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Flow Performance, Stability, and Sealability of Piston-Lift and Tilting-Disk Check Valves

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Flow Performance , Stability, And Sealability of Piston-Lift And Tilting-Disk Check Valves

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ABSTRACT

In 1987, Edward Valves undertook a major test program on check valves to supplement a database covering tests conducted over the previous forty years. Results were combined with mathematical models to develop sizing parameters which can be used to predict the performance of large and small Edward check valves in specific applications. Check valves may display unstable performance if they are oversized, and resulting wear may damage the valve and possibly cause problems with other equipment. Application of the new sizing parameters can help to assure trouble-free check valve performance in new applications, and it can help to predict problems before they occur in existing applications. This paper focuses on the Edward study of check valve performance and on the application of the mathematical valve sizing model that was developed from this study.

Introduction

Of the various types of valves, check valves generally receive the least attention. So long as a check valve performs its basic functions of allowing forward flow and limiting reverse flow, it is considered to be a passive component requiring no further concern.

A check valve must open and close in response to flow direction, but it relies on a relatively primitive balance between hydrodynamic forces and gravity for control of its operation. In spite of these limitations and neglected maintenance, check valves produce relatively few problems. However, when problems do occur, they may be serious.

Edward has conducted numerous flow test programs on its check valves, starting in the 1940s, and has published extensive engineering data derived from these tests over the years[1,2,3]. While proper use of this information has generally resulted in good check valve performance, Edward concluded that additional testing would provide a more reliable basis for sizing and applying check valves. A new program was undertaken in 1987 to improve upon the existing data.

The results of Edward's research, which appear in the Technical section of Edward catalog EV-100[4] and are summarized in this paper, provide general application information, useful for fossil-fuel power plants, process facilities, and other services. This information originally appeared, in June 1988, in Edward's *Check Valve Application Manual and User's Guide*[5], which was incorporated into the Edward catalog in September 1990. It was also presented at an EPRI symposium in October 1988[6].

At about the same time that Edward undertook its program, the Electric Power Research Institute (EPRI) initiated check valve research of its own, and in January 1988 published the results in an extensive report[7], which focused on application guidelines for nuclear services. The EPRI report addresses some subjects not covered by Edward research, and it provides significant guidelines for swing check valves. The information contained in catalog EV-100 is complementary with EPRI data, and is intended to fill in gaps related specifically to Edward ball, piston-lift, and tilting-disk check valves. In addition, Edward provides an alternative sizing method to that given in EPRI Application Guideline 2.1.1. However, the two methods do not conflict. A comparison of the two methods is given below, under the heading "Relationship With EPRI Check Valve Guidelines." Together, the Edward and EPRI guidelines cover the majority of check valves commonly in use.

In the Edward test program, ten size 1 and 2 ball and piston-lift check valves were tested, with emphasis on the inclined-bonnet piston-lift check valves that are widely used in power plants. In addition, Edward tested five size 4, 8, and 10 piston-lift check valves (horizontal, angle, and inclined-bonnet types). Finally, the test program also included one size 10 tilting-disk check valve.

The objectives of the program were twofold: first, to analyze the performance of Edward check valves; second, to gather the empirical data needed for a mathematical model to be used in sizing these valves for particular applications.

The test program considered three primary questions, the first two pertaining to forward flow and the third pertaining to flow in the reverse direction:

1. Will the valve open fully at normal flow, or will it assume a partial opening in balance with the disk weight?
2. If the valve is partially open, will it be stable or will it flutter and eventually fail?
3. When a flow reversal occurs, will the valve close and seal well at low differential pressure, or will it require a high differential in order to seal?

These questions are reflected in the three major phases of the tests, as described below.

Basic Flow Performance Tests

Edward subjected all of the valves to basic flow performance tests in straight pipe runs in circulating-water test loops. *Figure 1* shows a schematic diagram of the larger loop, which had a flow capacity of over 4000 U.S. gallons per minute. In these tests, 10 to 12 diameters of straight pipe were provided upstream and downstream of the check valves to minimize flow disturbances. The main emphasis of this testing was to collect data with valves partially open and to identify the flow rate required to open valves fully. Measurements included flow rate, upstream pressure and temperature, valve opening (lift), and valve pressure drop. In addition, the test included noise observations to identify the onset of cavitation.

Results of these tests established the relationships between the following sets of attributes:

- Flow coefficient (C_V) and lift
- Lift and flow rate
- Pressure drop and flow rate

These tests also established incipient cavitation and liquid pressure recovery coefficients for most valves.

An interesting observation from these water tests was that the check elements in most valves were relatively stable at all lifts and flow rates, even when cavitation occurred. Most valves displayed inherent stability, even at very small openings. However, the flow rates in the test loops (equipped with centrifugal pumps) were very steady; unsteady liquid flow might produce different results.

In contrast to the relative stability in liquid tests, limited testing with air flow demonstrated a trend toward disk instability at lifts under approximately 10 percent and at pressures below 30 psi, even with the check valve mounted in straight pipe. Since instability at small openings can produce repetitive disk-seat impact, such operation can be damaging and should be avoided.

Flow Disturbance Tests

Following the flow performance tests, Edward retested some of the valves with upstream flow disturbances. Single and double (out of plane) elbows were installed immediately upstream of the valves, and a throttled butterfly valve was installed immediately upstream and at various distances away from the inlet of selected valves. Edward conducted these tests using the same procedures employed in the basic flow performance tests so that performance with and without disturbances could be compared. Figures 2 and 3 illustrate two of these unusual test installations.

Table 1 summarizes the results of the flow disturbance tests. Elbows immediately upstream affected the stability of some check valves, but had no discernible effect on others. While the observed effects appeared minor, long-term effects of even

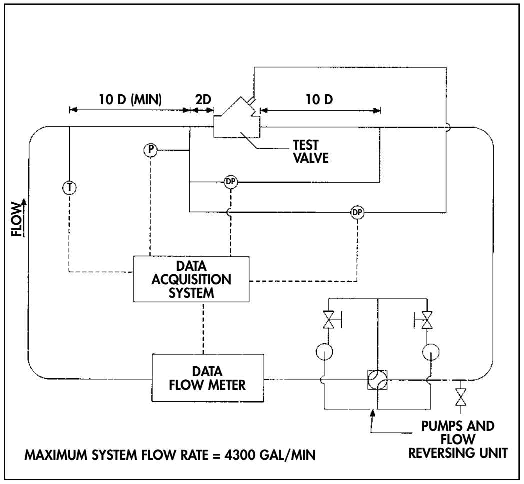

Figure 1: NPS 10 test loop and straight pipe flow test arrangement

minor disk flutter could include damage to seating or guiding surfaces. Table 1 also shows that the throttled butterfly valve had a more distinct effect on check valve performance and stability than elbows did. Effects included increased disk flutter and reduced valve opening at a given flow; in some cases, full check valve opening could not be achieved at any attainable flow. With the butterfly valve moved five diameters upstream of the check valve, adverse effects decreased but were not eliminated. Normal performance was restored with the butterfly 11 diameters upstream.

Special Seat Tightness Tests

Standard seat tightness tests only consider sealing performance under high differential pressures. However, some applications — e.g., various nuclear services — require tight sealing under very low differential pressures, and various types of metal-seated check valves have encountered problems. Edward designed its special tests to evaluate sealing under all these conditions and performed this test on most of the valves included in the overall test program.

In this test, Edward gradually increased the differential pressure from 0 to 110 percent of the cold working pressure of the valve, then reduced it back to 0. Generally, with increasing pressure, relatively high leakage was observed up to a threshold pressure at which the leakage rate suddenly decreased. Edward engineers have concluded that that this threshold represents the point where forces due to pressure are sufficient to shift the closure element into good metal-to-metal contact with the body seat. In tests of size 4 and larger piston-lift and tilting-disk check valves, the threshold pressure was less than 50 psi. Small forged-steel ball and piston-lift check valves were less consistent, sometimes seating at less than 50 psi and sometimes requiring 250 psi or more.

Seat tightness data, both from these tests and past research, have allowed Edward to produce tightness guidelines for the company's standard check valves. Generally, metal-seated check valves should not be expected to seal well at reversed differential pressures less than 50 psi. Some larger valves show sealing differential thresholds as low as 5 psi, but some small valves require much more than 50 psi to establish full metal-to-metal seating contact. Below the threshold, leakage rates may be relatively large and could lead to rapid leak-down from a tank with just a gravity or elevation head to seat the valve.

As a general rule, metal seat check valves should not be relied upon for isolation at low differential pressures. Leakage rates are difficult to predict. Once seated by high pressure, most check valves remain seated to lower pressures than would be required for initial seating as pressure increases. However, this is also difficult to predict.

Where tight shutoff is required, a stop-check valve is often a good compromise, because it can be seated with a handwheel or actuator to establish a seal when flow is not required. Obviously, the stem must be moved to the open position to restore normal check valve flow capability.

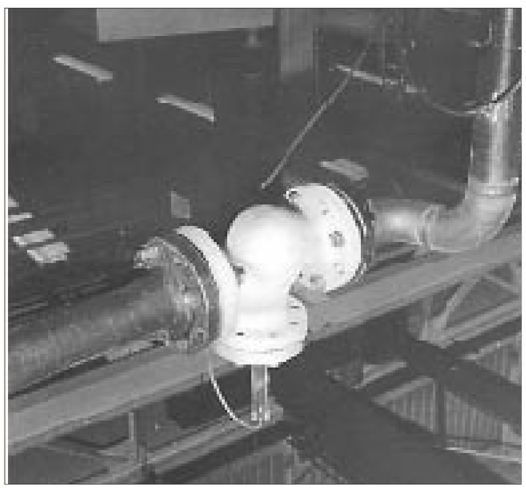

Figure 2: Flow disturbance test installation: A size 4, Class 600, 901 bonnet piston-lift check valve with two upstream elbows (out of plane).

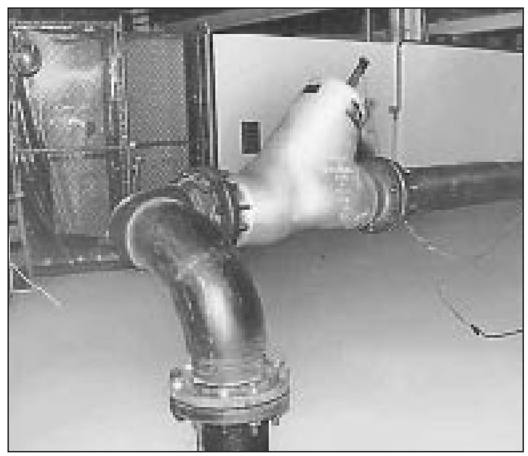

Figure 3: Another example flow disturbance test installation with two upstream elbows, this time employed with a size 10, Class 1500, Flite-Flow®" inclined-bonnet piston-lift check valve.

"Soft seats" are sometimes considered for low pressure sealing in check valves, and they can provide a solution for some applications. However, the selection of a proper plastic or elastomeric soft-seating material requires careful consideration of pressure, temperature, and line fluid effects. Some materials will perform well at high temperatures in air but not in water. Others are quickly destroyed by hydrocarbons. Thus, there are no "general purpose" soft-seated check valves.

Performance Predictions Based On Mathematical Models

While the test program described in this paper involved 16 different check valves, and data were available from prior tests of other valves, the Edward product line includes over 150 basically different configurations of standard check valves. The challenge was to define the performance of the entire product line using test data from a 10 percent sample. This information could then be used to size valves for new installations, as well as evaluate the performance of valves in existing services.

Prior to this project, Edward based check valve sizing guidelines on the principle of geometric similarity of basic valve types. This model included bias factors that allowed for valve size, based on the fact that larger valves have larger disks, which have a higher ratio of weight to surface area. While these guidelines generally gave good results, Edward recognized that individual valves have design characteristics that depart from ideal similarity. For example, disk weight may vary with pressure class as well as size (and also with material form, e.g., casting versus forging). For reasons such as these, Edward engineers decided that it was necessary to develop a mathematical model which could treat individual design characteristics as well as possible.

To develop the required model, Edward derived a set of equations to describe the hydrodynamic and gravitational forces acting on the check element, as functions of flow and disk lift. Test results provided the empirical coefficients needed for the hydrodynamic force equations. Experimental data also provided the coefficients used to predict check element position (lift) when the valve is less than fully open.

In addition, Edward engineers used the test program to verify the theoretical expectations of the mathematical model. For instance, tests of both globe and angle piston-lift check valves in two different sizes provided experimental verification of the model's scaling law, which allows empirical coefficients to be applied to valve sizes not tested. Special tests of globe and angle check valves with aluminum disks and lead-filled disks confirmed the model's ability to accurately predict the effect of check element weight on flow performance.

With the mathematical model substantially verified, the remaining task was to apply it to the entire Edward check valve product line. In the case of small forged-steel check (and stop-check) valves, Edward employed a simplified model. However, for the larger cast-steel piston-lift check valves, Edward engineers calculated the weight of every disk piston by means of computer-aided design, and then used these weights to calculate coefficients that were unique for each valve.

When applied to a particular valve, the mathematical model provides the flow rate required to open the valve to a specific lift, expressed as a percent of fully open. Once the flow rate is determined, one can calculate the pressure drop across the valve by using the flow coefficient of the valve at the lift of interest. Applying the model at 100 percent open allows calculation of the minimum flow required and the minimum pressure drop across the valve for full-open operation.

The primary coefficient describing each check (or stop-check) valve is a sizing parameter for full lift. In general, the sizing parameter is defined as follows:

$$SP = \frac{w}{\sqrt{\rho}}$$

Where SP = valve sizing parameter

w = weight flow rate (lb/hr)

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

Using this method, systems design engineers and valve users need only calculate the sizing parameter (SP) for their particular applications, since, in the typical problem statement, weight flow rate and fluid density are known quantities. Engineers and users can then compare this parameter with the sizing parameter required for full lift (SP_{FL}) for a specific check valve, as listed in the Edward catalog.

Edward full-lift sizing parameters were calculated assuming that the line fluid is room-temperature water, but the values are sufficiently accurate for common liquids, including water at typical feedwater temperatures. The Edward catalog also provides correction factors that allow application of the data to lighter fluids, such as gases or steam.

Typical ways the sizing parameter method are applied are:

- **Selecting a check valve size for a new application:** Choose a valve with $SP_{FL}<SP$ to assure full check valve opening at the flow rate of interest. If the valve size selected is less than the line size desired for other reasons (e.g., velocity limitation), the Edward catalog includes procedures for estimating the effects of upstream and downstream pipe reducers.
- **Evaluating an existing, installed valve:** Compare the SP for the application with the published SP_{FL} for the valve in question. If $SP<SP_{FL}$, the valve is not fully open. Since this may lead to problems, Edward provides other procedures for predicting actual valve opening and determining the acceptability of valve performance. These procedures are described briefly below, and are covered at length in the Edward catalog.

  For evaluating check valves operating at less than full opening, Edward employs the following ratios:

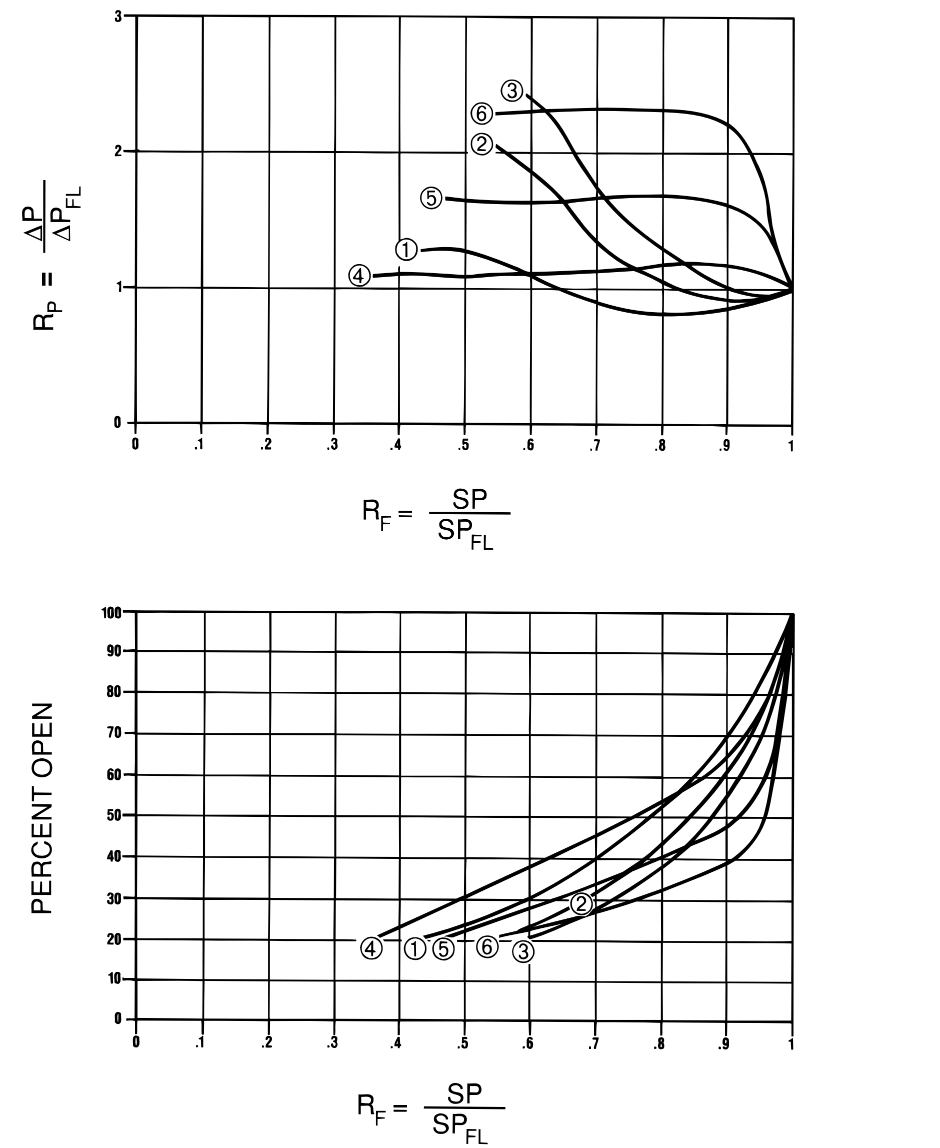

Figure 4: Edward cast-steel, globe, piston-lift check valve performance curves

$$R_F = \frac{SP}{SP_{FL}} \quad R_P = \frac{\Delta P}{\Delta P_{FL}}$$

Where R_F = the normalized sizing parameter
R_P = the normalized pressure drop
Δ_P = actual pressure drop
ΔP_{FL} = pressure drop at full opening

Graphs provide "percent open" predictions as a function of the normalized sizing parameter. Normalized pressure drop is also presented in graphs as another function of normalized sizing parameter. Figure 4, pg. 6, shows just one set of graphs illustrating these functions. The index numbers on these graphs represent specific groups of valves, and tables in the Edward catalog indicate which curves apply to which specific valves.

Other data presented in the catalog include coefficients for incipient cavitation and other coefficients for describing choked liquid and gas flows. While these conditions do not arise often in check valves, they may be important in some cases. For example, if an existing valve is replaced with one of a smaller size to assure full opening at a low flow condition, cavitation might occur at another high flow condition in a hot feedwater line.

Application To Specific Problems

The Technical section of the Edward catalog provides flow performance calculation methods in two parts. The first part, Basic Calculations, deals with valves in non-cavitating, non-choked flow applications. The second part, Corrections Required With Large Pressure Drops, addresses problems that may involve cavitating and/or choked flow. This second part is more complicated, but is not usually required for check (or stop-check) valves.

The two examples that follow serve to illustrate how one can use Edward procedures to size check valves and resolve problems that arise in specific check valve applications.

Example 1

Problem: Size, a cast-steel, globe, piston-lift check valve for the following boiler feedwater conditions and determine the pressure drop:

Design Conditions:	1715 psig at 275°F
Operating Conditions:	1500 psig at 275°F
Flow Rate:	635,400 lb/hr (water)

Solution:

1. Based on the design conditions, a Class 900 valve is required (per ASME/ANSI B16.34-1988).
2. Determine fluid density at the operating conditions (using Figure 22A of the catalog, reproduced as Table 2 of this paper, or other appropriate reference source)

   ρ = 58.1 lb/ft^3

3. Calculate sizing parameter for the application:

$$SP = \frac{w}{\sqrt{\rho}} = \frac{635,400}{\sqrt{58.1}} = 83,360$$

4. From Table 10 of the catalog (the relevant section is provided here in Table 3), select the largest Class 90 check valve with a sizing parameter for full lift (SP_{FL}) less than 83,360 to assure full valve opening. This is size 8 valve (SP_{FL} = 69,500). Note that the valve flow coefficient (C_V) is 910.

5. Calculate the pressure drop from equation 1C of the catalog:

$$\Delta P = \frac{1}{\rho}\left(\frac{w}{63.3\, F_P C_V}\right)^2$$

Where F_P = piping geometry correction factor

Assume separate calculations show that NPS 8 pipe is satisfactory, so F_P = 1.0.

$$\Delta P = \frac{1}{(58.1)}\left(\frac{635,400}{(63.3)(1.0)(910)}\right)^2 = 2.1\text{psi}$$

Note: *Usually the largest valve that will be fully open is the most desirable, to minimize both pressure drop and pumping costs in applications involving normally open flowing conditions. If this application involves infrequent operation, a size 6 valve might be used to minimize investment cost. Similar calculations would show that the size 6 valve would also be fully open and could be used if its pressure drop is acceptable.*

Example 2

Problem: Due to a change in feedwater system design and plant operating schedules, the valve selected in Example 1 must operate at a higher flow and temperature during full-power operation, and it must also operate at a reduced flow during limited hot-standby operating conditions. Evaluate valve operation at the following conditions:

Full power: 833,000 lb/hr at 1750 psig and 455°F

Hot standby: 400,000 lb/hr at 1750 psig and 455°F

Solution:

1. From the preceding example and Table 10 of the catalog (Table 3 of this paper):

   C_V = 910
   SP_{FL} = 69,500
   ΔP_{FL} = 1.5 psi

2. Determine fluid density, using Figure 22A of the catalog (or Table 2 of this paper):

   ρ = 51.2 lb/ft^3

3. Calculate sizing parameter at the full-power condition:

$$SP = \frac{w}{\sqrt{\rho}} = \frac{833{,}000}{\sqrt{51.2}} = 116{,}400$$

   Since SP > SP_{FL}, the check valve will be fully open.

4. Calculate pressure drop at the full-power condition:

$$\Delta P = \frac{1}{\rho}\left(\frac{w}{63.3\, F_P C_V}\right)^2 =$$

$$\frac{1}{(52.1)}\left(\frac{833{,}000}{(63.3)(1.0)(910)}\right)^2 = 4.1 \text{ psi}$$

5. Calculate sizing parameter at the hot standby condition:

$$SP = \frac{w}{\sqrt{\rho}} = \frac{400{,}000}{\sqrt{51.2}} = 55{,}900$$

   Since SP<SP_{FL}, the check valve will not be fully open.

6. In order to predict the valve opening and pressure drop at the hot-standby condition first calculate the normalized sizing parameter:

$$R_F = \frac{SP}{SP_{FL}} = \frac{55{,}900}{69{,}500} = 0.80$$

   Then, using the curves labeled "2" on Figures 17-A and 17-B in the catalog (reproduced as Figure 4 in this paper), find that R_P = 1.0 and "Percent Open" = 40. Finally, to determine the actual pressure drop, you use the following equation:

$$\Delta P = (R_P)(\Delta P_{FL}) = (1.0)(1.5) = 1.5 \text{ psi}$$

Thus, the methods presented in the catalog predict that the valve would be about 40 percent open under the hot-standby condition, with a pressure drop of approximately 1.5 psi. Other sections of the catalog may be reviewed for guidelines regarding the acceptability of this operating condition. Since this valve would be more than 25 percent open and this would be a limited part-time operating mode, the catalog suggests that this condition should be satisfactory. However, this valve should be monitored or inspected periodically for signs of flutter or Wear.

Relationship With EPRI Check Valve Guidelines

EPRI published its *Application Guidelines for Check Valves in Nuclear Power Plants* in January 1988. This report was stimulated by an Institute of Nuclear Power Operations study[8], which documented serious problems with check valves. As discussed above, that report focused on some different areas than covered by Edward in catalog EV-100 and summarized in this paper. The EPRI report emphasizes swing check valves, while the Edward catalog covers the specific ball, piston-lift, and tilting-disk check valves manufactured by Edward. As noted, the sizing method used in the Edward catalog is a non-conflicting alternative to EPRI Application Guideline 2.1.1.

The EPRI sizing method uses a minimum *velocity* for full opening of check valves and a "C" constant for determining this velocity. In Edward's experience, the majority of users express flow in terms of weight flow rather than velocity, so the sizing parameter is based on weight flow, expressed in lb/hr. However, the, Edward catalog provides "C" constants for all check valves for those who prefer to use the EPRI method.

Edward research indicates that EPRI Application Guideline 2.3.1, which was based on tests of swing check valves, is generally applicable to piston-lift and tilting-disk check valves downstream from flow disturbances. A comparison of results of very similar tests suggests that piston-lift and tilting-disk check valves may be influenced less by upstream elbows than some swing check valves, but any check valve immediately downstream from an elbow is a potential source of trouble, especially if it is not fully open.

Tests of check valves downstream from throttled butterfly valves show that all check valve types are adversely affected when located within a distance of 10 pipe diameters. Since both the EPRI and Edward tests were conducted in relatively low pressure loops, it is possible that throttled control valves operating with very high pressure drops might produce severe disturbances for even greater distances (particularly if there is significant cavitation). Therefore, the Edward catalog suggests minimum installation distances even more than 10 diameters in such cases.

The EPRI guidelines and Edward catalog EV100 both provide valuable check valve application engineering support for any engineer involved either in selecting a valve for a new application or in troubleshooting an existing application.

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TABLE 1
EFFECTS OF UPSTREAM FLOW DISTURBANCES ON CHECK VALVE PERFORMANCE
(Compared To Performance In Straight Pipe)

Valve Size & Type	Single* Elbow 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	Higher lift for same flow; disk flutter at lower lifts**	Higher lift for same flow	—	—	—
Size 4 angle piston-lift	No effect	No effect	—	—	—
Size 4 90° bonnet piston-lift	Same, lower or higher lift for same flow	No effect	Disk flutter and chatter; failure to achieve full opening	—	—
Size 8 angle piston-lift	No effect	—	—	—	—
Size 8 90° bonnet piston-lift	Disk flutter at partial lift	—	—	—	—
Size 10 inclined-bonnet piston-lift	Same or lower lift for same flow; slight disk wobble	No effect	Failure to achieve full opening; disk flutter & chatter	Failure to achieve full opening	No effect
Size 10 tilting-disk	No effect	Minor flutter	Same, lower or higher lift for same flow; disk flutter & chatter	Minor flutter	No effect

*Tests were conducted with 901 elbows in the horizontal plane and in the vertical plane (with flow both from above and below).
**One size 2 valve exhibited flutter at lower lifts; another was stable.

TABLE 2
SATURATED WATER — TEMPERATURE, PRESSURE & DENSITY (U.S. UNITS)

Water Temp. °F	32	70	100	200	300	400	500	600	705
Vapor Pressure, P_V	0.09	0.36	0.95	22.5	67	247	681	1543	3206
Water Density, ρ	62.4	62.3	62.0	60.1	57.3	53.7	49.0	42.3	19.9

Pressure in psia, density in lb/ft³

TABLE 3 EDWARD CAST STEEL GLOBE VALVE FLOW COEFFICIENTS CLASS 900										
NPS	All Stop & Check Valves				Check Valve Coefficients					Perf. Curves Fig. 4
	C_V	F_L	X_T	K_i	d	ΔP_{co}	ΔP_{FL}	SP_{FL}	C	
3	110	0.96	0.60	0.10	2.87	0.92	1.5	8510	53	4
4	200	0.97	0.60		3.87	1.3	2.3	19,500	66	5
5	305	0.97	0.61		4.75	1.3	2.5	30,600	69	4
6	530	0.81	0.42	0.07	5.75	1.2	1.5	41,500	64	3
8	910	0.81	0.42		7.50	1.3	1.5	69,500	63	2
10	1400	0.81	0.42		9.37	1.6	1.8	119,000	69	1
12	2000	0.81	0.42		11.12	1.8	2.1	182,000	75	2
14	2400	0.81	0.42		12.25	1.6	1.9	211,000	72	2

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