***Caution:*** *Check valve operation at less than 25% opening is not recommended. Any check valve that operates for sustained periods at partial openings should be monitored or inspected periodically for evidence of instability or wear.*

• If the minimum operating position is considered satisfactory, the pressure drop at the minimum flow condition may be calculated from equation (18), using the pressure-drop ratio (R_p) determined from the proper performance curve.

(4) Alternatives for high turndown applications – If the preceding steps show that the range of flow rates is too large for any single standard check valve, consult Edward Valves. Several alternatives may be considered:

• Either 90°-bonnet or angle-type stop-check or piston-lift check valves may be furnished with a special disk with an extended "skirt" as illustrated in Figure 15A. This skirt increases flow resistance at low flow rates, producing additional lifting force to help prevent operation at small openings.

Of course, the skirt also reduces the C_V of the valve somewhat when it is fully open and increases pressure drop at maximum flow. Nevertheless, a special disk sometimes solves difficult high turndown problems. A special disk also permits solution of some problems with existing valves that are "oversized."

• A *stop-check valve* may be used with the stem lifted just enough to provide a positive stop for the disk at very low flows (e.g., short-term start-up conditions). The stem should be lifted with increasing flow rate to maintain the disk-stopping action while preventing excessive pressure drop. At normal flow rates, the stem can be lifted to its fully open position, permitting normal check valve function. The stem may be actuated manually for infrequent start-up operations, or a motor actuator may be furnished for convenience if large flow rate variations are expected to be frequent.

> ***Caution:*** *This arrangement could produce cavitation or flow-choking problems if the flow rate is increased substantially without lifting the valve stem to compensate.*

• A small check or stop-check valve may be installed *in parallel* with a larger stop-check valve. The smaller valve may be sized for the minimum flow condition, and the larger stop-check may be held closed with the stem until the flow is sufficient to assure adequate lift. If necessary, the stem on the larger valve may be opened gradually with increasing flow to maintain disk-stopping action as in the example above. The smaller valve may be allowed to remain open at higher flow rates or, if a stop-check type is used, it may be closed if preferred. Either or both valves may be manually actuated or furnished with a motor actuator for convenience.

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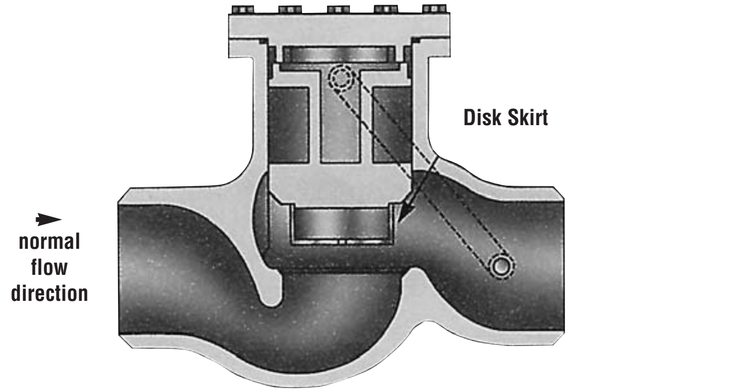

Figure 15A

2.5 Pipe Reducer Coefficients

The equations in the Flow Performance section of this catalog use a piping geometry factor, F_p, to account for the effect of pipe reducers attached directly to the valve. This permits the valve and pipe reducers to be treated as an assembly, i.e., F_pC_V is the flow coefficient of the valve/pipe reducer combination. Then, the pressure drop in the flow equations is the pressure drop of the assembly.

This method is also applicable when valves are furnished with oversized ends to fit larger diameter pipe. It should also be used to evaluate line-size valves used in pipe with a lower pressure rating than the valve, because such pipe may have less wall thickness and a larger inside diameter than the valve inlet diameter given in the valve data tabulations.

This section provides equations for calculation of the piping geometry factor, F_p, which should be used even in Basic Calculations when there is a significant difference between the pipe diameter and valve inlet diameter (d).

In addition, other coefficients (K_1, F_L, x_T) are affected by the presence of pipe reducers. Equations are also provided for correction of these terms, which are required only when evaluating significant valve-to-pipe diameter mismatch.

Note: These equations apply only where the valve diameter is less than the connecting pipe diameter.

2.5.1 Pipe Geometry Factor

Calculate upstream loss coefficient:

$$K_1 = 0.5\left[1-\left(\frac{d}{D_1}\right)^2\right]^2 \quad \text{(U.S. or metric)(1-1)}$$

Calculate downstream loss coefficient:

$$K_2 = \left[1-\left(\frac{d}{D_2}\right)^2\right]^2 \quad \text{(U.S. or metric)(1-2)}$$

Summation:

$$\Sigma K = K_1 + K_2 \quad \text{(U.S. or metric)(1-3)}$$

$$F_p = \sqrt{\frac{1}{1+\frac{\Sigma K}{890}\left(\frac{C_V}{d^2}\right)^2}} \quad \text{(U.S.)(1-4a)}$$

$$F_p = \sqrt{\frac{1}{1+468\Sigma K\left(\frac{C_V}{d^2}\right)^2}} \quad \text{(metric)(1-4b)}$$

Note: If D_1 and D_2 are not the same, use of F_p calculated in this manner accounts for energy losses associated with flow contraction and expansion, and the pressure drop calculated using this factor represents energy loss. Bernoulli effects may cause a different static pressure change between upstream and downstream pipes.

2.5.2 Other Coefficients

Correction of values of K_1, F_L and x_T requires an initial calculation of a Bernoulli coefficient to account for static pressure change in the inlet reducer:

$$K_{B1} = 1-\left(\frac{d}{D_1}\right)^4 \quad \text{(U.S. or metric)(1-5)}$$

Then, corrected values of each coefficient may be calculated, using the corresponding value from valve data tables as input:

$$K_{ii} = \frac{1}{F_p^2\left[\frac{1}{K_i}+\left(\frac{K_1+K_{B1}}{890}\right)\left(\frac{C_V}{d^2}\right)^2\right]} \quad \text{(U.S.)(1-6a)}$$

$$K_{ii} = \frac{1}{F_p^2\left[\frac{1}{K_i}+468(K_1+K_{B1})\left(\frac{C_V}{d^2}\right)^2\right]} \quad \text{(metric)(1-6b)}$$

$$F_{LL} = \frac{1}{F_p\sqrt{\frac{1}{F_L^2}+\left(\frac{K_1+K_{B1}}{890}\right)\left(\frac{C_V}{d^2}\right)^2}} \quad \text{(U.S.)(1-7a)}$$

$$F_{LL} = \frac{1}{F_p\sqrt{\frac{1}{F_L^2}+468(K_1+K_{B1})\left(\frac{C_V}{d^2}\right)^2}} \quad \text{(metric)(1-7b)}$$

$$x_{TT} = \frac{x_T}{F_p^2\left[1+\frac{x_T(K_1+K_{B1})}{1000}\left(\frac{C_V}{d^2}\right)^2\right]} \quad \text{(U.S.)(1-8a)}$$

$$x_{TT} = \frac{x_T}{F_p^2\left[1+416x_T(K_1+K_{B1})\left(\frac{C_V}{d^2}\right)^2\right]} \quad \text{(metric)(1-8b)}$$

where: K_i, F_L and x_T are values from valve data tables; K_{ii}, F_{LL} and x_{TT} are corrected values for valve/reducer assembly.

Nomenclature

C_V = valve flow coefficient. See Valve Reference Data.

d = valve-end inside diameter, inches, (mm). See Valve Reference Data.

D_1 = inside diameter of upstream pipe, inches, (mm). See Pipe Data Section.

D_2 = inside diameter of downstream pipe, inches, (mm). See Pipe Data Section.

F_L = liquid-pressure recovery coefficient, dimensionless*

F_p = piping-geometry factor, dimensionless

K_1 = pressure-loss coefficient for inlet reducer, dimensionless

K_2 = pressure-loss coefficient for outlet reducer, dimensionless

K_{B1} = pressure change (Bernoulli) coefficient for inlet reducer, dimensionless

ΣK = $K_1 + K_2$, dimensionless

K_i = incipient-cavitation coefficient, dimensionless*

x_T = terminal value of $\Delta P/p_1$ for choked gas or steam flow, dimensionless

***** Double subscripts (e.g., K_{ii}) represent values corrected for effects of pipe reducers.***

Table 1
Forged Steel Angle Univalve® Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

Class 1690 (PN 290) *All Stop valves, all Stop-Check valves, all Piston Check valves*

SIZE		ALL STOP & CHECK VALVES						CHECK VALVES* WITH SPRINGS (STD)					CHECK VALVES* WITHOUT SPRINGS				
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}		SP_{FL}		C	ΔP_{FL}		SP_{FL}		C
0.50	15	10.5	0.80	0.41	0.16	0.68	17.3	6.0	0.41	887	101	179	1.5	0.103	468	53	165
0.75	20	10.5				0.68	17.3			1522	172	179			804	91	165
1.00	25	10.5				0.68	17.3			1522	172	179			804	91	165
1.25	32	31				1.19	30.2			5326	604	179			2810	318	164
1.50	40	31				1.19	30.2			5066	574	179			2670	303	164
2.00	50	50				1.50	38.1			8620	977	180			4550	516	166
2.50	65	90				2.00	50.8			13,916	1580	179			7360	834	165
3.00	80	90				2.00	50.8			12,715	1440	179			6690	758	165

Class 2680 (PN 460) *All Stop valves, all Stop-Check valves, all Piston Check valves*

NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}		SP_{FL}		C	ΔP_{FL}		SP_{FL}		C
0.50	15	10.5	0.80	0.41	0.16	0.68	17.3	6.0	0.41	729	83	179	1.5	0.103	385	44	165
0.75	20	10.5				0.68	17.3			625	71	179			330	37	165
1.00	25	10.5				0.68	17.3			1140	129	179			604	68	165
1.25	32	19				0.94	23.9			3120	354	177			1650	187	163
1.50	40	19				0.94	23.9			2910	330	177			1540	175	163
2.00	50	50				1.50	38.1			7290	826	180			3850	436	166
2.50	65	89				2.00	50.8			10,400	1180	179			5490	622	165
3.00	80	89				2.00	50.8			10,400	1180	179			5490	622	165

NOTES: *See Table 9 for DP_{CO}.*
See notes following paragraph 2.4.1, page G-28, for discussion of C factor.
** Stop-check valves are only furnished without springs.*

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Table 1A
Forged Steel Univalve® Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

SIZE		ALL STOP & CHECK VALVES						CHECK VALVES* WITH SPRINGS (STD)					CHECK VALVES* WITHOUT SPRINGS				
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}		SP_{FL}		C	ΔP_{FL}		SP_{FL}		C
Class 1690 (PN 290) *All Stop valves, all Stop-Check valves, all Piston Check valves*																	
0.50	15	**7.0**				**0.464**	11.8			**886**	100	**210**			**443**	50.2	**105**
0.75	20	**12**				**0.612**	15.5			**1520**	172	**207**			**760**	86.0	**103**
1.00	25	**12**				**0.815**	20.7			**1520**	172	**117**			**760**	86.0	**58**
1.25	32	**42**				**1.160**	29.5			**5320**	602	**201**			**2660**	301	**101**
1.50	40	**40**	**0.66**	**0.27**	**0.16**	**1.338**	34.0	**4.0**	0.28	**5060**	574	**144**	**1.0**	0.069	**2530**	287	**72**
2.00	50	**68**				**1.687**	42.8			**8610**	975	**154**			**4300**	488	**77**
2.50	65	**110**				**2.125**	54.0			**13,900**	1580	**157**			**6960**	789	**79**
3.00	80	**100**				**2.624**	66.6			**12,700**	1430	**94**			**6330**	717	**47**
4.00	100	**85**				**3.438**	87.3			**10,800**	1220	**46**			**5380**	609	**23**
Class 2680 (PN 460) *All Stop valves, all Stop-Check valves, all Piston Check valves*																	
0.50	15	**7.0**				**0.464**	11.8			**886**	100	**210**			**443**	50.2	**105**
0.75	20	**12**				**0.612**	15.5			**760**	86.0	**103**			**380**	43.0	**52**
1.00	25	**11**				**0.599**	15.2			**1390**	158	**198**			**696**	78.9	**99**
1.25	32	**30**				**0.896**	22.8			**3800**	430	**241**			**1900**	215	**121**
1.50	40	**28**	**0.63**	**0.24**	**0.15**	**1.100**	28.0	**4.0**	0.28	**3540**	401	**149**	**1.0**	0.069	**1770**	201	**75**
2.00	50	**70**				**1.503**	38.2			**8860**	1000	**200**			**4430**	502	**100**
2.50	65	**100**				**1.771**	45.0			**12,700**	1430	**206**			**6330**	717	**103**
3.00	80	**100**				**2.300**	58.4			**12,700**	1430	**122**			**6330**	717	**61**
4.00	100	**90**				**3.152**	80.1			**11,400**	1290	**58**			**5700**	645	**29**
Class 4500 (PN760) *All Stop valves, all Stop-Check valves, all Piston Check valves*																	
0.50	15	**2.0**				**0.252**	6.4			**253**	28.7	**203**			**127**	14.3	**102**
0.75	20	**6.0**				**0.434**	11.0			**760**	86.0	**205**			**380**	43.0	**103**
1.00	25	**12**				**0.599**	15.2			**1520**	172	**216**			**760**	86.0	**108**
1.25	32	**12**				**0.808**	20.5			**1520**	172	**117**			**760**	86.0	**59**
1.50	40	**11**	**0.64**	**0.25**	**0.15**	**0.926**	23.5	**4.0**	0.28	**1390**	158	**82**	**1.0**	0.069	**696**	78.9	**41**
2.00	50	**48**				**1.156**	29.4			**6080**	688	**230**			**3040**	344	**115**
2.50	65	**62**				**1.400**	35.6			**7850**	889	**202**			**3920**	444	**101**
3.00	80	**60**				**1.700**	43.2			**7600**	860	**132**			**3800**	430	**66**
4.00	100	**55**				**2.200**	55.9			**6960**	789	**76**			**3480**	394	**37**

NOTES: ***See Table 9 for DP_{CO}.***
See notes following paragraph 2.4.1, page G-28, for discussion of C factor.
**** Stop-check valves are only furnished without springs.***

Table 1B
Forged Steel PressurCombo Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

CLASS 1690 (PN 290) *36124, 36128, 36224, 36228*

SIZE		PRESSURSEAT (DS)						PRESSUREATER (DE)						PRESSURCOMBO (DC)					
NPS	DN	C_V	F_L	x_T	K_i	d		C_V	F_L	x_T	K_i	d		C_V	F_L	x_T	K_i	d	
0.50	15	**5.0**				**0.464**	11.8	**5.0**				**0.464**	11.8	**4.1**				**0.464**	11.8
0.75	20	**6.1**				**0.612**	15.5	**5.9**				**0.612**	15.5	**4.5**				**0.612**	15.5
1.00	25	**6.1**				**0.815**	20.7	**5.6**				**0.815**	20.7	**4.4**				**0.815**	20.7
1.25	32	**12**				**1.160**	29.55	**11**				**1.160**	29.5	**8.0**				**1.160**	29.5
1.50	40	**12**	**.85**	**.50**	**.27**	**1.338**	34.0	**11**	**.80**	**.45**	**.24**	**1.338**	34.0	**8.0**	**.80**	**.45**	**.24**	**1.338**	34.0
2.00	50	**30**				**1.687**	42.3	**28**				**1.687**	42.8	**22**				**1.687**	42.8
2.50	65	**53**				**2.125**	54.0	**51**				**2.125**	54.0	**39**				**2.125**	54.0
3.00	80	**51**				**2.624**	66.6	**47**				**2.624**	66.6	**37**				**2.624**	66.6
4.00	100	**49**				**3.438**	87.3	**43**				**3.438**	87.3	**35**				**3.438**	87.3

Class 2680 (PN 460) *66124, 66128, 66224, 66228*

NPS	DN	C_V	F_L	x_T	K_i	d		C_V	F_L	x_T	K_i	d		C_V	F_L	x_T	K_i	d	
0.50	15	**5.0**				**0.464**	11.8	**5.0**				**0.464**	5.0	**4.1**				**0.464**	11.8
0.75	20	**4.6**				**0.612**	15.5	**4.5**				**0.612**	4.5	**3.8**				**0.612**	15.5
1.00	25	**6.0**				**0.599**	15.2	**5.7**				**0.599**	5.7	**4.5**				**0.599**	15.2
1.25	32	**12**				**0.896**	22.8	**12**				**0.896**	12	**8.9**				**0.896**	22.8
1.50	40	**12**	**.85**	**.50**	**.27**	**1.100**	28.0	**11**	**.80**	**.45**	**.24**	**1.100**	11	**8.3**	**.80**	**.45**	**.24**	**1.100**	27.9
2.00	50	**31**				**1.502**	38.2	**30**				**1.502**	30	**23**				**1.502**	38.2
2.50	65	**52**				**1.771**	45.0	**56**				**1.771**	56	**41**				**1.771**	45.0
3.00	80	**52**				**2.300**	58.4	**48**				**2.300**	48	**38**				**2.300**	58.4
4.00	100	**50**				**3.152**	80.1	**44**				**3.152**	44	**36**				**3.152**	80.1

Class 4500 (PN 760) *96124, 96128, 96224, 96228*

NPS	DN	C_V	F_L	x_T	K_i	d		C_V	F_L	x_T	K_i	d		C_V	F_L	x_T	K_i	d	
0.50	15	**1.9**				**0.252**	6.4	**1.5**				**0.252**	6.4	**1.4**				**0.252**	6.4
0.75	20	**4.6**				**0.434**	11.0	**4.4**				**0.434**	11.0	**3.8**				**0.434**	11.0
1.00	25	**6.1**				**0.599**	15.2	**5.8**				**0.599**	15.2	**4.5**				**0.599**	15.2
1.25	32	**6.1**				**0.808**	20.5	**5.6**				**0.808**	20.5	**4.4**				**0.808**	20.5
1.50	40	**5.9**	**.85**	**.50**	**.27**	**0.926**	23.5	**5.3**	**.80**	**.45**	**.24**	**0.926**	23.5	**4.3**	**.80**	**.45**	**.24**	**0.926**	23.5
2.00	50	**28**				**1.156**	29.4	**29**				**1.156**	29.4	**22**				**1.158**	29.4
2.50	65	**30**				**1.400**	35.6	**30**				**1.400**	35.6	**23**				**1.400**	35.6
3.00	80	**30**				**1.700**	43.2	**28**				**1.700**	43.2	**22**				**1.700**	43.2
4.00	100	**29**				**2.200**	55.9	**25**				**2.200**	55.9	**21**				**2.200**	55.9

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Table 2
Forged Steel Inclined Bonnet Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

Class 800 (PN 130) *Figure No. 848/848Y Stop valve, 868/868Y Stop-Check valve, 838/838Y Piston Check valve*

SIZE		ALL STOP & CHECK VALVES						CHECK VALVES* WITH SPRINGS (STD)					CHECK VALVES* WITHOUT SPRINGS				
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}		SP_{FL}		C	ΔP_{FL}		SP_{FL}		C
0.25	8	**1.4**				**0.364**	9.2			**198**	22.4	**76**			**68.6**	7.77	**26**
0.38	10	**3.3**				**0.493**	12.5			**467**	52.9	**98**			**162**	18.3	**34**
0.50	15	**3.3**				**0.546**	13.9			**467**	52.9	**80**			**162**	18.3	**28**
0.75	20	**5.7**	**0.72**	**0.30**	**0.20**	**0.742**	18.8	**5.0**	0.34	**722**	81.8	**67**	**0.6**	0.041	**250**	28.3	**23**
1.00	25	**13.5**				**0.957**	24.3			**1910**	216	**106**			**662**	75.0	**37**
1.25	32	**23.5**				**1.278**	32.5			**3330**	377	**104**			**1150**	131	**36**
1.50	40	**37.5**				**1.500**	38.1			**5290**	600	**120**			**1830**	208	**42**
2.00	50	**48.5**				**1.939**	49.3			**6860**	778	**93**			**2380**	269	**32**

Series 1500 *Figure No. 1048/1048Y Stop valve, 1068/1068Y Stop-Check valve, 1038/1038Y Piston Check valve*

NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}		SP_{FL}		C	ΔP_{FL}		SP_{FL}		C
0.25	8	**1.7**				**0.302**	7.7			**241**	27.3	**134**			**83.4**	9.45	**47**
0.38	10	**3.9**				**0.423**	10.7			**552**	62.5	**157**			**191**	21.7	**54**
0.50	15	**3.8**				**0.464**	11.8			**538**	60.9	**127**			**186**	21.1	**44**
0.75	20	**6.8**	**0.75**	**0.34**	**0.20**	**0.612**	15.5	**5.0**	0.34	**963**	109	**131**	**0.6**	0.041	**333**	37.8	**45**
1.00	25	**10.5**				**0.815**	20.7			**1490**	168	**114**			**515**	58.3	**39**
1.25	32	**28**				**1.160**	29.5			**3960**	449	**150**			**1370**	155	**52**
1.50	40	**26.5**				**1.338**	34.0			**3750**	425	**107**			**1300**	147	**37**
2.00	50	**41.5**				**1.687**	42.8			**5870**	665	**105**			**2030**	230	**36**

NOTES: *See Table 9 for* DP_{CO}.
See note following paragraph 2.4.1, page G-28, for discussion of C factor.

Table 3
Forged Steel Angle Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

SIZE		ALL STOP & CHECK VALVES						CHECK VALVES* WITH SPRINGS (STD)					CHECK VALVES* WITHOUT SPRINGS				
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}		SP_{FL}		C	ΔP_{FL}		SP_{FL}		C
Class 600 (PN 110) *Figure No. 829 Stop valves, 847 Stop-Check valves*																	
0.50	15	**3.3**	**0.55**	**0.19**	**0.11**	**0.546**	13.9	**6.0**	0.41	**512**	58.0	**87**	**0.8**	0.055	**187**	21.2	**32**
0.75	20	**5.7**				**0.742**	18.8			**884**	100	**82**			**323**	36.5	**30**
1.00	25	**17.5**				**0.957**	24.3			**2710**	307	**151**			**991**	112	**55**
1.25	32	**36**				**1.278**	32.5			**5580**	632	**174**			**2040**	231	**64**
1.50	40	**35**				**1.500**	38.1			**5430**	615	**123**			**1980**	224	**45**
2.00	50	**45.5**				**1.939**	49.3			**7050**	799	**96**			**2580**	292	**35**
Class 800 (PN 130) *Figure No. 849/849Y Stop valves, 869/869Y Stop-Check valves*																	
0.25	8	**2.6**	**0.64**	**0.25**	**0.16**	**0.364**	9.2	**6.0**	0.41	**403**	45.7	**155**	**0.8**	0.055	**147**	16.7	**57**
0.38	10	**2.9**				**0.493**	12.5			**450**	50.9	**94**			**164**	18.6	**34**
0.50	15	**2.8**				**0.546**	13.9			**434**	49.2	**74**			**159**	18.0	**27**
0.75	20	**4.8**				**0.742**	18.8			**744**	84.3	**69**			**272**	30.8	**25**
1.00	25	**10.5**				**0.957**	24.3			**1630**	184	**91**			**595**	67.3	**33**
1.25	32	**31**				**1.278**	32.5			**4810**	544	**150**			**1760**	199	**55**
1.50	40	**30**				**1.500**	38.1			**4650**	527	**105**			**1700**	192	**38**
2.00	50	**38.5**				**1.939**	49.3			**5970**	676	**81**			**2180**	247	**30**
Series 1500 *Figure No. 1049/1049Y Stop valves, 1069/1069Y Stop-Check valves*																	
0.25	8	**1.9**	**0.61**	**0.22**	**0.14**	**0.302**	7.7	**6.0**	0.41	**295**	33.4	**165**	**0.8**	0.055	**108**	12.2	**60**
0.38	10	**2.9**				**0.423**	10.7			**450**	50.9	**128**			**164**	18.6	**47**
0.50	15	**2.9**				**0.464**	11.8			**450**	50.9	**106**			**164**	18.6	**39**
0.75	20	**5.0**				**0.612**	15.5			**775**	87.8	**105**			**283**	32.1	**39**
1.00	25	**7.7**				**0.815**	20.7			**1190**	135	**92**			**436**	49.4	**33**
1.25	32	**20**				**1.160**	29.5			**3100**	351	**117**			**1130**	128	**43**
1.50	40	**20**				**1.338**	34.0			**3100**	351	**88**			**1130**	128	**32**
2.00	50	**33.5**				**1.687**	42.8			**5190**	588	**93**			**1900**	215	**34**
Series 1500 *Figure No. 1029 Stop valves, 1047 Stop-Check valves*																	
0.50	15	**2.7**	**0.65**	**0.24**	**0.16**	**0.464**	11.8	**6.0**	0.41	**419**	47.4	**99**	**0.8**	0.055	**153**	17.3	**36**
0.75	20	**4.7**				**0.612**	15.5			**729**	82.5	**99**			**266**	30.1	**36**
1.00	25	**7.5**				**0.815**	20.7			**1160**	132	**89**			**425**	48.1	**33**
1.25	32	**21**				**1.160**	29.5			**3260**	369	**123**			**1190**	135	**45**
1.50	40	**21**				**1.338**	34.0			**3260**	369	**93**			**1190**	135	**34**
2.00	50	**31.5**				**1.687**	42.8			**4920**	557	**88**			**1790**	203	**32**

NOTES: ***See Table 9 for DP_{CO}.***
See note following paragraph 2.4.1, page G-28, for discussion of C factor.
See Table 15, page G-55 for Hermavalves.

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Table 4
Edward Forged Steel Vertical Stem Globe Valve & 90° Bonnet Piston Check Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

Series 600 (PN 110) *Figure No. 828 Stop valve, 846 Stop-Check valve, 858 Piston Check valve*

SIZE		ALL STOP & CHECK VALVES						CHECK VALVES* WITH SPRINGS (STD)					CHECK VALVES* WITHOUT SPRINGS				
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}		SP_{FL}		C	ΔP_{FL}		SP_{FL}		C
0.50	15	**2.4**	**0.63**	**0.29**	**0.15**	**0.546**	13.9	**8.0**	0.55	**430**	48.7	**73**	**1.2**	0.083	**166**	18.8	**28**
0.75	20	**4.2**				**0.742**	18.8			**752**	85.2	**70**			**291**	33.0	**27**
1.00	25	**13.5**				**0.957**	24.3			**2400**	272	**133**			**929**	105	**52**
1.25	32	**27.5**				**1.278**	32.5			**4920**	558	**154**			**1910**	216	**59**
1.50	40	**27**				**1.500**	38.1			**4830**	548	**109**			**1870**	212	**42**
2.00	50	**35.5**				**1.939**	49.3			**6360**	720	**86**			**2460**	279	**33**

Series 1500 *Figure No. 1028 Stop valve, 1046 Stop-Check valve, 1058 Piston Check valve*

NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}		SP_{FL}		C	ΔP_{FL}		SP_{FL}		C
0.50	15	**3.6**	**0.68**	**0.27**	**0.17**	**0.464**	11.8	**8.0**	0.55	**645**	73.0	**153**	**1.2**	0.083	**250**	28.3	**59**
0.75	20	**6.2**				**0.612**	15.5			**1110**	126	**151**			**430**	48.7	**58**
1.00	25	**6.2**				**0.815**	20.7			**1110**	126	**85**			**430**	48.7	**33**
1.25	32	**18**				**1.160**	29.5			**3220**	365	**122**			**1250**	141	**47**
1.50	40	**17.5**				**1.338**	34.0			**3130**	355	**89**			**1210**	137	**35**
2.00	50	**24.5**				**1.687**	42.8			**4390**	497	**79**			**1700**	192	**30**

NOTES: ***See Table 9 for DP_{CO}.***
See note following paragraph 2.4.1, page G-28, for discussion of C factor.

Table 5
Forged Steel Ball Check Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

SIZE		CHECK VALVE FLOW COEFFICIENTS						CHECK VALVES WITH SPRINGS (STD)				
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}		SP_{FL}		C
Class 800 (PN 130) *Figure No. 832/832Y Ball Check valve*												
0.25	8	**1.5**	**0.53**	**0.16**	**0.11**	**0.364**	9.2	**6.0**	0.41	**233**	26.3	**89**
0.38	10	**3.5**				**0.493**	12.5			**543**	61.5	**114**
0.50	15	**3.5**				**0.546**	13.9			**543**	61.5	**93**
0.75	20	**6.1**				**0.742**	18.8			**946**	107	**88**
1.00	25	**14**				**0.957**	24.3			**2170**	246	**121**
1.25	32	**25**				**1.278**	32.5			**3880**	439	**121**
1.50	40	**39.5**				**1.500**	38.1			**6120**	694	**139**
2.00	50	**51.5**				**1.939**	49.3			**7990**	904	**108**
Series 1500 *Figure No. 1032/1032Y Ball Check valve*												
0.25	8	**1.1**	**0.77**	**0.37**	**0.16**	**0.302**	7.7	**6.0**	0.41	**171**	19.3	**95**
0.38	10	**2.5**				**0.423**	10.7			**388**	43.9	**110**
0.50	15	**2.4**				**0.464**	11.8			**372**	42.1	**88**
0.75	20	**4.3**				**0.612**	15.5			**667**	75.5	**91**
1.00	25	**6.6**				**0.815**	20.7			**1020**	116	**79**
1.25	32	**17.5**				**1.160**	29.5			**2710**	307	**103**
1.50	40	**17**				**1.338**	34.0			**2640**	299	**75**
2.00	50	**26.5**				**1.687**	42.8			**4110**	465	**74**
5000 CWP (345 Bar) *Figure No. 5160 Hydraulic Check valve*												
2.00	50	**14**	**0.96**	**0.57**	**0.24**	**1.502**	38.2	**20**	1.4	**3960**	449	**89**
10000 CWP (690 Bar) *Figure No. 160/160Y Hydraulic Check valve, 9160 Hydraulic Check valve*												
0.25	8	**0.40**	**0.96**	**0.57**	**0.24**	**0.133**	3.4	**20.0**	1.4	**113**	12.8	**326**
0.38	10	**0.80**				**0.205**	5.2			**227**	25.7	**275**
0.50	15	**1.3**				**0.252**	6.4			**368**	41.7	**295**
0.75	20	**3.5**				**0.434**	11.0			**991**	112	**268**
1.00	25	**2.9**				**0.599**	15.2			**821**	93.0	**117**
1.25	32	**3.5**				**0.808**	20.5			**991**	112	**77**
1.50	40	**3.5**				**0.926**	23.5			**991**	112	**58**
2.00	50	**14**				**1.156**	29.4			**3960**	449	**150**

NOTES: ***See Table 9 for DP_{CO}.***
See note following paragraph 2.4.1, page G-28, for discussion of C factor.

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Table 6
Hydraulic Stop Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

5,000 PSI (345 BAR) **CWP** *Figure No. 158/158Y Hydraulic Stop Valves*
10,000 PSI (690 BAR) **CWP** *Figure No. 5158, 9158 Hydraulic Stop Valves*

SIZE		ALL STOP VALVES						CHECK VALVES WITH SPRINGS (STD)			CHECK VALVES WITHOUT SPRINGS		
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}	SP_{FL}	C	ΔP_{FL}	SP_{FL}	C
0.25	8	1.6	0.48	0.30	.024	0.133	3.4	N/A					
0.38	10	1.6				0.205	5.2						
0.50	15	1.6				0.252	6.4						
0.75	20	3.6				0.434	11.0						
1.00	25	5.7				0.599	15.2						
1.25	32	9.1				0.808	20.5						
1.50	40	19				0.926	23.5						
2.00	50	33				1.156	29.4						

Table 7
Inclined Bonnet Blow-Off Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

SIZE		ALL STOP VALVES						CHECK VALVES WITH SPRINGS (STD)			CHECK VALVES WITHOUT SPRINGS		
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}	SP_{FL}	C	ΔP_{FL}	SP_{FL}	C
Class 300 (PN 50) *Figure No. 1441/1441Y*													
1.50	40	44	0.49	0.32	0.20	1.610	40.9	N/A					
2.00	50	67	0.69	0.44		2.067	52.5						
2.50	65	100	0.53	0.34		2.469	62.7						
Class 600 (PN 110) *Figure No. 1641/1641Y*													
1.50	40	43	0.55	0.35	0.20	1.500	38.1	N/A					
2.00	50	68	0.71	0.44		1.939	49.3						
2.50	65	110	0.56	0.35		2.323	59.0						

Table 8
Angle Blow-Off Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

SIZE		ALL STOP VALVES						CHECK VALVES WITH SPRINGS (STD)			CHECK VALVES WITHOUT SPRINGS		
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}	SP_{FL}	C	ΔP_{FL}	SP_{FL}	C

Class 300 (PN 50) *Figure No. 1443/1443Y*

NPS	DN	C_V	F_L	x_T	K_i	d	d (mm)	Check valves
1.50	40	**45**	**0.48**	**0.31**	**0.15**	**1.610**	40.9	**N/A**
2.00	50	**80**	**0.48**	**0.31**		**2.067**	52.5	
2.50	65	**110**	**0.53**	**0.34**		**2.469**	62.7	

Class 600 (PN 110) *Figure No. 1643/1643Y*

NPS	DN	C_V	F_L	x_T	K_i	d	d (mm)	Check valves
1.50	40	**41**	**0.60**	**0.38**	**0.15**	**1.500**	38.1	**N/A**
2.00	50	**81**	**0.50**	**0.31**		**1.939**	49.3	
2.50	65	**110**	**0.56**	**0.35**		**2.323**	59.0	

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Table 9
Crack-Open ∆P for Edward Forged Steel Check Valves, ΔP_{CO}- PSI (BAR)

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

VALVE TYPE	INSTALLATION ORIENTATION		VALVES WITH SPRINGS (STD)		VALVES WITHOUT SPRINGS	
Inclined, Bolted Bonnet Piston Lift	Horizontal	Bonnet up	**0.7 – 0.9**	0.05 – 0.06	**0.1 – 0.5**	0.007 – 0.03
	Horizontal	Bonnet sideways*	**0.3 – 0.8**	0.02 – 0.06	—	—
	Horizontal	Bonnet down*	**0.05 – 0.7**	0.003 – 0.05	—	—
	Vertical	Bonnet up	**0.7 – 1.0**	0.05 – 0.07	**0.1 – 0.3**	0.007 – 0.02
	Vertical	Bonnet down*	**0.05 – 0.7**	0.003 – 0.05	—	—
90°, Bolted Bonnet Piston Lift	Horizontal	Bonnet up	**0.8 – 1.0**	0.06 – 0.07	**0.1 – 0.6**	0.007 – 0.04
	Horizontal	Bonnet sideways*	**0.4 – 0.8**	0.03 – 0.06	—	—
	Horizontal	Bonnet down*	**0.05 – 0.6**	0.003 – 0.04	—	—
	Vertical		**0.4 – 0.8**	0.03 – 0.06	—	—
Inclined, Univalve® Piston Lift	Horizontal	Bonnet up	**1.0 – 1.5**	0.07 – 0.10	**0.4 – 0.8**	0.03 – 0.06
	Horizontal	Bonnet sideways*	**0.5 – 1.2**	0.03 – 0.08	—	—
	Horizontal	Bonnet down*	**0.05 – 1.1**	0.003 – 0.08	—	—
	Vertical	Bonnet up	**1.0 – 1.5**	0.07 – 0.10	**0.4 – 0.8**	0.03 – 0.06
	Vertical	Bonnet down*	**0.05 – 1.1**	0.003 – 0.08	—	—
Inclined, Ball Lift	Horizontal	Bonnet up	**0.9 – 1.7**	0.06 – 0.10	—	—
	Horizontal	Bonnet sideways*	**0.7 – 1.4**	0.05 – 0.10	—	—
	Horizontal	Bonnet down*	**0.5 – 1.2**	0.03 – 0.08	—	—
	Vertical	Bonnet up	**0.9 – 1.7**	0.06 – 0.10	—	—
	Vertical	Bonnet down*	**0.5 – 1.2**	0.03 – 0.08	—	—

****Not recommended because of possible accumulation of debris in valve neck.***

Figure 16
Edward Forged Steel Check Valve Flow Performance Curves

Figure 16-A

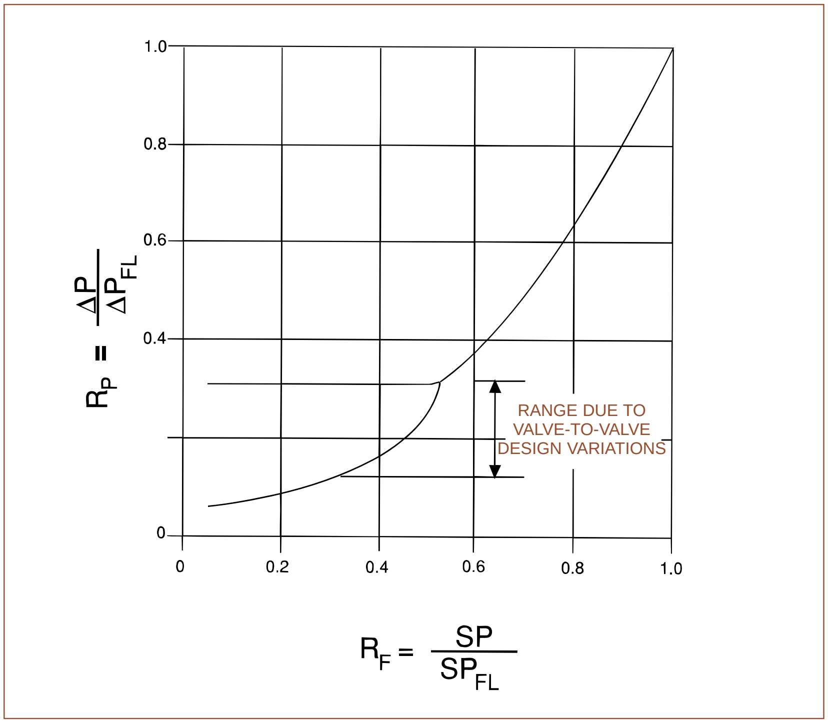

Figure 16-B

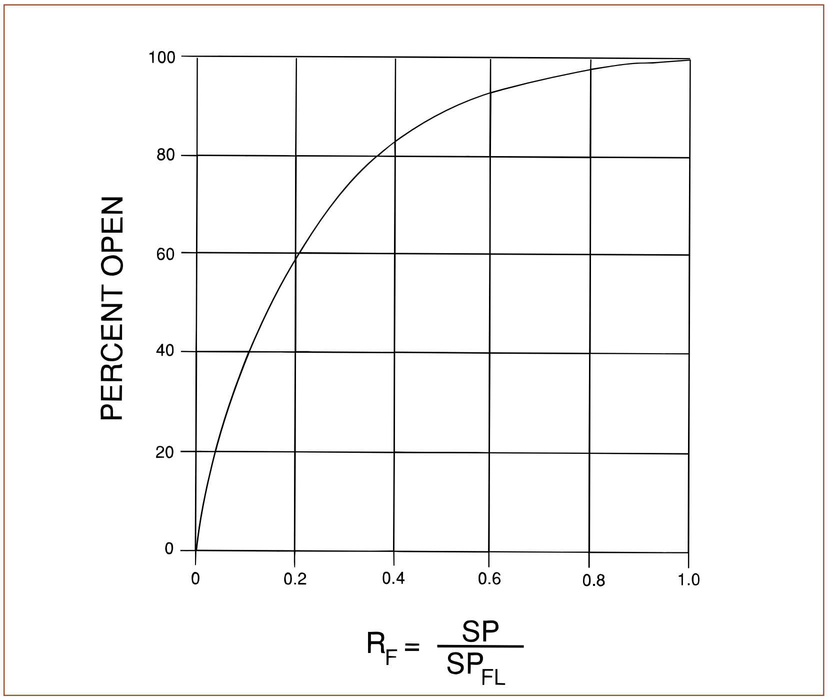

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Table 10
Edward Cast Steel Globe Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

SIZE		ALL STOP & CHECK VALVES						CHECK VALVE COEFFICIENTS							PERF. CURVES FIG. 17
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{CO}		ΔP_{FL}		SP_{FL}		C	
Class 300 (PN 50) *Figure No. 318/318Y Stop valves, 304/304Y Stop-Check valves, 394/394Y Check valves*															
2.5	65	110	0.53	0.34	0.20	2.50	63.5	0.58	0.040	0.31	0.021	5630	637	46	4
3	80	84	0.80	0.43	0.06	3.00	76.2	0.79	0.054	1.3	0.088	5990	679	34	4
4	100	120	0.79	0.43	0.06	4.00	102	0.80	0.055	1.4	0.095	8980	1020	29	4
5	125	215	0.79	0.43	0.06	5.00	127	0.97	0.067	1.8	0.12	18,100	2050	37	4
6	150	335	0.80	0.44	0.06	6.00	152	1.2	0.084	2.3	0.16	31,900	3610	45	1
8	200	580	0.76	0.39	0.06	8.00	203	1.2	0.086	1.2	0.085	40,800	4620	33	1
10	250	1000	0.77	0.40	0.06	10.00	254	1.2	0.081	1.1	0.079	67,600	7660	34	1
12	300	1550	0.77	0.40	0.06	12.00	305	1.3	0.092	1.2	0.084	107,000	12,100	38	1
Class 600 (PN 110) *Figure No. 616/616Y, 618/618Y, 716Y Stop valves, 606/604Y, 706Y Stop-Check valves, 694/694Y, 690/690Y, 794Y Check valves*															
2.5	65	84	0.97	0.61	0.10	2.50	63.5	0.79	0.054	1.3	0.088	5990	679	49	4
3	80	120	0.97	0.61	0.10	3.00	76.2	0.80	0.055	1.4	0.095	8980	1020	51	4
4	100	215	0.97	0.60	0.10	4.00	102	0.97	0.067	1.8	0.12	18,100	2050	58	4
5	125	335	0.97	0.61	0.10	5.00	127	1.2	0.084	2.3	0.16	31,900	3610	65	4
6	150	580	0.81	0.42	0.07	6.00	152	1.2	0.086	1.2	0.085	40,800	4620	58	1
8	200	1000	0.81	0.42	0.07	7.87	200	1.2	0.081	1.1	0.079	67,600	7660	56	1
10	250	1550	0.81	0.42	0.07	9.75	248	1.3	0.092	1.2	0.084	107,000	12,100	57	1
12	300	2200	0.81	0.42	0.07	11.75	298	1.5	0.10	1.4	0.099	169,000	19,100	62	1
14	350	2650	0.81	0.42	0.07	12.87	327	1.6	0.11	1.5	0.10	205,000	23,200	63	1
Class 900 (PN 150) *Figure No. 4016/4016Y, 4316Y Stop valves, 4006/4006Y, 4306Y Stop-Check valves, 4094/4094Y, 4394Y Check valves*															
3	80	110	0.96	0.60	0.10	2.87	72.9	0.92	0.063	1.5	0.10	8510	964	53	4
4	100	200	0.97	0.60	0.10	3.87	98.2	1.3	0.090	2.3	0.16	19,500	2210	66	5
5	125	305	0.97	0.61	0.10	4.75	121	1.3	0.092	2.5	0.18	30,600	3470	69	4
6	150	530	0.81	0.42	0.07	5.75	146	1.2	0.085	1.5	0.10	41,500	4700	64	3
8	200	910	0.81	0.42	0.07	7.50	191	1.3	0.093	1.5	0.10	69,500	7870	63	2
10	250	1400	0.81	0.42	0.07	9.37	238	1.6	0.11	1.8	0.12	119,000	13,500	69	1
12	300	2000	0.81	0.42	0.07	11.12	282	1.8	0.12	2.1	0.14	182,000	20,600	75	2
14	350	2400	0.81	0.42	0.07	12.25	311	1.6	0.11	1.9	0.13	211,000	23,900	72	2

See note following paragraph 2.4.1, page G-28, for discussion of C factor.

Table 10 (con't.)
Edward Cast Steel Globe Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

Class 1500 (PN 260) *Figure No. 2016, 7516/7516Y Stop valves, 2006Y, 7506/7506Y Stop-Check valves, 2094Y, 7594/7594Y Check valves*

SIZE		ALL STOP & CHECK VALVES						CHECK VALVE COEFFICIENTS							PERF. CURVES FIG. 17
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{CO}		ΔP_{FL}		SP_{FL}		C	
2.5	65	**72**	**0.92**	**0.54**	**0.08**	**2.25**	57.2	**0.76**	0.052	**1.3**	0.091	**5230**	592	**53**	**5**
3	80	**110**	**0.89**	**0.51**		**2.75**	69.9	**0.92**	0.063	**1.5**	0.10	**8510**	964	**57**	**4**
4	100	**200**	**0.85**	**0.47**		**3.62**	91.9	**1.3**	0.088	**2.3**	0.16	**19,300**	2190	**75**	**5**
5	125	**300**	**0.83**	**0.44**	**0.07**	**4.37**	111	**1.2**	0.080	**2.2**	0.15	**28,600**	3240	**76**	**4**
6	150	**465**	**0.80**	**0.42**		**5.37**	136	**1.4**	0.094	**1.4**	0.096	**35,000**	3960	**62**	**2**
8	200	**790**	**0.81**	**0.42**		**7.00**	178	**1.6**	0.11	**1.4**	0.097	**59,300**	6720	**62**	**1**
10	250	**1250**	**0.81**	**0.42**		**8.75**	222	**1.5**	0.10	**1.4**	0.100	**93,900**	10,600	**63**	**1**
12	300	**1750**	**0.81**	**0.42**		**10.37**	263	**1.5**	0.11	**1.8**	0.12	**147,000**	16,600	**70**	**3**
14	350	**2100**	**0.81**	**0.42**		**11.37**	289	**1.7**	0.12	**2.1**	0.14	**190,000**	21,500	**75**	**3**

Class 2500 (PN 420) *Figure No. 3916/3916Y, 4416Y Stop valves, 3906/3906Y, 4406Y Stop-Check valves, 3994/3994Y, 4494Y Check valves*

SIZE		ALL STOP & CHECK VALVES						CHECK VALVE COEFFICIENTS							PERF. CURVES FIG. 17
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{CO}		ΔP_{FL}		SP_{FL}		C	
2.5	65	**47**	**0.97**	**0.60**	**0.10**	**1.87**	47.5	**1.1**	0.075	**1.3**	0.088	**3370**	382	**49**	**6**
3	80	**68**	**0.97**	**0.61**		**2.25**	57.2	**1.4**	0.093	**1.6**	0.11	**5480**	620	**55**	**6**
4	100	**110**	**0.96**	**0.60**		**2.87**	72.9	**0.96**	0.066	**1.4**	0.095	**8280**	938	**51**	**5**
5	125	**175**	**0.97**	**0.60**		**3.62**	91.9	**1.4**	0.097	**2.2**	0.15	**16,600**	1880	**65**	**5**
6	150	**310**	**0.81**	**0.42**	**0.07**	**4.37**	111	**1.5**	0.11	**1.6**	0.11	**24,600**	2790	**66**	**3**
8	200	**530**	**0.81**	**0.42**		**5.75**	146	**2.2**	0.15	**2.2**	0.15	**49,800**	5640	**77**	**2**
10	250	**845**	**0.81**	**0.42**		**7.25**	184	**1.5**	0.10	**1.5**	0.11	**66,600**	7540	**65**	**2**
12	300	**1200**	**0.81**	**0.42**		**8.62**	219	**1.6**	0.11	**1.7**	0.11	**97,700**	11,100	**67**	**3**

See note following paragraph 2.4.1, page G-28, for discussion of C factor.

Figure 17
Edward Cast Steel Globe Piston Lift Check Valve Performance Curves

Figure 17-A

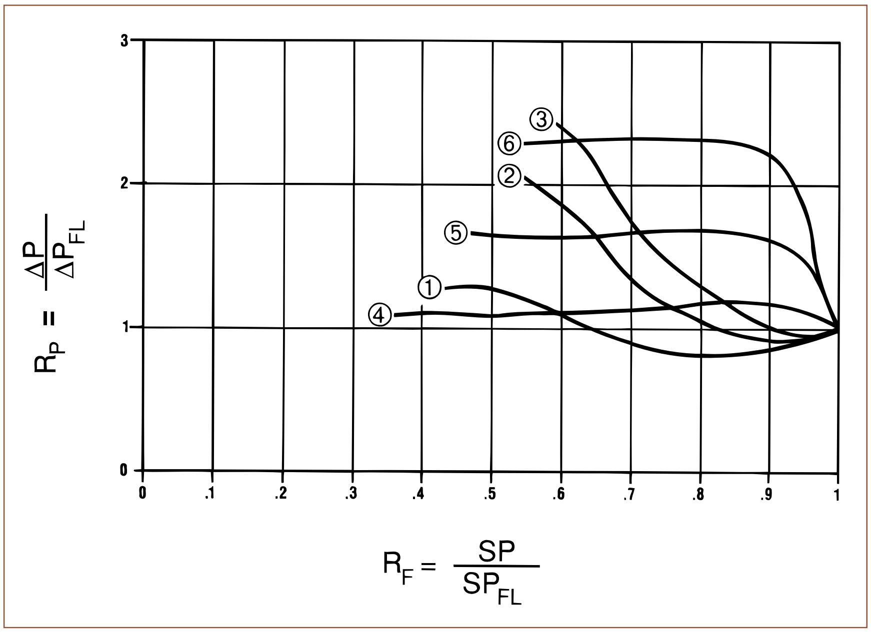

Figure 17-B

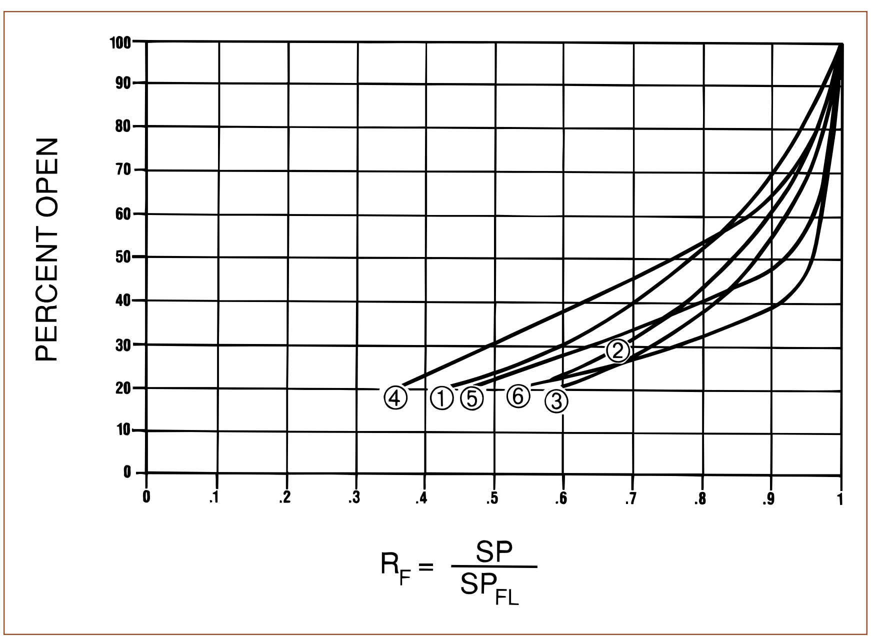

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Table 11
Edward Cast Steel Angle Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

SIZE		ALL STOP & CHECK VALVES						CHECK VALVE COEFFICIENTS							PERF. CURVES FIG. 18
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{CO}		ΔP_{FL}		SP_{FL}		C	
Class 300 (PN 50) *Figure No. 319/319Y/329/329Y, Stop valves, 303/303Y, Stop-Check valves, 391/391Y/393/393Y Check valves*															
2.5	65	110	0.53	0.34	0.15	2.50	63.5	0.63	0.043	0.46	0.032	4940	559	40	5
3	80	135	0.59	0.24		3.00	76.2	0.79	0.054	0.55	0.038	6300	714	36	5
4	100	195	0.58	0.23		4.00	102	0.80	0.055	0.59	0.041	9460	1070	30	5
5	125	345	0.59	0.23		5.00	127	0.97	0.067	0.75	0.052	18,900	2140	39	4
6	150	535	0.59	0.24	0.07	6.00	152	1.2	0.084	0.96	0.066	33,200	3760	47	1
8	200	860	0.59	0.23		8.00	203	1.2	0.086	0.75	0.052	47,200	5340	38	1
10	250	1500	0.59	0.23		10.00	254	1.2	0.081	0.70	0.048	78,200	8860	40	1
12	300	2250	0.59	0.23		12.00	305	1.3	0.092	0.74	0.051	124,000	14,000	44	1
Class 600 (PN 110) *Figure No. 617/617Y, 619/619Y, 717Y Stop valves, 605/605Y, 607/607, 707Y Stop-Check valves, 691/691Y, 695/695Y, 795Y Check valves*															
2.5	65	135	0.62	0.25		2.50	63.5	0.79	0.054	0.55	0.038	6300	714	51	5
3	80	195	0.62	0.25		3.00	76.2	0.80	0.055	0.59	0.041	9460	1070	54	5
4	100	345	0.62	0.25		4.00	102	0.97	0.067	0.75	0.051	18,800	2130	60	4
5	125	535	0.62	0.25	0.08	5.00	127	1.2	0.084	0.96	0.066	32,200	3760	68	4
6	150	860	0.64	0.25		6.00	152	1.2	0.086	0.75	0.052	47,200	5340	67	1
8	200	1500	0.63	0.25		7.87	200	1.2	0.081	0.70	0.048	78,200	8860	64	1
10	250	2250	0.63	0.25		9.75	248	1.3	0.092	0.74	0.051	124,000	14,000	66	1
12	300	3300	0.63	0.25		11.75	298	1.5	0.10	0.88	0.061	196,000	22,200	72	1
14	350	3950	0.63	0.25		12.87	327	1.6	0.11	0.90	0.062	237,000	26,900	73	1
Class 900 (PN 150) *Figure No. 4017/4017Y, 4317Y Stop valves, 4007/4007Y, 4307Y Stop-Check valves, 4095/4095Y, 4395Y Check valves*															
3	80	180	0.62	0.24		2.87	72.9	0.92	0.063	0.64	0.044	8980	1020	56	5
4	100	325	0.62	0.25		3.87	98.2	1.5	0.10	1.2	0.081	22,200	2510	75	5
5	125	485	0.63	0.25		4.75	121	1.2	0.083	1.0	0.072	31,200	3530	70	5
6	150	790	0.63	0.25	0.08	5.75	146	1.3	0.092	1.0	0.071	50,900	5770	78	3
8	200	1350	0.63	0.25		7.50	190	1.4	0.099	1.0	0.071	86,600	9810	78	3
10	250	2100	0.63	0.25		9.37	238	1.7	0.12	1.3	0.090	152,000	17,200	88	3
12	300	2950	0.63	0.25		11.12	282	1.8	0.13	1.4	0.093	218,000	24,700	90	2
14	350	3600	0.63	0.25		12.25	311	1.5	0.10	1.3	0.091	261,000	29,600	89	2
16	400	6450	0.56	0.19		14.00	356	1.9	0.13	0.74	0.051	350,000	39,700	91	2
18	450	*	*	*	0.06	15.75	400	*	*	*	*	*	*	*	*
20	500	10,000	0.56	0.19		17.50	444	1.7	0.11	0.76	0.052	553,000	62,600	92	3
24	600	14,500	0.56	0.19		21.00	533	2.6	0.18	1.1	0.073	940,000	106,000	109	3

See note following paragraph 2.4.1, page G-28, for discussion of C factor.

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Table 11 (con't.)
Edward Cast Steel Angle Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

Class 1500 (PN 260) *Figure No. 2017Y, 7517/7517Y Stop valves, 2007Y, 7507/7507Y Stop-Check valves, 2095Y, 7595/7595Y Check valves*

SIZE		ALL STOP & CHECK VALVES						CHECK VALVE COEFFICIENTS							PERF. CURVES FIG. 18
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{CO}		ΔP_{FL}		SP_{FL}		C	
2.5	65	**115**	**0.59**	**0.22**	**0.06**	**2.25**	57.2	**0.75**	0.052	**0.58**	0.040	**5560**	630	**56**	**6**
3	80	**180**	**0.57**	**0.21**		**2.75**	69.9	**0.92**	0.063	**0.64**	0.044	**8980**	1020	**60**	**5**
4	100	**320**	**0.55**	**0.19**		**3.62**	91.9	**1.50**	0.10	**1.20**	0.081	**22,000**	2490	**86**	**5**
5	125	**475**	**0.54**	**0.18**		**4.37**	111	**1.30**	0.093	**1.20**	0.083	**33,000**	3740	**88**	**5**
6	150	**690**	**0.63**	**0.25**	**0.08**	**5.37**	136	**1.50**	0.10	**1.00**	0.069	**43,800**	4970	**77**	**3**
8	200	**1150**	**0.63**	**0.25**		**7.00**	178	**1.60**	0.11	**0.99**	0.068	**73,900**	8370	**77**	**3**
10	250	**1850**	**0.63**	**0.25**		**8.75**	222	**1.60**	0.11	**1.20**	0.083	**127,000**	14,400	**85**	**3**
12	300	**2550**	**0.63**	**0.25**		**10.37**	263	**1.80**	0.13	**1.40**	0.094	**190,000**	21,500	**90**	**3**
14	350	**3100**	**0.63**	**0.25**		**11.37**	289	**1.70**	0.12	**1.30**	0.091	**225,000**	25,500	**89**	**3**
16	400	**5550**	**0.56**	**0.19**	**0.06**	**13.00**	330	**2.00**	0.14	**0.79**	0.055	**313,000**	35,400	**94**	**3**
18	450	**5350**	**0.54**	**0.19**		**14.62**	371	**2.00**	0.14	**0.86**	0.059	**313,000**	35,400	**75**	**3**
20	500	*	*	*	*	**16.37**	416	*	*	*	*	*	*	*	*
24	600	*	*	*	*	**19.62**	498	*	*	*	*	*	*	*	*

See note following paragraph 2.4.1, page G-28, for discussion of C factor.
**** Consult Edward Sales Representative***

Class 2500 (PN 420) *Fig. No. 3917/3917Y, 4417Y Stop valves, 3907/3907Y, 4407Y Stop-Check valves, 3995/3995Y, 4495Y Check valves*

SIZE		ALL STOP & CHECK VALVES						CHECK VALVE COEFFICIENTS							PERF. CURVES FIG. 18
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{CO}		ΔP_{FL}		SP_{FL}		C	
2.5	65	**75.5**	**0.62**	**0.24**	**0.08**	**1.87**	47.5	**1.1**	0.075	**0.57**	0.039	**3610**	409	**53**	**6**
3	80	**110**	**0.62**	**0.24**		**2.25**	57.2	**1.3**	0.091	**0.69**	0.048	**5770**	653	**58**	**6**
4	100	**180**	**0.62**	**0.24**		**2.87**	72.9	**0.96**	0.066	**0.61**	0.042	**8810**	998	**55**	**6**
5	125	**280**	**0.62**	**0.25**		**3.62**	91.9	**1.4**	0.097	**0.97**	0.067	**17,600**	1990	**68**	**5**
6	150	**455**	**0.63**	**0.25**		**4.37**	111	**1.5**	0.11	**0.96**	0.066	**28,300**	3210	**76**	**2**
8	200	**790**	**0.63**	**0.25**		**5.75**	146	**2.3**	0.16	**1.4**	0.096	**59,000**	6680	**91**	**2**
10	250	**1250**	**0.64**	**0.25**		**7.25**	184	**1.5**	0.10	**0.93**	0.064	**76,500**	8660	**74**	**2**
12	300	**1750**	**0.63**	**0.25**		**8.62**	219	**1.8**	0.13	**1.3**	0.088	**127,000**	14,400	**87**	**3**
14	350	**3400**	**0.40**	**0.10**	**0.05**	**9.50**	241	**2.1**	0.14	**0.89**	0.061	**204,000**	23,100	**115**	**3**
16	400	**3500**	**0.54**	**0.18**		**10.87**	276	**2.1**	0.14	**0.85**	0.058	**204,000**	23,100	**88**	**3**
18	450	**5450**	**0.50**	**0.15**		**12.25**	311	**2.5**	0.17	**1.00**	0.069	**347,000**	39,300	**118**	**3**
20	500	**5500**	**0.55**	**0.18**		**13.50**	343	**2.5**	0.17	**1.00**	0.070	**351,000**	39,800	**98**	**3**
22	550	**6900**	**0.55**	**0.18**		**14.87**	378	**2.5**	0.17	**0.97**	0.067	**429,000**	48,600	**99**	**3**
24	600	*	*	*	*	*	*	*	*	*	*	*	*	*	*

See note following paragraph 2.4.1, page G-28, for discussion of C factor.
**** Consult Edward Sales Representative***

Figure 18
Edward Cast Steel Angle Piston Lift Check Valve Performance Curves

Figure 18-A

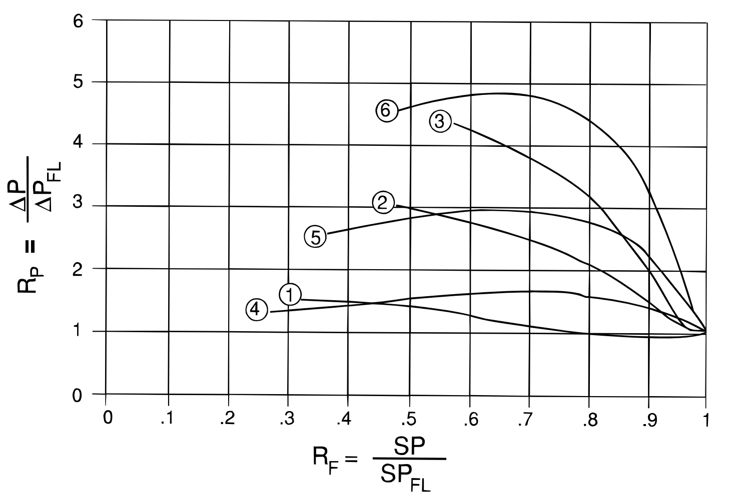

Figure 18-B

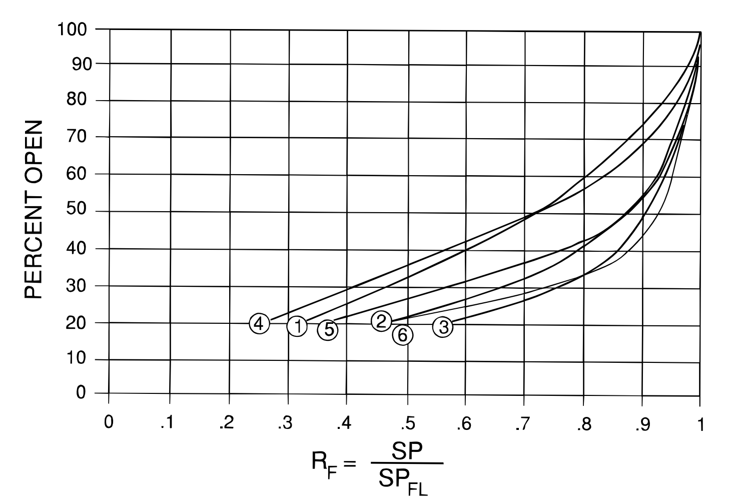

Table 12
Edward Cast Steel Flite-Flow® Stop & Stop-Check Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

Class 300/400 (PN 50/68) *Figure No. 1314, 1314Y, 1329, 1329Y Stop valves; 1302, 1302Y Stop-Check valves; 1390, 1390Y, 1392, 1392Y Piston Lift Check valves*

SIZE NPS	SIZE DN	ALL STOP & CHECK VALVES C_V	F_L	x_T	K_i	d	d (mm)	CHECK VALVE COEFFICIENTS ΔP_{CO}	ΔP_{CO} (metric)	ΔP_{FL}	ΔP_{FL} (metric)	SP_{FL}	SP_{FL} (metric)	C	PERF. CURVES FIG.19
2-1/2	65	110	0.53	0.34	0.02	2.50	64	0.9	0.06	0.91	0.063	6,750	765	55	1, 2
3	80	295	0.52	0.20	0.08	3.00	76	0.8	0.06	0.64	0.044	15,000	1,680	85	4, 4
4	100	525	0.52	0.20	0.08	4.00	102	0.8	0.06	0.66	0.046	27,000	3,070	86	4, 4
6	150	1,200	0.52	0.20	0.08	6.00	152	0.7	0.05	0.71	0.049	63,000	7,120	89	4, 4
8	200	2,100	0.52	0.20	0.08	8.00	200	0.9	0.06	0.67	0.046	109,000	12,400	87	4, 4
10	250	3,300	0.52	0.20	0.08	10.00	248	1.0	0.07	0.76	0.052	181,000	20,500	92	4, 4
12	300	4,750	0.52	0.20	0.08	12.00	305	1.1	0.08	0.87	0.060	279,000	31,500	99	4, 4
14	350	4,750	0.52	0.20	0.08	12.00	305	1.1	0.08	0.87	0.060	279,000	31,500	99	4, 4
16	400	4,750	0.53	0.22	0.09	12.00	305	1.5	0.10	0.87	0.060	279,000	31,500	99	4, 4

Class 600/700 (PN 110/120) *Figure No. 614, 614Y, 714Y Stop valves; 602, 602Y, 702Y Stop-Check valves; 692, 692Y, 792Y Piston Lift Check valves*

SIZE NPS	SIZE DN	ALL STOP & CHECK VALVES C_V	F_L	x_T	K_i	d	d (mm)	CHECK VALVE COEFFICIENTS ΔP_{CO}	ΔP_{CO} (metric)	ΔP_{FL}	ΔP_{FL} (metric)	SP_{FL}	SP_{FL} (metric)	C	PERF. CURVES FIG.19
3	80	295	0.52	0.20	0.08	3.00	76.2	0.8	0.06	0.44	0.030	12,400	1,400	70	4, 4
4	100	525	0.52	0.20	0.08	4.00	102	0.8	0.06	0.47	0.032	22,900	2,590	73	4, 4
6	150	1,200	0.52	0.20	0.08	6.00	152	0.7	0.05	0.53	0.037	54,500	6,170	77	4, 4
8	200	2,050	0.52	0.20	0.08	7.87	200	0.9	0.06	0.68	0.047	106,000	12,000	87	4, 4
10	250	3,100	0.52	0.20	0.08	9.75	248	1.0	0.07	0.85	0.059	182,000	20,600	98	4, 4
12	300	4,550	0.52	0.20	0.08	11.75	298	1.1	0.08	0.96	0.066	281,000	31,800	104	4, 4
14	350	4,550	0.52	0.20	0.08	11.75	298	1.1	0.08	0.96	0.066	281,000	31,800	104	4, 4
16	400	7,150	0.56	0.19	0.04	14.75	375	1.5	0.10	1.05	0.072	463,000	52,400	108	4, 4
20	500	11,000	0.52	0.20	0.08	18.25	484	1.4	0.10	0.96	0.066	677,000	76,700	104	1, 1
24	600	16,000	0.56	0.19	0.04	22.00	558	1.2	0.08	0.86	0.076	935,000	106,000	98	1, 2

Class 900/1100 (PN 150/190) *Figure No. 4014, 4014Y, 4314Y Stop valves; 4002, 4002Y, 4302Y Stop-Check valves; 4092, 4092Y, 4392Y Piston Lift Check valves*

SIZE NPS	SIZE DN	ALL STOP & CHECK VALVES C_V	F_L	x_T	K_i	d	d (mm)	CHECK VALVE COEFFICIENTS ΔP_{CO}	ΔP_{CO} (metric)	ΔP_{FL}	ΔP_{FL} (metric)	SP_{FL}	SP_{FL} (metric)	C	PERF. CURVES FIG.19
3	80	270	0.52	0.02	0.08	2.87	72.9	0.9	0.06	0.52	0.036	12,400	1,400	77	4, 4
4	100	490	0.52	0.02	0.08	3.87	98.2	0.9	0.06	0.53	0.037	22,600	2,550	77	4, 4
6	150	1,100	0.52	0.02	0.08	5.75	146	0.7	0.05	0.50	0.034	48,500	5,490	75	4, 4
8	200	1,850	0.52	0.02	0.08	7.50	191	0.8	0.06	0.65	0.045	94,200	10,700	85	4, 4
10	250	2,900	0.52	0.02	0.08	9.37	238	1.0	0.07	0.84	0.058	167,000	18,900	97	4, 4
12	300	4,050	0.52	0.02	0.08	11.12	282	1.1	0.08	0.93	0.064	248,000	28,100	102	4, 4
14	350	4,050	0.52	0.02	0.08	11.12	282	1.1	0.08	0.93	0.064	248,000	28,100	102	4, 4
16	400	6,450	0.52	0.02	0.08	14.00	356	1.3	0.09	1.09	0.075	426,000	48,200	111	4, 4

See note following paragraph 2.4.1, page G-28, for discussion of C factor.
**** Consult Edward Sales Representative***

Table 12 (con't.)
Edward Cast Steel Flite-Flow® Stop & Stop-Check Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

Class 1500/1800 (PN 260/310) *Figure No. 2014Y, 7514Y Stop valves; 2002Y, 7502Y Stop-Check valves; 2092Y, 7592Y Check valves*

SIZE NPS	DN	C_V	F_L	x_T	K_i	d	d (metric)	ΔP_{CO}	ΔP_{CO} (metric)	ΔP_{FL}	ΔP_{FL} (metric)	SP_{FL}	SP_{FL} (metric)	C	PERF. CURVES FIG. 19 A, B
3	80	270	0.52	0.20	0.08	2.87	72.9	1.0	0.07	0.51	0.035	12,200	1,380	75	4, 4
4	100	425	0.52	0.20		3.62	91.9	1.0	0.07	0.62	0.043	21,200	2,400	82	4, 4
6	150	950	0.61	0.23	0.05	5.37	136	1.3	0.09	0.73	0.050	51,200	5,800	90	1, 3
8	200	1,600	0.61	0.23		7.00	178	1.5	0.10	0.74	0.051	87,800	9,940	91	1, 2
10	250	2,500	0.61	0.23		8.75	222	1.5	0.10	0.89	0.061	150,000	17,000	100	1, 2
12	300	3,550	0.61	0.23		10.37	263	1.7	0.12	1.01	0.070	225,000	25,500	107	1, 2
14	350	3,550	0.59	0.22		10.37	263	1.7	0.12	1.01	0.070	225,000	25,500	106	1, 2
16	400	5,550	0.61	0.23		13.00	330	1.8	0.12	1.09	0.075	366,000	41,500	110	1, 2
18	450	5,550	0.59	0.22		13.00	330	1.8	0.12	1.09	0.075	366,000	41,500	110	1, 2
20	500	8,800	0.61	0.23		16.37	416	2.2	0.15	1.46	0.101	673,000	76,200	128	1, 2
24	600	8,800	0.59	0.23	0.06	16.37	416	2.3	0.16	*	*	*	*	*	*

Class 2500/2900 (PN 460/490) *Figure No. 3914Y, 4414Y Stop valves, 3902Y, 4402Y Stop-Check valves, 3992Y, 4492Y Check valves*
Class 2900 (PN 490) Size 3 and 4 only with figure numbers the same as Class 2500 valves.

SIZE NPS	DN	C_V	F_L	x_T	K_i	d	d (metric)	ΔP_{CO}	ΔP_{CO} (metric)	ΔP_{FL}	ΔP_{FL} (metric)	SP_{FL}	SP_{FL} (metric)	C	PERF. CURVES FIG. 19 A, B
3	80	165	0.52	0.20	0.08	2.25	57.2	1.1	0.08	0.71	0.049	8,850	1,000	89	4, 4
4	100	270	0.52	0.20		2.87	72.9	0.9	0.06	0.70	0.048	14,300	1,620	88	4, 4
6	150	625	0.61	0.23	0.05	4.37	111	1.5	0.11	0.84	0.058	36,300	4,110	97	1, 2
8	200	1,100	0.61	0.23		5.75	146	2.1	0.15	1.13	0.078	73,000	8,270	112	1, 2
10	250	1,750	0.61	0.22		7.25	184	1.5	0.10	0.80	0.055	97,600	11,100	95	1, 2
12	300	2,450	0.61	0.22		8.62	219	1.7	0.12	0.96	0.066	151,000	17,100	103	1, 3
14	350	3,550	0.53	0.17		10.37	263	1.9	0.13	1.17	0.081	242,000	27,400	115	1, 2
16	400	3,550	0.60	0.22		10.37	263	1.9	0.13	1.17	0.081	242,000	27,400	115	1, 2
18	450	5,550	0.55	0.18		13.00	330	2.3	0.16	1.38	0.095	412,000	46,700	124	1, 2
20	500	5,550	0.54	0.18		13.00	330	2.3	0.16	1.38	0.095	412,000	46,700	124	1, 2
24	600	8,100	0.60	0.22		15.69	399	2.4	0.17	1.61	0.111	648,000	73,400	134	1, 2

Series 4500 *Figure No. 4514Y, 5014Y Stop valves, 4502Y, 5002Y Stop-Check valves, 4592Y, 5092Y Check valves*

SIZE NPS	DN	C_V	F_L	x_T	K_i	d	d (metric)	ΔP_{CO}	ΔP_{CO} (metric)	ΔP_{FL}	ΔP_{FL} (metric)	SP_{FL}	SP_{FL} (metric)	C	PERF. CURVES FIG. 19 A, B
4	100	135	0.66	0.26	0.06	2.37	60.2	1.2	0.08	0.97	0.067	8,290	939	75	1, 2
6	150	305	0.64	0.24		3.37	85.6	1.5	0.10	1.75	0.121	25,300	2,870	113	1, 2
8	200	740	0.48	0.14	0.3	4.75	121	2.3	0.16	0.83	0.057	42,800	4,840	97	1, 2
10	250	1,100	0.51	0.16		5.75	146	1.7	0.12	0.86	0.059	63,600	7,200	98	1, 3

See note following paragraph 2.4.1, page G-28, for discussion of C factor.
*** Consult Edward Sales Representative***

Table 12 (con't.)
Edward Cast Steel Flite-Flow® Stop & Stop-Check Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

SIZE		ALL STOP & CHECK VALVES						CHECK VALVE COEFFICIENTS							PERF. CURVES FIG. 19
NPS	DN	C_v	F_L	x_T	K_i	d		ΔP_{CO}		ΔP_{FL}		SP_{FL}		C	
Class 2000 (PN 340) *Figure No. 2214Y, 3214Y Stop valves; 2002Y, 3202Y Stop-Check valves; 2292Y, 3292Y Check valves*															
12	300	**2950**	**0.52**	**0.20**	**0.08**	**9.50**	241	**1.7**	0.12	**0.85**	0.059	**172,600**	19,500	**97**	**4, 4**
14	350	**2950**	**0.52**	**0.20**		**9.50**	241	**1.7**	0.12	**0.85**	0.059	**172,600**	19,500	**97**	**4, 4**

Figure 19
Cast Steel Flite-Flow® Piston Lift Check Valve Performance Curves

Figure 19-A

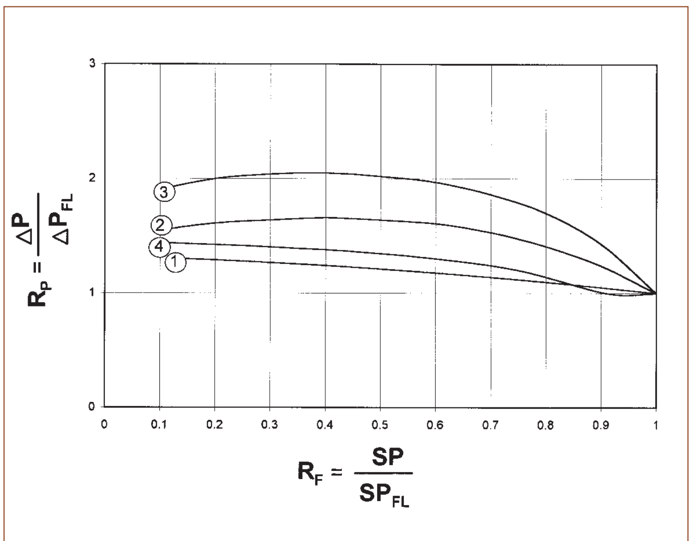

Figure 19-B

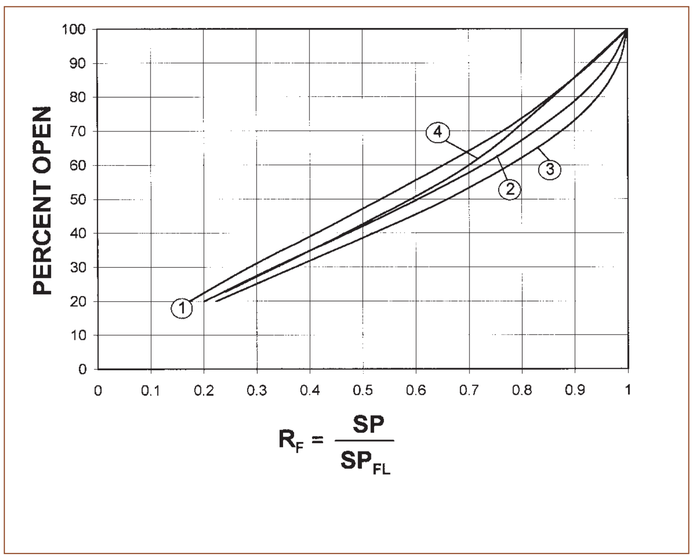

Table 13
Edward Cast Steel Tilting Disk Check Valve Flow Coefficients[1]

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

SIZE		CHECK VALVE FLOW COEFFICIENTS						CHECK VALVE COEFFICIENTS					PERF. CURVES FIG. 20
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}		SP_{FL}		C	
Class 600 (PN 110) *Figure No. 670Y, 770Y*													
6	150	1110	0.57	0.20		6.00	152	0.80	0.055	62,300	7,060	88	1
8	200	1850	0.57	0.20		7.87	200	1.0	0.069	115,000	13,000	95	1
10	250	2850	0.57	0.20		9.75	248	1.1	0.076	187,000	21,200	100	1
12	300	4100	0.57	0.20		11.75	298	1.2	0.083	285,000	32,300	105	1
14	350	4050	0.56	0.20	0.05	12.87	327	1.2	0.083	285,000	32,300	88	1
16	400	6500	0.57	0.20		14.75	375	1.4	0.097	481,000	54,500	113	1
18	450	8100	0.57	0.20		16.50	419	1.5	0.10	622,000	70,500	116	1
20	500	9950	0.57	0.20		18.25	464	1.6	0.11	786,000	89,000	120	1
Class 900 (PN 150) *Figure No. 970Y, 4370Y*													
2.5	65	195	0.44	0.12	0.02	2.25	57.2	1.0	0.069	12,200	1,380	123	1
3	80	245	0.57	0.20		2.87	72.9	0.60	0.041	12,200	1,380	75	1
4	100	215	0.59	0.23		3.87	98.2	0.80	0.055	12,200	1,380	41	1
6	150	990	0.57	0.20		5.75	146	0.80	0.055	56,800	6,430	87	1
8	200	1700	0.57	0.20		7.50	190	0.80	0.055	97,000	11,000	88	2
10	250	2400	0.56	0.20	0.05	9.37	238	0.90	0.062	145,000	16,400	84	2
12	300	3450	0.56	0.20		11.12	282	1.1	0.076	233,000	26,400	96	1
14	350	3300	0.56	0.20		12.25	311	1.3	0.090	233,000	26,400	79	1
16	400	4950	0.56	0.20		14.00	356	1.3	0.090	360,000	40,800	94	1
18	450	4700	0.57	0.21		15.75	400	1.5	0.10	360,000	40,800	74	1
20	500	9150	0.57	0.20		17.50	444	1.2	0.083	713,000	80,800	119	1
Class 1500 (PN 260) *Figure No. 1570Y, 2070Y*													
2.5	65	195	0.44	0.12	0.02	2.25	57.2	1.0	0.069	12,200	1,380	123	1
3	80	245	0.52	0.17		2.75	69.9	0.60	0.041	12,200	1,380	82	1
4	100	225	0.57	0.22		3.62	91.9	0.70	0.048	12,200	1,380	47	1
6	150	970	0.51	0.16		5.37	136	0.90	0.062	56,800	6,430	100	1
8	200	1650	0.51	0.16		7.00	178	0.90	0.062	97,000	11,000	101	2
10	250	2400	0.54	0.18	0.05	8.75	222	0.90	0.062	145,000	16,400	96	2
12	300	3450	0.53	0.17		10.37	263	1.1	0.076	233,000	26,400	110	1
14	350	3400	0.56	0.20		11.37	289	1.2	0.083	233,000	26,400	92	1
16	400	5050	0.57	0.20		13.00	330	1.3	0.090	360,000	40,800	108	1
18	450	4900	0.56	0.20		14.62	371	1.4	0.097	360,000	40,800	86	1
24	600	10,500	0.56	0.20		19.62	498	1.5	0.10	824,000	93,400	109	1

See note following paragraph 2.4.1, page G-28, for discussion of C factor.
[1] Crack open pressure drop ΔP_{CO} values are generally less than 0.25 psi (0.01 bar).

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Table 13 (con't.)
Edward Cast Steel Tilting Disk Check Valve Flow Coefficients[1]

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

SIZE		CHECK VALVE FLOW COEFFICIENTS						CHECK VALVE COEFFICIENTS					PERF. CURVES FIG. 20
NPS	DN	C_V	F_L	x_T	K_i	d		ΔP_{FL}		SP_{FL}		C	
Class 2500 (PN 420) *Figure No. 2570Y, 4470Y*													
2.5	65	125	0.47	0.13	0.01	1.87	47.5	2.4	0.17	12,200	1,380	178	1
3	80	195	0.44	0.12		2.25	57.2	1.0	0.069	12,200	1,380	123	1
4	100	245	0.57	0.20	0.05	2.87	72.9	0.60	0.041	12,200	1,380	75	1
6	150	655	0.50	0.15		4.37	111	0.40	0.028	26,500	3,000	71	1
8	200	990	0.57	0.20		5.75	146	0.80	0.055	56,700	6,420	87	2
10	250	1650	0.54	0.18		7.25	184	0.90	0.062	97,000	11,000	94	2
12	300	2400	0.53	0.17		8.62	219	0.50	0.034	156,000	17,700	107	1
14	350	3250	0.47	0.14		9.50	241	1.3	0.090	233,000	26,400	131	1
16	400	3450	0.57	0.20		10.87	276	1.1	0.076	233,000	26,400	100	1
18	450	5050	0.51	0.16		12.25	311	1.3	0.090	360,000	40,800	122	1
20	500	5000	0.56	0.20		13.50	343	1.3	0.090	360,000	40,800	101	1
Class 4500 (PN 760) *Figure No. 4570Y, 5070Y Check valves*													
6	150	420	0.43	0.11	.03	3.76	95.5	0.70	0.048	21,900	2480	79	1
8	200	675	0.45	0.12		4.75	121	0.8	0.055	37,000	4190	84	1

See note following paragraph 2.4.1, page G-28, for discussion of C factor.
[1] Crack open pressure drop ΔP_{CO} values are generally less than 0.25 psi (0.01 bar).

Figure 20
Tilting Disk Check Valve Performance Curves

Figure 20-A

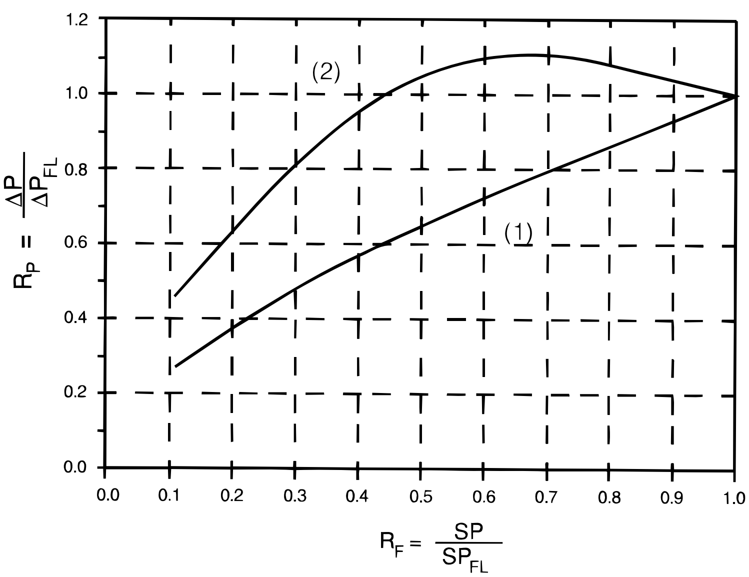

Figure 20-B

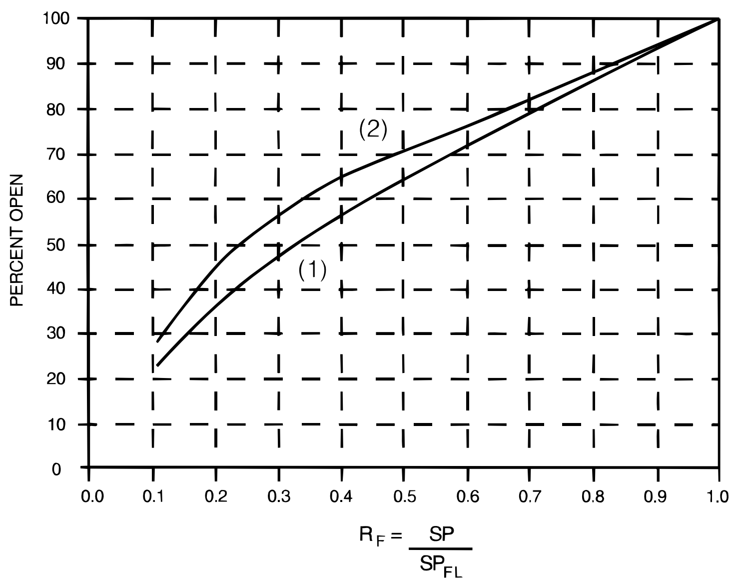

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Table 14
Edward Cast Steel Equiwedge® Gate Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

Regular Port Gate Valves

Class 600 (PN 110) *Figure No. A1611 Stop valves*

Size NPS	Size DN	C_V	F_L	X_T	K_i	d	
2.5	65	395	0.74	0.23	0.02	2.50	63.5
3.0	80	325	0.57	0.19	0.02	3.00	76.2
4.0	100	545	0.58	0.20	0.03	4.00	102
6.0	150	2350	0.38	0.08	0.02	6.00	152

Class 900 (PN 150) *Figure No. A1911, Stop valves*

Size NPS	Size DN	C_V	F_L	X_T	K_i	d	
2.5	65	270	0.88	0.33	0.02	2.25	57.2
3.0	80	340	0.60	0.20	0.03	2.87	72.9
4.0	100	570	0.40	0.18	0.02	3.87	98.2

Regular Port Gate Valves

Class 600 (PN 110) *Figure No. A1611Y Stop valves*

Size NPS	Size DN	C_V	F_L	X_T	K_i	d	
2.5	65	385	0.76	0.25	0.02	2.50	63.5
3.0	80	365	0.55	0.16	0.02	2.90	73.7
4.0	100	625	0.53	0.16	0.03	3.83	97.3
6.0	150	2350	0.41	0.09	0.02	5.75	146

Class 900 (PN 150) *Figure No. A1911Y Stop valves*

Size NPS	Size DN	C_V	F_L	X_T	K_i	d	
2.5	65	280	0.75	0.24	0.02	2.12	53.8
3.0	80	400	0.61	0.18	0.03	2.62	66.5
4.0	100	670	0.54	0.15	0.02	3.62	91.9

Regular Port Gate Valves

Class 600 (PN 110) *Figure No. 1611/ 1611Y, 1711Y Stop valves*

Size NPS	Size DN	C_V	F_L	X_T	K_i	d	
2.5	65	380	0.77	0.25	0.02	2.50	63.5
3.0	80	610	0.44	0.10	0.02	3.00	76.2
4.0	100	1250	0.41	0.08	0.03	4.00	102
6.0	150	3250	0.40	0.07	0.02	6.00	152
8.0	200	5300	0.35	0.06	0.02	7.87	200
10.0	250	8550	0.34	0.06	0.01	9.75	248
12.0	300	12,000	0.31	0.05	0.01	11.75	298
14.0	350	14,000	0.32	0.05	0.01	12.87	327
16.0	400	18,500	0.32	0.05	0.01	14.75	375
18.0	450	25,500	0.30	0.05	0.01	16.50	419
20.0	500	30,500	0.31	0.05	0.01	18.25	464
22.0	550	36,500	0.30	0.05	0.01	20.12	511
24.0	600	46,500	0.30	0.05	0.01	22.00	559
26.0	650	53,500	0.30	0.05	0.01	23.75	603
28.0	700	62,500	0.29	0.04	0.01	25.50	648
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—

Venturi Port Gate Valves

Class 600 (PN 110) *Figure No. 1611BY, 1711BY Stop valves*

Size NPS	Size DN	C_V	F_L	X_T	K_i	d	
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—
8x6x8	200x150x200	2650	0.33	0.07	0.03	7.87	200
10x8x10	250x200x250	4500	0.32	0.07	0.02	9.75	248
12x10x12	300x250x300	7100	0.32	0.06	0.02	11.75	298
14x12x14	350x300x350	9900	0.32	0.06	0.02	12.87	327
16x14x16	400x350x400	12,000	0.31	0.06	0.02	14.75	375
18x16x18	450x400x450	17,500	0.29	0.05	0.01	16.50	419
20x18x20	500x450x500	22,000	0.30	0.06	0.02	18.25	464
22x20x22	550x500x550	29,000	0.28	0.05	0.01	20.12	511
24x20x24	600x500x600	24,500	0.30	0.06	0.02	22.00	559
26x22x26	650x550x650	30,000	0.30	0.06	0.02	23.75	603
28x24x28	700x600x700	40,500	0.29	0.05	0.01	25.50	648
30x26x30	750x650x750	46,500	0.29	0.05	0.01	27.37	695
32x28x32	800x700x800	52,000	0.30	0.05	0.01	29.25	743

Table 14 (con't.)
Edward Cast Steel Equiwedge® Gate Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

Regular Port Gate Valves

Class 900 (PN 150) *Figure No. 1911/ 1911Y, 14311Y Stop valves*

Size NPS	Size DN	C_V	F_L	X_T	K_i	d	
2.5	65	380	0.63	0.17	0.02	2.25	57.2
3.0	80	455	0.44	0.11	0.03	2.87	72.9
4.0	100	990	0.42	0.09	0.02	3.87	98.2
6.0	150	2350	0.41	0.09	0.02	5.75	146
8.0	200	4200	0.37	0.07	0.02	7.50	190
10.0	250	6250	0.40	0.08	0.02	9.37	238
12.0	300	9500	0.36	0.07	0.02	11.12	282
14.0	350	12,000	0.35	0.06	0.02	12.25	311
16.0	400	15,000	0.35	0.06	0.02	14.00	356
18.0	450	19,500	0.33	0.06	0.02	15.75	400
20.0	500	26,000	0.35	0.06	0.02	17.50	444
22.0	550	28,000	0.38	0.07	0.02	19.25	489
24.0	600	38,000	0.32	0.05	0.01	21.00	533
26.0	650	45,000	0.32	0.05	0.01	22.75	578
28.0	700	52,500	0.31	0.05	0.01	24.50	622
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—

Class 1500 (PN 260) *Figure No. 11511/11511Y, 12011Y Stop valves*

Size NPS	Size DN	C_V	F_L	X_T	K_i	d	
2.5	65	305	0.78	0.26	0.02	2.25	57.2
3.0	80	420	0.52	0.14	0.03	2.75	69.9
4.0	100	760	0.47	0.12	0.03	3.62	91.9
6.0	150	1650	0.54	0.15	0.04	5.37	136
8.0	200	3150	0.48	0.12	0.03	7.00	178
10.0	250	5500	0.40	0.08	0.02	8.75	222
12.0	300	6850	0.42	0.09	0.02	10.37	263
14.0	350	9700	0.40	0.08	0.02	11.37	289
16.0	400	12,000	0.39	0.08	0.02	13.00	330
18.0	450	15,000	0.37	0.07	0.02	14.62	371
20.0	500	18,500	0.37	0.07	0.02	16.37	416
22.0	550	23,000	0.37	0.07	0.02	18.00	457
24.0	600	27,000	0.37	0.08	0.02	19.62	498
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—

Venturi Port Gate Valves

Class 900 (PN 150) *Figure No. 1911BY, 14311BY Stop valves*

Size NPS	Size DN	C_V	F_L	X_T	K_i	d	
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—
8x6x8	200x150x200	2000	0.37	0.09	0.03	7.50	190
10x8x10	250x200x250	3500	0.35	0.08	0.02	9.37	238
12x10x12	300x250x300	5950	0.35	0.08	0.02	11.12	282
14x12x14	350x300x350	7700	0.39	0.09	0.03	12.25	311
16x14x16	400x350x400	10,000	0.35	0.07	0.02	14.00	356
18x16x18	450x400x450	14,000	0.32	0.06	0.02	15.75	400
20x18x20	500x450x500	18,000	0.32	0.06	0.02	17.50	444
22x20x22	550x500x550	25,000	0.31	0.06	0.02	19.25	489
24x20x24	600x500x600	23,000	0.31	0.06	0.02	21.00	533
26x22x26	650x550x650	28,000	0.31	0.06	0.02	22.75	578
28x24x28	700x600x700	33,500	0.31	0.06	0.02	24.50	622
30x26x30	750x650x750	38,000	0.32	0.06	0.02	26.25	667
32x28x32	800x700x800	48,000	0.29	0.05	0.01	28.00	711

Class 1500 (PN 260) *Figure No. 11511BY, 12011BY Stop valves*

Size NPS	Size DN	C_V	F_L	X_T	K_i	d	
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—
8x6x8	200x150x200	1650	0.43	0.12	0.04	7.00	178
10x8x10	250x200x250	2950	0.41	0.11	0.03	8.75	222
12x10x12	300x250x300	4500	0.40	0.10	0.03	10.37	263
14x12x14	350x300x350	7050	0.37	0.08	0.02	11.37	289
16x14x16	400x350x400	8700	0.37	0.08	0.02	13.00	330
18x16x18	450x400x450	11,000	0.37	0.08	0.02	14.62	371
20x18x20	500x450x500	13,500	0.36	0.08	0.02	16.37	416
22x20x22	550x500x550	18,000	0.34	0.07	0.02	18.00	457
24x20x24	600x500x600	17,000	0.35	0.07	0.02	19.62	498
26x22x26	650x550x650	20,500	0.35	0.07	0.02	21.25	540
28x24x28	700x600x700	24,000	0.36	0.08	0.02	23.00	584

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Table 14 (con't.)
Edward Cast Steel Equiwedge® Gate Valve Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

Regular Port Gate Valves

Class 2500 (PN 420) *Figure No. 12511/ 12511Y, 14411Y Stop valves*

Size NPS	Size DN	C_V	F_L	X_T	K_i	d	d
2.5	65	150	0.78	0.50	0.02	1.87	47.5
3.0	80	230	0.58	0.18	0.04	2.25	57.2
4.0	100	340	0.59	0.19	0.04	2.87	72.9
6.0	150	910	0.61	0.19	0.05	4.37	111
8.0	200	1850	0.51	0.14	0.04	5.75	146
10.0	250	2950	0.48	0.12	0.03	7.25	184
12.0	300	4350	0.46	0.11	0.03	8.62	219
14.0	350	5150	0.47	0.12	0.03	9.50	241
16.0	400	7050	0.46	0.11	0.03	10.87	276
18.0	450	8950	0.46	0.11	0.03	12.25	311
20.0	500	11,500	0.45	0.11	0.03	13.50	343
22.0	550	14,000	0.45	0.11	0.03	14.87	378
24.0	600	17,500	0.43	0.10	0.03	16.25	413
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—

Venturi Port Gate Valves

Class 2500 (PN 420) *Figure No. 12511B/ 12511BY, 14411BY Stop valves*

Size NPS	Size DN	C_V	F_L	X_T	K_i	d	d
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—
8x6x8	200x150x200	1000	0.44	0.12	0.04	5.75	146
10x8x10	250x200x250	1650	0.46	0.14	0.04	7.25	184
12x10x12	300x250x300	2750	0.43	0.11	0.03	8.62	219
14x12x14	350x300x350	3900	0.46	0.13	0.03	9.50	241
16x14x16	400x350x400	4850	0.44	0.12	0.03	10.87	276
18x16x18	450x400x450	6450	0.43	0.11	0.03	12.25	311
20x18x20	500x450x500	8200	0.44	0.12	0.03	13.50	343
22x20x22	550x500x550	11,500	0.39	0.10	0.03	14.87	378
24x20x24	600x500x600	10,500	0.39	0.10	0.03	16.25	413
26x22x26	650x550x650	13,000	0.39	0.09	0.02	17.62	448
28x24x28	700x600x700	16,000	0.39	0.09	0.03	19.00	483

Table 15
Edward Forged Steel Hermavalve® Flow Coefficients

Bold faced numerals are in U.S. customary units or dimensionless.
Brown numerals are in metric units.

NPS	DN	REGULAR PORT HERMAVALVES Fig. No. 15004/15104, 15008/15108, 16004, 16008				REDUCED PORT HERMAVALVES Fig. No. 15014/15114, 15018/15118, 16014, 16018				d	
		C_V	F_L	x_T	K_i	C_V	F_L	x_T	K_i		
0.05	15	4.9	0.46	0.31	0.07	—	—	—	—	0.464	11.8
0.75	20	6.1	0.52	0.36	0.09	—	—	—	—	0.612	15.5
1.00	25	11	0.55	0.38	0.10	6.1	0.51	0.36	0.09	0.815	20.7
1.50	40	32	0.62	0.39	0.13	11	0.53	0.37	0.09	1.338	34.0
2.00	50	50	0.68	0.40	0.15	32	0.57	0.37	0.11	1.687	42.8
2.50	65	—	—	—	—	50	0.59	0.37	0.12	2.125	54.0

Figure 21
Ratio of Specific Heats (k) for Some Gases

k = 1.3	Ammonia	Carbon Dioxide	Dry Steam	Methane	Natural Gas
k = 1.4	Air	Carbon Monoxide	Hydrogen	Nitrogen	Oxygen

Figure 22A
Saturated Water - Temperature, Pressure & Density

(U.S. Units)

Water Temp. °F	32	70	100	200	300	400	500	550	600	650	700	705
Vapor Pressure, p_v	0.09	0.36	0.95	11.5	67	247	681	1045	1543	2208	3094	3206
Water Density, ρ	62.4	62.3	62.0	60.1	57.3	53.7	49.0	46.0	42.3	37.4	27.3	19.7

P = Pressure in psia, ρ = Density in lb./ft^3

Figure 22B
Saturated Water - Temperature, Pressure & Density

(Metric)

Water Temp. °C	0	25	50	100	150	200	250	300	350	370	374
Vapor Pressure, p_v	.006	.032	.123	1.01	4.76	15.6	39.8	85.9	165.4	211	221
Water Density, ρ	1000	997	988	958	917	865	799	712	574	452	315

P = Pressure in Bar Absolute, ρ = Density in Kg/m^3

Figure 23 Density of Steam

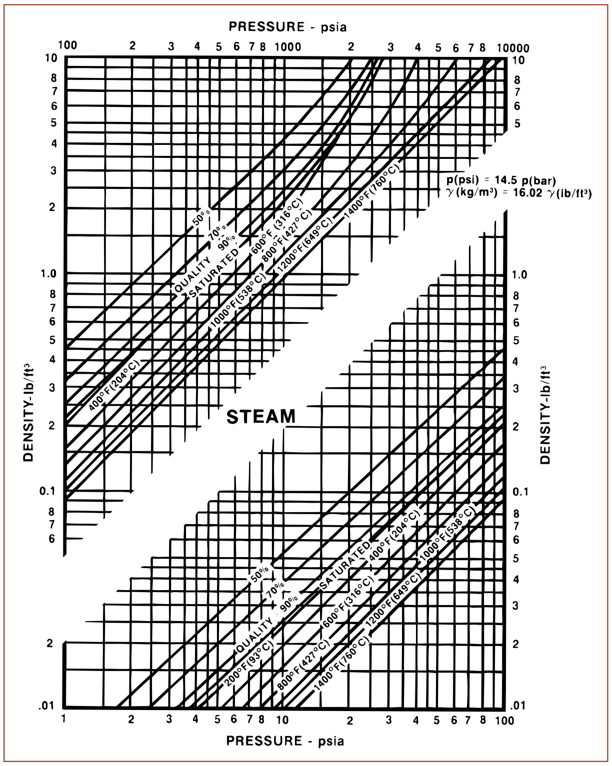


Figure 24 Density of Air

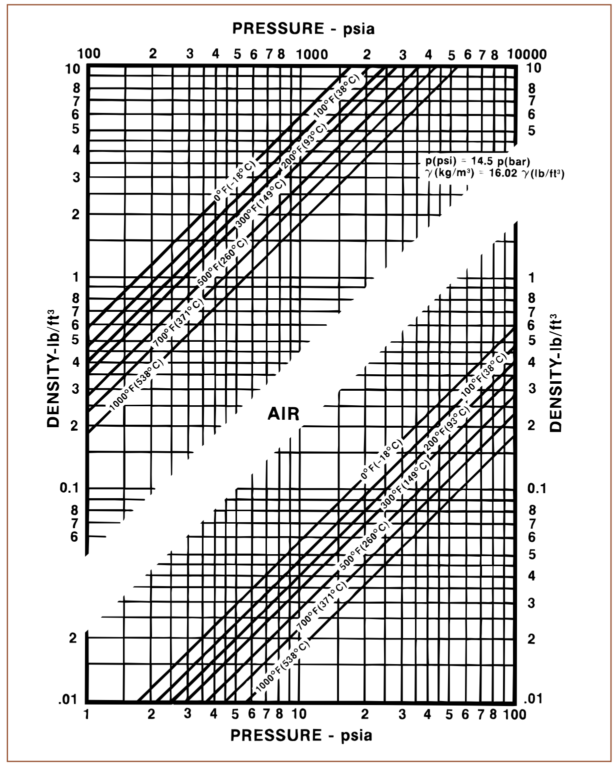
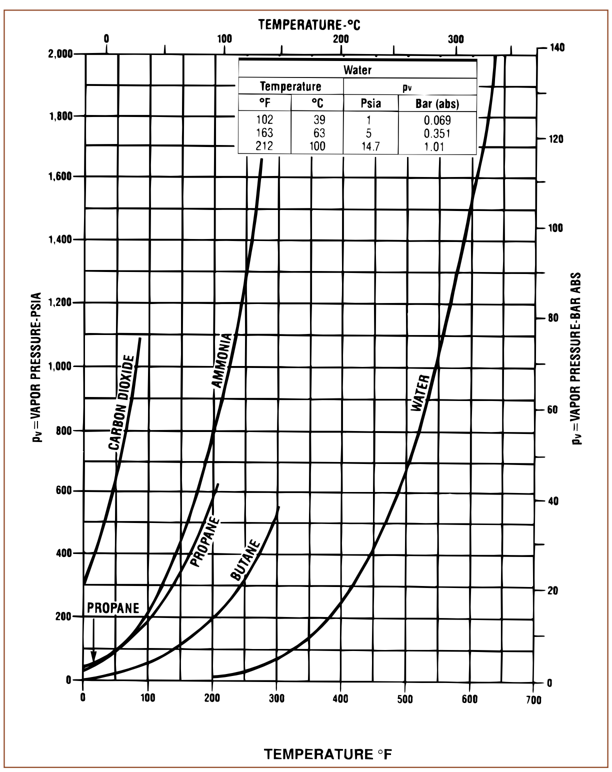


Figure 25 Vapor Pressure of Liquid



Conversion of Measurement Units

Length

1 in. = 25.4 mm	1 mile = 5280 ft
1 in. = 2.54 cm	1 mile = 1.609 km
1 in. = 0.0254 m	1 km = 3281 ft
1 ft = 0.3048 m	1 m = 39.37 in.

Area

1 in.^{2} = 645.2 mm^2	1 m^2 = 10.76 ft^2
1 in.^{2} = 6.452 cm^2	1 m^2 = 1550 in.^{2}
1 ft^2 = 144 in.^{2}	

Volume

1 in.^{3} = 16.39 cm^3	1 m^3 = 35.31 ft^3
1 ft^3 = 1728 in.^{3}	1 m^3 = 264.2 U.S. gal.
1 U.S. gal. = 231 in.^{3}	1 m^3 = 220 Imp. gal.
1 U.S. gal. = 0.1337 ft^3	1 m^3 = 1000 liters
1 U.S. gal. = 0.8327 Imp. gal.	1 liter = 61.02 in.^{3}
	1 liter = 1000 cm^3
1 U.S. gal. = 3.7854 liters	
1 ft^3 = 28.32 liters	1 ml = 1 cm^3

Density

1 lb/ft^3 = 16.02 kg/m^3
1 lb./ft^3 = 0.01602 g/cm^3
1 lb./in.^{3} = 1728 lb/ft^3

density = specific gravity x reference density
density = 1/specific volume

Specific Volume

specific volume = 1/density

Temperature

$$T(°C) = \frac{T(°F - 32)}{1.8}$$

T(°F) = 1.8 T(°C) + 32
T(°R) = T(°F) + 460
T(°K) = T(°C) + 273
T(°R) = 1.8 T(°K)

where:
°C = degrees Celsius
°F = degrees Fahrenheit
°K = degrees Kelvin (absolute temperature)
°R = degrees Rankine (absolute temperature)

Specific Gravity – Liquids

$$G_l = \frac{\text{density of liquid}}{\text{density of water at reference condition}}$$

Commonly used relations are:

$$G_l = \frac{\text{density of liquid}}{\text{density of water at 60°F and atmospheric pressure}} = \frac{\rho\,(\text{lb/ft}^3)}{62.38\,(\text{lb/ft}^3)}$$

and

$$G_l = \frac{\text{density of liquid}}{\text{density of water at 4°C and atmospheric pressure}} = \frac{\rho\,(\text{kg/m}^3)}{1000\,(\text{kg/m}^3)}$$

For practical purposes, these specific gravities may be used interchangeably, as the reference densities are nearly equivalent.

Specific gravities are sometimes given with two temperatures indicated, e.g.,

$$G_l\,\frac{60°F}{60°F},\quad G_l\,\frac{15.5°C}{4°C},\quad G_l\ 60°F/60°$$

The upper temperature is that of the liquid whose specific gravity is given, and the lower value indicates the water temperature of the reference density. If no temperatures are shown, assume that the commonly used relations apply.

For petroleum liquids having an "API degrees" specification:

$$G_l\ 60°F/60° = \frac{141.5}{131.5 + \text{API degrees}}$$

Pressure

1 Mpa = 145 psi	1 psi = 6895 Pa
1 pond = 1 gf	1 psi = 6895 N/m^2
1 std atm = 14.696 psi	1 Pa = 1 N/m^2
1 std atm = 1.0133 bar	1 bar = 14.50 psi
1 std atm = 1.0133 x 10^5 N/m^2	1 bar = 100,000 N/m^2
	1 kgf/cm^2 = 14.22 psi

1 std atm = 760 torr absolute pressure =
gage pressure + atmospheric pressure

Specific Gravity – Gases

$$G_g = \frac{\text{density of gas (at pressure and temperature of interest)}}{\text{density of air (at same pressure and temperature)}}$$

Because the relation between density, pressure and temperature does not always behave in an ideal way (i.e., ideally, density is proportional to pressure divided by temperature, in absolute units), use of the above relation requires that the pressure and temperature of interest be specified. This means that the specific gravity of a gas as defined may vary with pressure and temperature (due to "compressibility" effects).

Frequently, specific gravity is defined using:

$$G_g = \frac{\text{molecular weight of gas}}{\text{molecular weight of air}} = \frac{M_w}{28.96}$$

If this relation is used to calculate density, one must be careful to consider "compressibility" effects. When the pressure and temperature of interest are at or near "standard" conditions (14.73 psia, 60°F) or "normal" conditions (1.0135 bar abs, 0°C), specific gravities calculated from either of the above relations are essentially equal.

Pressure Head

1 foot of water at 60°F = 0.4332 psi

$$p(\text{psi}) = \frac{\rho(\text{lb/ft}^3) \times h(\text{feet of liquid})}{144}$$

$$p(\text{N/m}^2) = \frac{\rho(\text{kg/m}^3) \times h(\text{meters of liquid})}{0.1020}$$

$$p(\text{bar}) = \frac{\rho(\text{kg/m}^3) \times h(\text{meters of liquid})}{10200}$$

1 meter of water at 20°C = 9.790 kN/m^2
1 meter of water at 20°C = 97.90 mbar
1 meter of water at 20°C = 1.420 psi

Flow Rate

- mass units

1 lb/hr = 0.4536 kg/hr
1 metric tonne/hr = 2205 lb/hr

- liquid volume units

1 U.S. gpm = 34.28 BOPD
BOPD = barrels oil per day
1 U.S. gpm = 0.8327 Imp. gpm
1 U.S. gpm = 0.2273 m^3/hr
1 U.S. gpm = 3.785 liters/min
1 m^3/hr = 16.68 liters/min
1 ft^3/s = 448.8 U.S. gpm

- mixed units

w(lb/hr) = 8.021 q(U.S. gpm) x ρ(lb/ft^3)
w(lb/hr) = 500 q(U.S. gpm of water at 70°F or less)

In the following:
STP (standard conditions) refers to 60°F, 14.73 psia
NTP (normal conditions) refers to 0°F, 1.0135 bar abs

$$G_g = \frac{\text{molecular weight of gas}}{\text{molecular weight of air}} = \frac{M_w}{28.96}$$

w(lb/hr) = 60 q(scfm of gas) x ρ(lb/ft^3) at STP
w(lb/hr) = q(scfh of gas) x ρ(lb/ft^3) at STP
w(lb/hr) = 4.588 q(scfm of gas) x G_g
w(lb/hr) = 0.07646 q(scfh of gas) x G_g
w(lb/hr) = 3186 q(MMscfd of gas) x G_g
Mmscfd = millions of standard cubic feet per day
w(kg/hr) = q(normal m^3/hr of gas) x ρ(kg/m^3 at NTP)
w(kg/hr) = 1.294 q(normal m3/hr of gas) x G_g

3. Edward Valve Design Standards and Features

Engineering and research efforts – both analytical and experimental – have contributed to innovative leadership by Edward Valves through the introduction or practical development of some major industrial valving features:

- Integral hardfaced seats in globe and angle valves to permit compact valve designs and to resist erosion and wear.

- Impactor handwheels and handles to permit tight shutoff of manually operated globe and angle valves.

- Body-guided globe and angle valve disks to minimize wear and ensure alignment with seats for tight sealing.

- Inclined-bonnet globe valves with streamlined flow passages to minimize pressure drop due to flow.

- Equalizers for large check and stop-check valves to ensure full lift at moderate flow rates and to prevent damage due to instability.

- Compact pressure-seal bonnet joints to eliminate massive bolted flanges on large, high-pressure valves:

  – First with wedge-shaped metal gaskets with soft coatings, optimized over more than four decades to provide tight sealing in most services.

  – Now, for the severest services, with composite gaskets using flexible graphite and special anti-extrusion rings to assure tight sealing, even with severe temperature transients – overcomes need for field re-tightening and eases disassembly for maintenance.

- Optimized stem-packing chambers and packing-material combinations to ensure tight stem sealing:

  – First with asbestos-based materials and then with asbestos-free materials.

- Hermetically sealed globe valves with seal-welded diaphragm stem seals to prevent stem leakage in critical applications, including nuclear.

- Gate valves with flexible double-wedge construction to ensure tight sealing at both low and high pressures and to prevent sticking difficulties when opening.

- Qualified stored-energy actuators for quick-closing valves in safety-related nuclear-plant applications – and qualified valve-actuator combinations that are used in main-steam isolation service throughout the world.

Edward valve expertise, acquired over more than 85 years, is shared with national and international codes-and-standards committees and other technical societies and groups whose activities influence industrial valves. This cooperation has included participation in the development of every issue of ASME/ANSI B16.34 as well as most issues of ASME/ANSI B16.5 (Pipe Flanges and Flanged Fittings), which applied to steel valves before ASME/ANSI B16.34 was first issued in 1973. Edward Valves representatives have also been active in preparation of ISO (International Standards Organization) standards. In addition, Edward representatives have participated where appropriate with trade organizations such as EPRI, INPO and various nuclear power-plant owners' groups in addressing valve issues.

3.1 Codes and Standards

Edward valves are designed, rated, manufactured and tested in accordance with the following standards where applicable:

- ASME B16.34-1996 – Valves: flanged, threaded and welding end.

- ASME/ANSI B16.10-1992 – Face-to-face and end-to-end dimensions of valves.

- ASME B16.11 – Forged Fittings, Socket-welding and Threaded.

- ASME Boiler and Pressure-Vessel Code – Applicable sections including Nuclear Section III.

- ASME and ASTM Material Specifications – Applicable sections.

- MSS Standard Practices – Where appropriate: Edward sealability acceptance criteria are equal to or better than those in MSS SP-61.

Users should note that ASME/ANSI B16.34-1996 has a much broader scope than the previous editions. While this standard previously covered only flanged-end and butt welding-end valves, the 1988 edition covered socket welding-end and threaded-end valves as well. With this revision, the standard now addresses practically all types, materials and end configurations of valves commonly used in pressure-piping systems. All Edward valves in this catalog with a listed class number (e.g. Class 1500) comply with ASME B16.34.

In addition to the standards listed, special requirements such as those of API and NACE are considered on application.

3.2 Pressure Ratings

Edward valve-pressure ratings are tabulated in pressure-versus-temperature format. The temperatures range from -20°F (-29°C) to the maximum temperature permitted for each specific design and pressure-boundary material. Typically, pressure ratings decrease with increasing temperature, approximately in proportion to decreases in material strength.

Valves in this catalog with a listed class number are rated in accordance with ASME B16.34-1996. This standard establishes allowable working pressure ratings for each class number and material. These ratings also vary with class definitions as described below.

Standard Class (Ref: Paragraph 2.1.1 of ASME B16.34-1996) – These lowest ratings apply to all flanged-end valves as well as any threaded-end or welding-end valves that do not meet the requirements for other classes. Typically, ratings for these valves are consistent with ratings listed for flanges and flanged fittings of similar materials in ASME/ANSI B16.5-1988.

Special Class (Ref: Paragraph 2.1.2 of ASME B16.34-1996) – These ratings apply to threaded-end or welding-end valves which meet all requirements for a Standard Class rating and in addition meet special nondestructive examination (NDE) requirements. Valve bodies and bonnets are examined by volumetric and surface examination methods and upgraded as required. Pressure ratings for Special Class valves are higher than those for Standard Class valves (particularly at elevated temperatures) because of the improved assurance of soundness of pressure boundaries and because they are not subject to the limitations of flanged and gasketed end joints.

Limited Class (Ref: Paragraph 2.1.3 of ASME B16.34-1996) – These ratings apply only to threaded-end or welding-end valves in sizes 2-1/2 and smaller, with generally cylindrical, internal-wetted pressure boundaries. Limited Class valves meet all requirements for Standard Class valves, and body designs must also satisfy special reinforcement rules to compensate for irregularities in shape. Typically, the regions of minimum wall thickness in these valves are very localized, so minor plasticity in such regions at high temperature will not adversely affect valve geometry. Pressure ratings for Limited Class valves are the same as those for Special Class valves at lower temperatures, but Limited Class ratings are higher at very high temperatures [above 900°F (482°C) for ferritic steels and above 1050° (565°C) for austenitic steels].

3. Edward Valve Design Standards and Features (con't.)

It should be understood that flanged-end valves can be supplied only as Standard Class valves with numerically even pressure-class designations (300, 600, 900, 1500, 2500), for consistency with mating flanges in piping systems. Threaded-end or welding-end valves can be supplied with the same designations or as Class 4500 (for which there is no standard for flanged-end connections). In addition, threaded-end or welding-end valves can be furnished with intermediate ratings or class designations (ref: paragraph 2.1.4 of ASME B16.34-1996), up to Class 2500 for threaded ends and up to Class 4500 for welding-ends. For example, Class 2680 welding-end Univalves, can be applied in superheater-drain applications that could not be satisfied with a Class 2500 valve rating.

Series or CWP

A few valves in this catalog with "Series" or "CWP" designations are designed, rated, manufactured and tested to Edward proprietary standards. These valve designs, qualified by decades of successful field performance, will provide safe and reliable service in applications where an ASME/ANSI rating is not required by a piping code or other specifications.

These valve designs and ratings are generally, but not completely, in conformance with recognized national standards (e.g., some employ high-strength materials not listed in standards). These valves have a history of excellent performance and safety, and they may be applied with confidence in applications where ASME/ANSI ratings are not required.

Notes:

1. While Edward cast-steel valves described in this catalog have even listed ratings (e.g., 1500), many designs provide more wall thickness than required in critical areas. Accordingly, welding-end valves can often be offered with intermediate ratings (ref: Paragraph 6.1.4 of ASME B16.34-1996) moderately higher than the nominal class ratings. With appropriate revisions to testing procedures, this can allow somewhat higher pressure ratings than those listed in the tabulations. Consult Edward Valves and provide information on specific required design pressure and temperature conditions.

2. Pressure ratings for carbon steel (A105 and A216 WCB) valves are tabulated for temperatures through 1000°F (538°C), which is consistent with ASME B16.34-1996. As noted in that standard, these materials are permissible but not recommended for prolonged usage at above about 800°F (427°C). This precaution is related to the possibility that carbides in carbon steel may be converted to graphite.

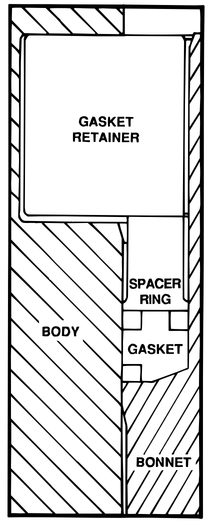
3. Other codes or standards applicable to piping systems may be more restrictive than ASME B16.34-1996 in limiting allowable pressures for valves. For example, ASME B31.1-1995 (Power Piping) does not permit use of carbon steel (A105 and A216 WCB) at design temperatures above 800°F (427°C). Users must consider all codes or regulations applicable to their systems in selecting Edward valves.

4. The maximum tabulated temperatures at which pressure ratings are given for Edward valves are in some cases less than the maximum temperatures given in ASME B16.34-1996 for valves of the same material. The maximum tabulated temperatures in this catalog may reflect limitations of materials used for other valve parts (e.g., stems). Use of Edward valves at temperatures above the maximum tabulated values may result in degradation and is not recommended.

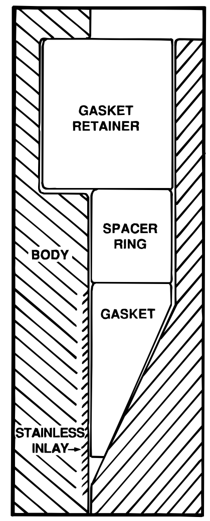
3.3 Pressure-Seal Construction

The time-proven Edward pressure-seal bonnet seals more effectively as pressure increases, because the pressure forces the sealing elements into closer contact. Metal pressure-seal gaskets with soft plating employ optimum contact angles and materials for each applicable valve type, size and pressure-class rating. The gaskets yield initially under bolting load and then under pressure, to provide excellent sealing contact.

Newest designs for highest pressure/temperature services employ improved composite pressure-seal gaskets with flexible graphite rings. Edward leadership in proof-testing of flexible graphite stem packings clearly showed the superior sealing characteristics of this material, and continued research led to the development of a test-proven bonnet closure that provides highest sealing integrity. The composite pressure-seal provides excellent sealing at *low and high pressures*, even under severe pressure/temperature transients. It provides easier disassembly for maintenance, seals over minor scratches and does not depend on re-tightening under pressure after re-assembly.



Composite Pressure Seal Construction



Typical Pressure Seal Construction

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3. Edward Valve Design Standards and Features (con't.)

3.4 Hardfacing

Integrity of seating surfaces on bodies, wedges and disks in gate, globe, and check valves is essential for tight shutoff. Valve body seats must be hardfaced, and wedges and disks must either be hardfaced or made from an equivalent base material.

The standard seating material for most Edward valves is cobalt-based Stellite 21®, which has excellent mechanical properties and an exceptional performance history. As compared to Stellite 6®, which was used in many early Edward valves and is still used in many competitive valves, Stellite 21® is more ductile and impact resistant. These properties provide superior resistance to cracking of valve seating surfaces in service.

Stellite 21 is used either as a complete part made from a casting (as in Univalve® disks and small Equiwedge® gate valve wedges) or as a welded hardsurfacing deposit. Depending on valve size and type, hardsurfacing material is applied by a process that assures highest integrity (PTA, MIG, etc.).

While the as-deposited (or as-cast) hardness of Stellite 21 is somewhat lower than that of Stellite 6, Stellite 21 has a work-hardening coefficient that is five times that of Stellite 6. This provides essentially equivalent hardness after machining, grinding, and exposure to initial seating stresses. In addition, low friction coefficients attainable with Stellite 21 provide valuable margins in assuring valve operation with reasonable effort or actuator sizing.

The properties of Stellite 21 also provide an advantage to the user long after a valve leaves the Edward plant. If a large valve seat is severely damaged in a localized area, as may occur due to closing on foreign objects, the seat may be repaired locally and refinished, in such cases, where a valve cannot be adequately preheated before welding, a Stellite 6 seat may crack during the repair process – requiring either removal of the valve from the line or in situ removal replacement of the complete seat.

Some Edward valves have used solid disks made of hardened ASTM A-565 Grade 616 or 615 stainless steel. This corrosion-resistant alloy has been proven in seating and erosion tests and in service. This material can be furnished in certain valves for nuclear-plant services where reduced cobalt is desirable. Similar iron-base trim materials are used in production of certain standard valves. Extensive research on other cobalt-free valve trim materials has also identified other alloys which provide good performance under many service conditions. Consult Edward Valves about any special trim requirements.

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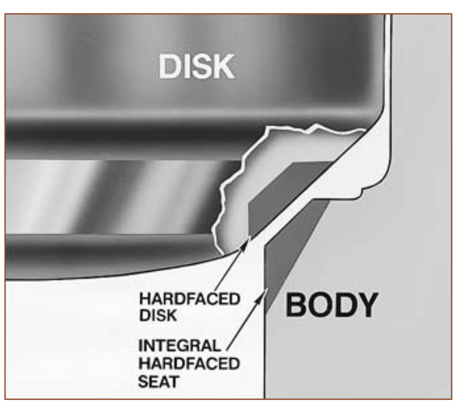
3.5 Valve-Stem Packing

Stem sealing is an extremely important valve performance feature, since seal leakage can represent energy loss, a loss of product and a potential environmental or safety hazard. Consequently, Edward stop and stop-check valves employ stem packings that have been qualified by extensive testing.

The search for improved sealing performance was a primary reason for seeking out new stem-packing materials to replace asbestos-based packings. The demand of many valve users to discontinue use of asbestos due to health risks was an important secondary reason. Since there are no simple laboratory tests that will predict sealing performance based on measurable properties of packing materials, hundreds of tests have been necessary with various packings in valves or valve mockups.

Some packings required frequent adjustments due to wear, extrusion or breakdown, and some could not be made to seal at all after relatively brief testing. All standard Edward stop and stop-check valves now employ flexible graphite packing which provides excellent stem sealing. However, the key to its success involves retaining the graphitic material with special, braided end rings to prevent extrusion. Various end rings are used, depending on the valve pressure class and expected service-temperature range. All Edward valves assembled since January 1986 have been asbestos-free.

See V-REP 86-2 for more information.



4. Miscellaneous Technical Data

4.1 Edward Technical Articles

NUMBER	TITLE
V-REP 74-3	A Hermetically Sealed Valve for Nuclear Power Plant Service
V-REP 75-5	Development of the Edward Equiwedge Gate Valve
V-REP 78-3	Nuclear Containment of Postulated Feedwater Linebreak
V-REP 78-4	Quick-Closing Isolation Valves – The Equiwedge Alternative
V-REP 79-4	Valve Clamp Ring Stress Analysis
V-REP 80-1	Univalve Evolution – Another Advance
V-REP 80-3	The Type A Stored Energy Actuator – Development and Qualification
V-REP 81-1	Model for Check Valve/Feedwater System Waterhammer Analysis
V-REP 81-2	Minimizing Use of Cobalt and Strategic Materials in Valves
V-REP 82-1	Asbestos-Free Stem Packing for High Temperature Valves
V-REP 82-2	Quick-Closing Equiwedge Isolation Valves Global Qualification
V-REP 84-1	Avoiding Aluminum Nitride Embrittlement in Steel Castings for Valve Components
V-REP 85-2	Quick Closing Equiwedge Isolation Valves Global Qualification
V-REP 86-2	Tests of Asbestos-Free Stem Packings for Valves for Elevated Temperature Service
V-REP 90-1	Design Basis Qualification of Equiwedge Gate Valves for Safety-Related MOV Applications
V-REP 90-2	Flow Performance, Stability and Sealability of Piston Lift and Tilting Disk Check Valves
V-REP 90-3	Edward Cast Steel, Pressure-Seal Valves: Research and Development
V-REP 91-1	Pressure Locking and Overpressurization of Double Seated Valves
V-REP 92-1	Check and Stop-Check Valves for High Turndown Applications
V-REP 93-1	PressurCombo
V-REP 95-1	Hermavalve-A Zero Emissions Valve

Copies of the above Technical Articles are available upon request.

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4.2 Sources for Additional Information

For further guidance on selection, shipping and storage, installation, operation, and maintenance of valves, readers are referred to the following documents:

MSS Valve User Guide
MSS SP-92

Available from:
Manufacturers Standardization Society of the Valve and Fittings Industry, Inc.
127 Park Street N.E.
Vienna, Virginia 22180

Aging and Service Wear of Check Valves Used in Engineering Safety-Feature Systems of Nuclear PowerPlants

Nureg/CR-4302
Volume 1
Ornl-6193/V1
Volume 1. Operating Experience and Failure Identification

Available from:
Superintendent of Documents
U.S. Government Printing Office
P.O. Box 37082
Washington, D.C. 20013-7982

And from:
National Technical Information Service
Springfield, Virginia 22161
EPRI Report No. NP 5479
Application Guidelines for Check Valves in Nuclear Power Plants

Available from:
Electric Power Research Institute
Research Reports Center
P.O. Box 50490
Palo Alto, CA 94303