APPLICATION OF BUTTERFLY VALVES FOR FREE DISCHARGE, MINIMUM PRESSURE DROP, AND FOR CHOKING CAVITATION

Butterfly valves are commonly used as control valves in applications where the pressure drops required of the valves are relatively low. Butterfly valves can be used in applications as either shutoff valves (on/off service) or as throttling valves (for flow or pressure control). As shutoff valves, butterfly valves offer excellent performance within the range of their pressure rating. Typical uses would include isolation of equipment, fill/drain systems, bypass systems, and other like applications where the only criteria for control of the flow/pressure is that it be on or off. Although butterfly valves have only a limited ability to control pressure or flow, they have been widely used as control valves because of the economics involved. The control capabilities of a butterfly valve can also be significantly improved by coupling it with an operator and electronic control package.

Butterfly valves allow high flows with relatively low pressure loss from the valves, and are typically used for flow control for valve openings from 30 to 70 degrees of full open. At valve openings greater than 70 degrees, the pressure loss of a butterfly valve is too low to produce any significant effect on flow or the energy loss of a flow system. Two special applications for a butterfly valve include the use of a valve for free discharge and the use of a butterfly valve for flashing or choking cavitation service. Free discharge typically produces high pipe velocities at moderate pressure drops, and choking cavitation typically produces high velocities with large pressure drops. The following paper also includes a review of flow coefficients and the application of non-line sized valves, or valves installed in larger diameter pipes with the aid of pipe reducers.

Butterfly Valves Types

The butterfly valves used in water works applications vary from traditional “resilient seated” butterfly valves to the more specialized AWWA valves as detailed in AWWA’s C504 standard. Regulations do not generally require the use of AWWA valves in water works service but most operators will specify AWWA valves for the majority of applications based on their experience with AWWA valves and their reputation for long service life.

Basic resilient seated valves are designed in accordance with industry standards such as MSS SP-67 and most frequently come in lugged and wafer configurations. They are considered by most to be a “general service” type of valve and as such, are usually restricted to use in the ancillary systems within water treatment and distribution systems. They are usually limited in size to around 48 inches in diameter. and pressure ratings for these valves are typically 75, 150, or 300 psi. The design of the valves with different ratings are very similar. The stem diameter, disc thickness, and disc to seat interference are the main features which differ as the rating changes. Flow characteristics and limits are not generally defined in the standards/specifications nor are they defined by manufacturers in most cases.

AWWA valves are considered by most users to be a more robust valve design and in most cases have greater body wall thicknesses, larger shaft diameters, and thicker discs than resilient seated valves with equivalent pressure ratings. In addition, AWWA valves would most frequently be supplied in a double flanged configuration rather than the lugged or wafer styles although all of these configurations are available in the marketplace. Currently the AWWA C504 standard defines three shutoff pressure ratings and two flow velocity sub-ratings. Pressure rating breaks are made at 25, 75, and 150 psi. Flow velocities are defined for the AWWA valves as specific breaks are made in the line velocities at 8 and 16 feet per second. An increasing number of applications however, are pushing beyond these traditional pressure/flow velocity limits with applications at 250 and 300 psi becoming common.

A third category of butterfly valve, the high performance butterfly valve, is occasionally utilized in water treatment and distribution applications. These valves are characterized by higher pressure and/or temperature ratings, different disc seating/sealing technology, and premium materials. These valves frequently follow the design guidelines outlined in MSS SP-68. The primary distinguishing design feature is a stem which is offset from the centerline of the pipe. This offset design generates a camming action which allows for different types of seating technologies when compared to resilient seated valves. These alternate seating technologies frequently incorporate advanced seal materials, such as PTFE, RTFE, and other advanced polymers which help the valve achieve higher pressure and temperature ratings. A high performance butterfly valve is therefore a butterfly valve which has extended the performance boundaries (in terms of pressure, temperature, and/or corrosion resistance) of traditional butterfly valves.

Butterfly Valve Characteristics

Within the quarter turn family of valves, ball valves are dominant in the sub-2 inch size category. Above 2 inches in size, butterfly valves become much more prevalent primarily for economic reasons. Butterfly valves exhibit a number of characteristics which make them especially well suited to use in general service type applications. Those characteristics include:

- Economical: Because butterfly valves can be designed in a physically compact package where the closure mechanism (disc) is partially contained by the system piping in the open position, they are very economic in terms of material usage. Since material costs are a significant portion of the product cost, the final valve is also very cost efficient.
- Compact: The compact package also makes for a valve which is comparatively light and therefore relatively easy to handle. The compactness lends the valves to installation in places where other valves will not easily fit while the lightness makes installation and maintenance relatively easy.
- Configuration: The variety of body styles in which butterfly valves are available lends the valves to a variety of installation configurations and preferences. Typical body style offerings include wafer, semi-lug, lugged, and flanged valves.
- Performance: Butterfly valves are generally good performing valves in addition to being

economical. Performance characteristics for most designs include bi-directional sealing, low pressure loss at high flow rates, simple operation, and reliable sealing.

· Actuation: A variety of actuation options exist for butterfly valves including handle operation (typically only valves 8” and smaller), manual gear operators, and a wide variety of power operators including pneumatic, electric, and hydraulic.

Butterfly Valve Seats

For butterfly valves where the seat is mounted in the valve body, four basic types of seat exist:

1. Boot seats are the traditional seat style. This type of seat is a separate component which typically has a “U” shaped cross section and completely lines the ID of the valve body. They are retained in the valve body by interlocking “dovetails” at either face of the valve body. The primary advantage of this type of seat is its replaceability but they are generally also available in a wider variety of materials.
2. Cartridge seats also completely line the ID of the valve body but are generally rigid seats by virtue of a “hoop” of rigid material encapsulated by a elastomer jacket. The resulting assembly forms a rigid cylindrical shaped seat. These type of seats are also replaceable and can offer advantages in vacuum applications.
3. Molded in seats are physically bonded to the valve body during the rubber molding process. This style of valve offers good performance characteristics but is generally more limited in material options and seat wear or damage is not repairable. The primary advantage of this type of seat is its pressure rating and its ability to be used in dead-end or end of line applications where no end flange is supporting the valve.
4. Mechanically retained seats are most common in high performance valves and in the water works industry where large diameter valves make the other seat styles described above impractical to manufacture. Seats are retained by a number of methods including mechanical metal fasteners and retention keys formed with advanced adhesives. A few valve designs in the industry mount the elastomeric seat on the disc edge. In this style of valve, mechanical seat retention is used almost exclusively.

The boot, cartridge, and molded in seats are most common in the smaller valve sizes (through roughly 24”) and define the range of seat configurations for “resilient seated” butterfly valves . The mechanically retained seats are typically only used on larger diameter (24” and larger) AWWA style valves and in high performance valves.

Seat materials for butterfly valves vary but the predominant elastomers utilized would be EPDM and NBR. Additionally, other elastomers in common use include CR (trade name Neoprene), SBR, and occasionally more exotic materials such as fluoroelastomers, silicones, or others. When high performance valves are utilized, thermoplastic seats are also frequently used. PTFE and RTFE are the most common materials but other material types are used for some applications.

In general terms, any of these materials described here are suitable in water service. The conditions which lead to a preference of one material over another are just temperature or other chemical constituents in the water. For example, EPDM gives excellent service in high temperature (up to approximately 300 degrees F) water and low grade steam. In this environment its performance is usually significantly superior to NBR which is generally restricted to somewhere in the 200 degree F neighborhood. EPDM is not, however, a good performer in environments where hydrocarbons are found while NBR is excellent in environments where there are hydrocarbons. Generalizations such as this about these material families can be made but the actual performance of the seat material varies widely even within the material family due to differences in the compounding of the material. Because of this, it is extremely important that application data be provided to the manufacturer in order for the best seat material recommendation to be made. The primary information that is needed to make the material selection is as follows: temperature (min. / max. and typical), operating cycle, and chemical makeup of the water including concentrations. From this information, a recommendation can usually be made or detailed questions can be generated to help finalize the selection.

Minimum Pressure Drop

A common misconception in sizing large butterfly valves is the need for a minimum pressure drop or large flow coefficient at full open. Most high performance (low pressure loss) butterfly valves have flow coefficients large enough that the need or selection for minimum pressure drop and maximum flow coefficient is irrelevant. The pressure loss of a full open butterfly valve is so small in comparison to the overall loss of the flow system, that it has very little effect. The larger the flow coefficient at full open, the smaller the range of openings for flow control. The concern for minimum pressure loss is often misguided because of the uncertainty of the pressure losses for rest of the flow system. The minimum pressure loss of a butterfly valve is usually less than the variation or accuracy of determining the pipe friction loss for the system.

Free Discharge

Free discharge from a butterfly valve is similar in nature to the condition of choking or flashing cavitation. Both conditions choke the flow passages and separation zones of the valve with vapor or air. The result of choking is then a limit of flow for a given upstream pressure and a reduction in the efficiency or flow coefficient of the valve. The free discharge flow and dynamic flow torques are less than those calculated for the same pressure drop of a non-cavitating or non-free discharge condition. Knapp (1950) has classified cavitation as being either a vaporous or gaseous type. Free discharge is similar in nature to gaseous cavitation in that free air is drawn into the separation zones from downstream of the valve instead of from the liquid flow itself.

Choking Cavitation

Choking or flashing cavitation in a valve is a maximum design limit that results in a level of extreme cavitation. Free discharge at lower pipe velocities and flows will not produce undesirable effects. Dynamic flow and actuator torques are lower than expected, and any cavitation will be significantly reduced. The presence of free air will cushion the cavitation collapse and in many cases will prevent cavitation vapor from even forming. The major disadvantage of free discharge is the reduction in flow and flow coefficient. However, it is important to note that free discharge conditions can produce much larger pressure differentials and velocities than from similar conditions in which downstream piping will limit flow and pressure drop. Care must be taken in designing for free discharge from a butterfly valve to avoid pipe velocities that exceed the maximum design velocities (ie 12 to 16 fps for some classes of AWWA valves).

Choking cavitation occurs when the local pressure inside a valve decreases to the vapor pressure of the liquid, and the contracted flow through the valve flashes to vapor. At choking cavitation, the maximum flow for a given upstream pressure (regardless of downstream pressure) is reached. Typically, the flows to produce large vapor pockets downstream of the valve are very large, and often exceed the maximum design flow and design structural stresses for the valve and actuator. Flashing cavitation can also produce precipitous effects to downstream piping and other flow components. It is not recommended to operate butterfly valves at or near any limit of choking or flashing cavitation

Flow Coefficients

The most common formula used to relate the pressure differential to the flow is the Darcy-Wiesbach equation (Equation 1).

$$\Delta H = K\frac{V^2}{2g} \tag{1}$$

where K is referred to as the resistance coefficient, ΔH is the net pressure differential or head loss in feet (meters), V is the flow velocity at the inlet in fps (m/s), and g is the gravitational constant 32.2 fps^2 (9.806 m/s^2). The parameter $V^2/2g$ is also referred to as the velocity head. As simple as the Darcy-Wiesbach equation is, the coefficient K has been used to denote a number of flow phenomena such as the cavitation parameter and numerous flow coefficients and constants.

A flow coefficient equation widely used by the valve industry (per ISA S75.01) is Equation 2.

$$C_V = \frac{Q_{GPM}}{\sqrt{\Delta P / Sg}} \tag{2}$$

Where ΔP is the pressure drop in units of psi, Q_{gpm} is in units of gpm, and Sg is the specific gravity of the fluid.

It is very important to note that the ΔP used in the ISA Cv_{ISA} equation is not the net pressure drop ΔP_{net}. The pressure drop used in the ISA Cv_{ISA} equation is determined from testing requirements that include an additional 8 diameters in length of pipe and friction loss in the measured pressure drop. ISA S75.01 does not allow for the subtraction of the manifold loss between pipe taps and test valve from ΔP_{ISA}. For valves with a flow coefficient Cv/d^2 less than 20 (where d is in units of inches), the difference between ΔP_{net} and ΔP_{ISA} is negligible. The difference between pressure drops can be significant for Cv/d^2 values greater than 20. Most water works valves have values of Cv/d^2 greater than 20. Equations 3 and 4 were derived (ref Rahmeyer and Driskell) for converting the ISA test pressure drop (including 8 diameters of manifold loss) to a net pressure drop.

$$\Delta P_{net} = \Delta P_{ISA} - 0.008986\, Sg\; f\, \frac{Q^2}{d^4} \tag{3}$$

Where Q is in units of gpm, d is in units of inches, and ΔP is in units of psi.

$$\left[\frac{C_{V\,net}}{d^2}\right]^2 \frac{\left[\dfrac{C_{V\,ISA}}{d^2}\right]^2}{1 - 0.008986\, Sg\; f \left[\dfrac{C_{V\,ISA}}{d^2}\right]^2} = \tag{4}$$

Another difficulty with using Cv as a loss coefficient is that Cv is not dimensionless. The units of Cv are a function of gpm and psi. However, the net resistance coefficient K is dimensionless, and can be converted directly to Cv_{net} by Equation 5.

$$\left[\frac{C_{V\,net}}{d^2}\right]^2 = \frac{890.6032}{K} \tag{5}$$

A flow coefficient that has limited use by several researchers (Tullis and Ball 1973) is the coefficient C_d (Equation 6). C_d is dimensionless and is dervied directly from K (Equations 6 and 7), and has two important advantages in its use. The range of C_d will always vary from 0 to 1, where the value 1 represents an ideal valve with no pressure loss. Most types of numerical analysis or modeling require a relationship of flow coefficient as a function of an independent variable such as valve opening. Experience has show that the generation or curve fit of such relationships produces much simpler and accurate relationships if the flow coefficient is in the form of C_d.

$$C_d = \frac{V}{\sqrt{2g\,\Delta H + V^2}} = \sqrt{\frac{1}{K+1}} \tag{6}$$

$$K = \frac{1}{C_d{}^2} - 1 \tag{7}$$

Cavitation Parameters

The cavitation parameter or index is a dimensionless ratio used to relate the conditions which inhibit cavitation (P_2-P_V) to the conditions which cause cavitation (ΔP). There are numerous forms of the dimensionless number or parameter which have been used to mathematically describe cavitation. The most fundamental form is σ_2 of Equation 8 which uses the downstream valve pressure ($P_{2)}$, the vapor pressure of the liquid ($P_{V)}$, and the pressure drop of the valve (ΔP). It is necessary to use σ_2 for determining specific cavitation effects such as size and scale effects. However, σ_2 can be difficult to use for scaling or sizing a control valve because of the lack or uncertainty of the value of the downstream valve pressure P_2. Equation 9 is an alternate form of the cavitation parameter σ that allows the use of the upstream pressure P_1 instead of the downstream pressure. This form of σ is much better suited for scaling and sizing valves. It is also important to notes that σ_2 can be converted to σ by just adding the value of 1. Equation 9 is the cavitation parameter that will be used for this paper.

$$\sigma_2 = \frac{P_2 - P_V}{\Delta P} \tag{8}$$

$$\sigma = \frac{P_1 - P_V}{\Delta P} = \sigma_2 + 1 \tag{9}$$

The cavitation parameter can be used to predict the pressure drop or discharge at which a control valve or orifice will begin to experience a given level of cavitation. If the σ calculated for the actual operating pressures of a valve or orifice is less than the value of σ for a cavitation limit, the valve or orifice will experience a level of cavitation more severe than that associated with the limit.

The choking cavitation limit represented by σ_{CHOKED} can be related to the Pressure Recovery Factor F_L by Equation 10. F_L is a form of the choking cavitation parameter that is used by most valve manufactures and valve design procedures (ISA 1985).

$$F_L = \frac{Q_{GPM}}{C_V \sqrt{P_1 - 0.96\, P_V}} = \frac{1}{\sqrt{\sigma_{CHOKED}}} \qquad (10)$$

Butterfly valves have been tested by many different researchers and valve vendors for cavitation limits. Equations 11 and 12 show the best fit equations from numerous tests of all sizes and models of valves for the cavitation limits of constant and choking cavitation.

$$\sigma_{CONSTANT} = 1.161 + \frac{9.4364}{\sqrt{K}} \qquad (11)$$

$$\sigma_{CHOKED} = 1.0851 + \frac{2.0762}{\sqrt{K}} \qquad (12)$$

Equations 11 and 12 had minimum linear regression coefficients of $R^2 = 0.97$ and a scatter or accuracy of 10 to 15% accuracy when applying the equations for different sizes and models of valves. Equations 11 and 12 have been provided as an aid for the preliminary sizing of butterfly valves for constant cavitation and for free discharge. The most accurate method for predicting cavitation limits is to still use actual test data for a specific valve type and size.

Coefficient of Dynamic Flow Torque

Dynamic flow torque is the flow torque produced by the flow forces on the valve disc about the valve shaft. Most butterfly valves produce a dynamic flow torque that has the rotational direction that will close the valve. Some types of eccentric butterfly valves that are installed with the valve seats upstream of the valve shaft will experience flow torque above 70 to 80 degrees of opening that will tend to open the valve. The dimensionless flow torque coefficient that has been used by many valve vendors and designers is C_{TDP} of Equation 13. It must be cautioned that there are other definitions and applications of C_{TDP} that are not dimensionless. For example, it is common to derive C_{TDP} from units of ft.lbs of torque, inches of disc diameter, and psi of pressure drop. This application is not dimensionless and produces coefficient values that are 1/12 th that of a dimesionless coefficient.

Another dimensionless coefficient for dynamic torque is C_{TV} of Equation 14. C_{TV} differs from C_{TDP} in that it is based on velocity instead of pressure drop. The advantage of C_{TV} is that because velocity is used to calculate dynamic torque, C_{TV} can be applied directly to applications of installing small sizes of valves in larger pipes and for use in free discharge. Equation 15 can be used to calculate C_{TV} from values of C_{TDP}. It must be cautioned that the units normally used with C_{TV} are foot pounds of torque instead of inch pounds as used with C_{TDP}.

$$C_{TDP} = \frac{Dynamic\ Flow\ Torque}{\Delta P\, d^3} = \frac{T}{\Delta P\, d^3} \tag{13}$$

$$C_{TV} = K \frac{C_{TDP}}{2} \tag{14}$$

$$C_{TV} = \frac{T}{\rho\, V^2\, d^3} \tag{15}$$

Effect of Free Discharge and Choking on Flow and Torque Coefficients

The flow coefficients and torque coefficients for a butterfly valve are determined by testing the valves in a test setup with pressurized flow both upstream and downstream of the test valve. Both types of coefficients are based on the pressure drop associated with the flow profiles and separation zones produced in pressurized flow. As discussed, choking cavitation and free discharge cause changes to the flow profiles and separation zones, and almost always reduce the efficiency or increase the pressure loss of the valve.

To predict flow and dynamic flow torque for the conditions of choking cavitation, equations 16, 17, 18, and 19 were derived from the choked cavitation parameter to correct the flow and torque coefficients for choked flow.

$$\Delta P^* = \frac{P_1 - P_V}{\sigma_{CHOKED}} \tag{16}$$

$$K^* = K\,\sigma_{CHOKED} \tag{17}$$

$$C_V^* = C_V\, F_L = \frac{C_V}{\sqrt{\sigma_{CHOKED}}} \tag{18}$$

$$C_{TDP}^* = C_{TDP}\, F_L^{\,2} = \frac{C_{TDP}}{\sigma_{CHOKED}} \tag{19}$$

Where K*, Cv* and C_{TDP}* are the corrected coefficients to be used to calculate flow and torque for choking cavitation. It can be rationalized that free discharge is a form of choking cavitation in which the flow is choked by free air instead of vapor. Tests have verified the use of the corrected coefficients (Equations 16 through 19) to calculate flow and torque for free discharge conditions. Equations 20, 21, and 22 are then used to calculate the pipe velocity, flow, and dynamic torque for a free discharge condition.

$$V_{\text{FREE DISCHARGE}} = \sqrt{\frac{2g\,\Delta H}{K^*}} \tag{20}$$

$$Q_{\text{GPM FREE DISCHARGE}} = C_V * \sqrt{P_1 - Pa} \tag{21}$$

$$T_{\text{FREE DISCHARGE}} = C_{TD} * \left(P_1 - Pa\right) d^3 \tag{22}$$

Where P_a is the atmospheric pressure. Table 1 compares calculations for free discharge with actual test data for a 12-inch symmetrical and a 36-inch eccentric butterfly valve.

Table 1 Free Discharge Test Data and Calculations

	12" SYM. BFV	36" ECC. BFV
Standard Valve Test Coefficients	....	....
Valve Opening	50 degrees	30 degrees
Cv	1,645	6,950
K	6.829	29.699
C_{TDP}	0.0783	0.0171
σ_{CHOKED}	1.985	1.1815
F_L	0.71	0.92
Free Discharge Tests	....	....
P_1-P_a	20 psig	42 psig
$Q_{MEASURED\ FREE\ DISCHARGE}$	5,137 gpm	41,405 gpm
$V_{MEASURED\ FREE\ DISCHARGE}$	14.57 fps	13.33 fps
$T_{MEASURED\ FREE\ DISCHARGE}$	1,423 in. lbs	26,813 in. lbs
Calculations	....	....
Cv*	1,167	6,394
K*	13.5554	35.0893
C_{TDP}*	0.03945	0.01446
$Q_{CALCULATED\ FREE\ DISCHARGE}$	5,220 gpm	41,437 gpm
$V_{CALCULATED\ FREE\ DISCHARGE}$	14.81 fps	13.34 fps
$T_{CALCULATED\ FREE\ DISCHARGE}$	1,363 in. lbs	27,485 in. lbs

<u>Non-line Sized Valves</u>

Control valves are not always installed in the same line size as the valve. Non-line size valve installations create additional connection losses which need to be added to the net valve loss. Therefore, it is necessary to correct the net pressure drop of the valve (ΔP) to the total loss (ΔP*) of the valve and pipe reducers. Equation 23 shows the Pressure Correction Factor used with Equation 24 to convert the net pressure loss to the total loss of a non-line sized valve. The equations to correct non-line sized valves are based on the assumption that the pipe reducers will be standard symmetrical pipe fittings (Hutchison 1976 and ISA 1985).

$$C_R = 1 + \frac{1.5\left(1 - \frac{d^2}{D^2}\right)^2}{K} \tag{23}$$

$$\Delta P^*(valve\ and\ reducer) = C_R\, \Delta P\,(net\ valve) \tag{24}$$

The total flow coefficient K* of the valve and reducers can be calculated from either Equation 25 or 26.

$$K^* = K\, C_R \tag{25}$$

$$K^* = K + 1.5\left(1 - \frac{d^2}{D^2}\right)^2 \tag{26}$$

The upstream pressure P_1* of the non-line sized valve and reducers can be calculated from Equation 2 and the Cavitation Correction Factor, C_S, of Equation 28.

$$\left(P_1^* - P_V\right) = \left(P_1 - P_V\right) + C_S\, \Delta P \tag{27}$$

$$C_S = \frac{1.5}{K}\left[1 - \left(0.67\frac{d^2}{D^2}\right) - \left(0.33\frac{d^4}{D^4}\right)\right] \tag{28}$$

The cavitation parameter for the combined valve and pipe reducers can then be calculated from the net cavitation limit with Equation 29. If the valve is line sized (d=D), the correction factor C_S will equal 0 and the correction factor C_R will equal 1.

$$\sigma^*(valve\ and\ reducers) = \frac{\sigma(net\ valve) + C_S}{C_R} \tag{29}$$

The coefficient of dynamic torque used with a valve and reducers, C_{TDP}*, can be calculated from Equation 30. It must be cautioned that the application of C_{TDP}* is based on the diameter of the valve and the pressure drop of the combined valve and pipe reducers. To avoid any confusion in calculating dynamic torque, the use of the coefficient C_{TV} is recommended where d is the diameter of the valve and V is the velocity of the valve inlet.

$$C_{TDP}^* = C_{TDP}\, C_R \tag{30}$$

Equations 23 through 30 are based on theoretical derivations and on the procedures presented by the ISA S75.01 Standards (1985) for control valves. The equations in ISA and in this paper have been verified with limited testing.

<u>Cavitation Scale Effects</u>

$$\sigma = \left(\sigma_R - 1\right) PSE\ SSE + 1 \tag{31}$$

$$(\sigma - 1) = \left(\sigma_R - 1\right) PSE \tag{32}$$

$$PSE = \left[\frac{\left(P_1 - P_V\right)}{\left(P_1 - P_V\right)_R}\right]^n \tag{33}$$

Values of n for incipient and constant limits of cavitation for typical butterfly valves are n =0.25 to 0.28. The value of n for incipient damage is n = 0.18 for a butterfly valve, and n = 0 for the limit of choking cavitation.

$$SSE + \left(\frac{d}{d_R} \right)^{Y} \tag{34}$$

$$\sigma = \sigma_R \; SSE \tag{35}$$

From numerous tests of butterfly valves, the cavitation limits of incipient and constant cavitation can be scaled with Equation 36. The value of Y for the limits of incipient damage and choking are Y = 0.

$$Y = 0.159\, K^{-1/8} \tag{36}$$

<u>Valve Size Scale Effects</u>

For different sizes of geometrically similar valves, the flow coefficients K, Cv/d^2, and Cd are constant. The flow coefficient Cv is not constant with valve size and has to be scaled by pipe area.

$$C_V = C_{V\ R} \left(\frac{d^2}{{d_R}^2} \right) \tag{37}$$

However, butterfly valves of identical design and model, typically will vary slightly in geometry for different sizes because of the structural requirements for the disc and disc shaft. The valve seats also differ in size ratio because many of the valve sizes will have the same height and thickness of seat. James Ball (Ball and Tullis 1973) presented a study of the variation of flow resistance for similar butterfly valves with differences in valve disc diameter (d_o) and thickness (t). The valves that were different models and types of butterfly valves that included high performance, with seats upstream and downstream, and symmetrical and boot seated butterfly valves. His data was presented in a plot of the flow coefficient Cd versus the ratio t/d_o. Equation 38 is the regression of the data presented in his plot. The regression analysis had a R^2 of 96.3% and a maximum scatter of data of ±8%. It is suggested to use Equation 38 as a means to extrapolate know values from one valve size to another.

$$K = -2.14 + 80.24\left(\frac{t}{d_O}\right) - 1007.3\left(\frac{t}{d_O}\right)^2 + 5{,}996\left(\frac{t}{d_O}\right)^3 - 16{,}797\left(\frac{t}{d_O}\right)^4 + 18{,}031\left(\frac{t}{d_O}\right)^5 \tag{38}$$