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Recommended Practice

Considerations for Evaluating Control Valve Cavitation

ISA-RP75.23, Considerations for Evaluating Control Valve Cavitation

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Preface

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1 Scope

This recommended practice is intended for control valves used in the control of process fluids and is not intended to apply to fluid power components. The reader and user should be familiar with fluid mechanics fundamentals and ISA standards ANSI/ISA S75.01 and ANSI/ISA S75.02 on valve sizing and testing. Definitions of terms in this document are intended for general understanding; more rigorous definitions are found in the references.

Noise measurement and prediction methods are beyond the current scope of this document. Methods of liquid flow noise measurement and prediction may be found in standards of the International Electrotechnical Commission, CEI/IEC documents 534-8-2 and 534-8-4. The relationship between cavitation parameters used in this recommended practice and those of the IEC documents is discussed in Annex B.

2 Purpose

Cavitation as an applied science has not evolved to the highly refined level of that supporting the more traditional control valve sizing calculations. However, there is a great need by users and manufacturers alike for practical information in this area. The purpose of this document is to supply that information, and to that end it is necessarily broad in scope. It embodies several objectives:

a) to provide educational material in a background section that condenses the literature and educates the reader in state-of-the-art valve cavitation knowledge and practice;

b) to establish a basis for communication by defining cavitation parameters and nomenclature;

c) to propose methods for evaluating the cavitation characteristics of individual control valves through testing procedures and application experience; and

d) to offer guidelines for selecting control valves for given applications.

ISA Subcommittee SP75.16 recognizes that the science of cavitation is in its infancy in terms of defining the behavior of cavitation in complex valve geometry. The final objective of this recommended practice is to promote additional research and testing. Subsequently, this practice can serve as a starting point for those seeking to advance the state of the art.

3 Definition of terms

Terms used are per ISA-S75.05 and additional terms as follows:

3.1 cavitation: A two-stage process associated with the flow of liquids. The first stage involves the formation of vapor cavities or bubbles in the flow stream as a result of the local static pressure in the flow stream dropping below the liquid vapor pressure. The second stage of the process is the subsequent collapse or implosion of the vapor cavities back to the liquid state when the local static pressure again becomes greater than the fluid vapor pressure. (The evaluation of "gaseous" cavitation, i.e. the sudden dissolution of dissolved gases in a liquid, is not currently within the scope of this document.)

3.2 cavitation coefficient: A characteristic number for σ (e.g., σ_i, σ_c, σ_{mv}, σ_{id}, σ_{ch}), determined for a given valve, valve opening, and pressure conditions, which corresponds to the numerical value of the cavitation index at which the levels of incipient cavitation, constant cavitation, maximum vibration cavitation, incipient damage, and choking cavitation occur.

3.3 cavitation index: The value for the operating service conditions of a valve, expressed as σ and numerically equal to $(P_1 - P_v)/(P_1 - P_2)$.

3.4 cavitation level: The degree to which cavitation is occurring, i.e., incipient, constant, incipient damage, choking, or maximum vibration. Levels can be determined by testing for vibration, pitting or metal loss, and changes in valve capacity (C_v).

3.5 choking cavitation: A limiting flow condition in which vapor formation is enough to limit the rate of flow through the valve to some maximum value. Further increases in flow rate through the valve are only possible by increasing the valve inlet pressure, because reducing downstream pressure will no longer increase flow rate.

3.6 constant cavitation: An early level of cavitation characterized by mild, steady popping or crackling sounds that may be audible or detected by vibration measurements. It is the next higher inflection point on the cavitation profile above the point of incipient cavitation. (See Figure 1.) This level is represented by the constant cavitation coefficient σ_c.

3.7 duty cycle: The ratio of the amount of time a valve spends performing one particular function to the valve's total installed time period. It may be expressed as a percentage of total time (service time vs. installed time).

3.8 flashing: A flow condition in which vapor pockets formed inside a valve persist downstream of the valve because the valve outlet pressure is at or below the fluid vapor pressure.

3.9 flow separation: A flow condition in which the fluid boundary layer flows away from the boundary wall instead of flowing along the wall. A turbulent wake exists downstream of the point of flow separation that is characterized by the presence of vortices. These vortices contain regions of high local fluid velocities and hence low, local pressures. The areas of low pressure are potential sites for vapor formation.

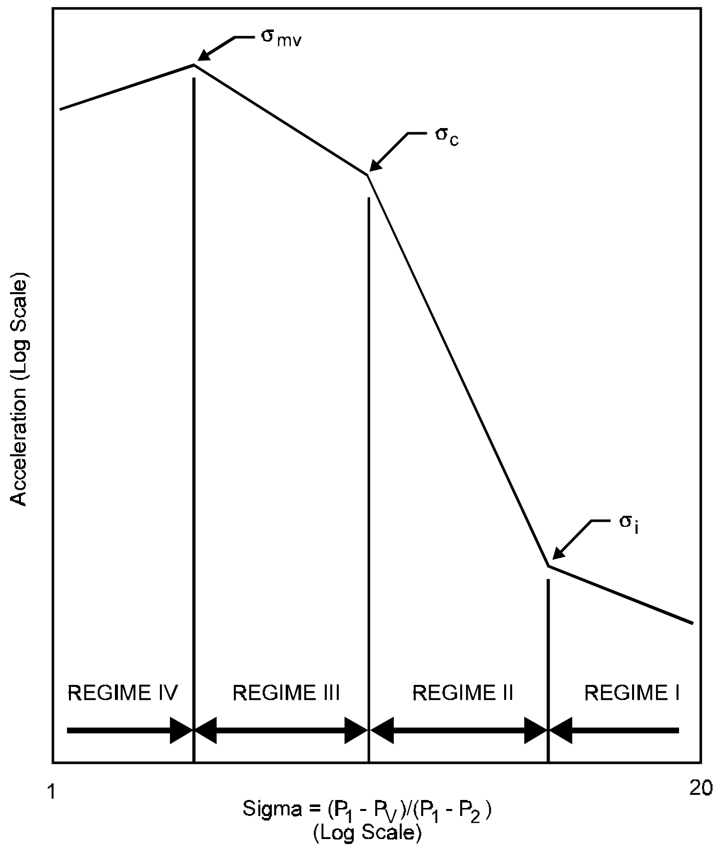

Figure 1 — Cavitation parameter plot

NOTE — This is a classical curve illustrating acceleration versus sigma. Tested valves may not result in this specific configuration or exhibit all the inflection points or coefficients shown above. Test data are subject to expert interpretation.

3.10 incipient cavitation: The onset of cavitation, where only small vapor bubbles are formed in the flow stream. (See Figure 1.) This level is represented by the incipient cavitation coefficient σ_i or $1/x_{Fz}$.

3.11 incipient damage: A cavitation level sufficient to begin minor, observable indications of pitting damage. (This is not to be confused with incipient cavitation. See 3.10.)

3.12 influences: Factors or effects that change the damage rate or extent of damage but do not change the numerical value of cavitation coefficients. See Figure B.2.

3.13 manufacturer's recommended cavitation limit: An operational limit expressed as a cavitation coefficient σ_{mr} supplied by the valve manufacturer for a given valve type, size, opening, and reference upstream pressure. Application of the limit may require scale effect and influence factors if the service conditions and valve size are different than for the reference pressure and size.

3.14 maximum vibration cavitation: The level of cavitation associated with peak vibration measurements and determined from a cavitation level plot at the peak separating Regime III and Regime IV. The test conditions at this point define the conditions for calculating the valve cavitation coefficient σ_{mv}. See Figure 1.

3.15 pressure recovery: The increase in fluid static pressure that occurs as fluid moves through a valve from the vena contracta to the valve's outlet and downstream piping. The recovery, which may be expressed as the difference P_2 - P_{vc}, is caused by the velocity-reducing, diffusing action of the downstream geometry.

3.16 scale effects: Differences in cavitation coefficients occurring between the flow test conditions and actual valve operating conditions. These scale effects result from differences in valve size and operating pressures. Scaling equations are used to modify the reference values of cavitation coefficients supplied by valve manufacturers in order to evaluate equipment at other than reference conditions. See Figure B.1.

3.17 vapor pressure: The pressure, for a specified fluid temperature, at which both the liquid and vapor phases of a fluid exist in equilibrium. The vapor pressure is more commonly thought of as the thermodynamic saturation pressure.

3.18 vena contracta: The minimum area of a flow stream. It is smaller than the area causing the flow constriction, because the streamlines continue to converge for a short distance beyond the constriction. Average flow velocity is highest and mean static pressure is lowest in the vena contracta. However, local vortex pressures in separation regions and turbulent boundary layers can be lower than the vena contracta pressure.

4 Nomenclature

Nomenclature used is per ANSI/ISA S75.01 and additionally as follows:

a Empirical characteristic exponent for calculating PSE

b A characteristic exponent for calculating SSE; determined from reference valve data for geometrically similar valves.

C_v	Valve flow coefficient*, $C_v = q(G_f/\Delta P)^{1/2}$
C_{vR}	Valve flow coefficient of a reference valve
d	Valve inlet inside diameter, inches (mm)
d_R	Valve inlet inside diameter of tested reference valve, inches (mm)
D_1	Internal diameter of upstream pipe, inches (mm)
D_2	Internal diameter of downstream pipe, inches (mm)
e	Napierian base, e = 2.71828... (for natural logarithms)
F_{DC}	Duty cycle factor for modifying the intensity index
F_F	Liquid critical pressure ratio factor*
F_L	Liquid pressure recovery factor*
F_p	Piping factor for ISA valve sizing*
F_T	Temperature factor for modifying the intensity index
F_U	Velocity factor for modifying the intensity index
G_f	Specific gravity of the liquid at inlet flowing conditions
I	Intensity index
K_{B1}	Bernoulli coefficient for upstream pipe reducer*
K_{B2}	Bernoulli coefficient for downstream pipe expansion*
K_1	Head loss coefficient for upstream pipe reducer*
K_2	Head loss coefficient for downstream pipe increaser*
$N_{1\text{-}4}$	Numerical constants for units of measure used in equations. See Table 1.
P_a	Atmospheric pressure, psia (kPa)
P_1	Valve inlet static pressure, psia (kPa)
P_2	Valve outlet static pressure, psia (kPa)
PSE	Pressure scale effect
P_v	Absolute fluid vapor pressure of liquid at inlet temperature, psia (kPa)
P_{vc}	Fluid static pressure in valve vena contracta*, psia (kPa)
q	Volumetric flow rate, gpm (m^3/h)
SSE	Size scale effect
SPL	Sound Pressure Level referenced to 20 x 10^{-6} Pascal (2.0 x 10^{-5} N/m^2)
T	Fluid Temperature, °F (°C)
T_{ave}	Average temperature between a liquid's freezing and boiling temperatures for a specified pressure, T_{ave} is equal to $(T_F + T_B)/2$, °F (°C)
T_B	Boiling temperature of a liquid for specified pressure, °F (°C)
T_F	Freezing temperature of a liquid, °F (°C)
U	Average velocity at the valve inlet, ft/s (m/s)

*More completely defined in ANSI/ISA S75.01 and ANSI/ISA S75.02

U_o Pitting threshold velocity determined at the valve inlet, ft/s (m/s)

x_{Fz} Coefficient of incipient cavitation per IEC-534-8-2; $x_{Fz} \approx 1/\sigma_i$

ΔP Measured valve differential pressure, psi (kPa)

ΔP_{ch} Pressure drop at choking, psi (kPa)

σ Cavitation index equal to $(P_1 - P_v)/(P_1 - P_2)$ at service conditions, i.e., σ(service)

σ_2 Alternate cavitation index equal to $(P_2 - P_v)/(P_1 - P_2)$ at service conditions. See B.5.6.

σ_c Coefficient for constant cavitation; σ_c is equal to $(P_1\text{-}P_v)/\Delta P$ at the conditions causing mild, steady cavitation

σ_{ch} Coefficient for choking cavitation; σ_{ch} is equal to $(P_1\text{-}P_v)/[F_L^2(P_1\text{-}F_F P_v)]$ at the point associated with choking in the valve

σ_i Coefficient of incipient cavitation; σ_i is equal to $(P_1\text{-}P_v)/\Delta P$ at the point where incipient cavitation begins to occur

σ_{id} Coefficient of incipient damage for cavitation; σ_{id} is equal to $(P_1\text{-}P_v)/\Delta P$ at the conditions causing onset of damage by cavitation

σ_{mr} Coefficient of manufacturer's recommended minimum limit of the cavitation index for a specified valve; σ_{mr} is equal to minimum recommended value of $(P_1\text{-}P_v)/(P_1\text{-}P_2)$

σ_{mv} Coefficient of cavitation causing maximum vibration as measured on a cavitation parameter plot (See Figure 1.)

σ_p Cavitation coefficient σ_v that has been adjusted for the effects of installing a smaller-than-line-size valve with reducers in the pipeline.

σ_R A reference value of a cavitation coefficient.

σ_{ss} Cavitation index scaled for service pressure and size effects for use in intensity index calculations; σ_{ss} is equal to $[(\sigma/SSE\text{-}1)/PSE] + 1$

σ_v Cavitation coefficient for a valve, scaled for a valve size and pressure other than the originally tested size and pressure, that has geometric similarity to the tested valve. It does not include the effect of reducers.

Table 1 — Numerical constants for cavitation equations

Constant		Units Used in Equation	
N		d, D	U, U_0
N_{c1}	1.00 0.00155	in mm	- -
N_{c2}	890 0.00214	in mm	- -
N_{c3}	1.00 25.4	in mm	- -
N_{c4}	0.078 0.256	- -	ft/s m/s

5 Overview

5.1 Cavitation is a phenomenon that can accompany the flow of liquids through control valves. Failure to account for cavitation can result in potentially costly performance problems. To prevent this situation, it is important that personnel responsible for control valve specifications understand the nature of cavitation and fundamental abatement technology. The purpose of this section is to provide the reader with a brief introduction to the subject. For a more comprehensive treatment of the same subject, the reader is directed to Annex B. Familiarity with this material is encouraged. Successful solutions to cavitation problems still rely heavily on engineering judgments stemming from insight into cavitation basics.

5.2 Simply viewed, cavitation consists of the formation, growth and rapid collapse of cavities in a liquid. These vapor cavities (bubbles) are formed whenever the prevailing fluid pressure falls below the vapor pressure of the liquid. They subsequently collapse if the pressure again rises above the vapor pressure.

5.3 Different specific sources of pressure changes cause cavitation, but they all arise from the flow of the liquid through the control valve. Cavitation usually begins in the low pressure regions associated with boundary layer separation. This may occur even though the mean pressure is greater than the vapor pressure. Mean pressure (the average static pressure in the plane perpendicular to the flow path) will decrease as the liquid passes through the various restrictions in the valve trim. The degree and extent of cavitation escalates when the mean pressure falls below the vapor pressure in these regions.

5.4 Unacceptable noise levels, excessive vibrations, and physical damage to the valve and adjacent hardware are the foremost problems associated with cavitation. These problems all arise from the collapse of the vapor cavities. Material damage results from shock waves and micro-jets, established during cavity collapse, impinging on the boundary surfaces. Corrosion further aggravates these mechanical attack mechanisms. The physical appearance of cavitation damage varies from a "frosted glass" appearance to a rough, cinder-like surface texture.

5.5 Another "side effect" of cavitation is an apparent decrease in the efficiency of the valve. The compressibility introduced to the fluid when a portion of the liquid vaporizes can ultimately lead to a choked flow condition similar to a flashing fluid.

5.6 While treated simply in this section, cavitation is a very complex sequence of events. Not all cavitation necessarily results in the problems mentioned above. However, attempts to model the behavior of the cavitating liquid have not met with universal success. Distinguishing "problem causing" cavitation from acceptable behavior presents some very real challenges.

5.7 Historically, the control valve industry has adopted the practice of describing cavitation applications in terms of a single, unadjusted parameter. In this approach, the suitability of a given control valve is determined by comparing the value of this parameter evaluated at operating conditions to an "operating limit" for that control valve.

5.8 While appealing from a user standpoint, the approach described above suffers from some major drawbacks. First, the definition of the parameter and the manner in which it is used have varied significantly from manufacturer to manufacturer. While the principle underlying the method is basically the same, the differences in appearance lead to much confusion. Furthermore, the complexity of cavitation renders it difficult to predict the exact behavior in any given service on the basis of a single, unadjusted parameter. Many service factors can affect the apparent level of cavitation. Unfortunately, no currently known model fully describes the intensity or extent of cavitation under universally varying conditions regardless of the number of parameters employed.

5.9 The operating limit used as the basis for comparison has, in many instances, been equal to the value of the pressure recovery factor, F_L. If a valve is operated at the limit defined by the pressure recovery factor, the valve is at or near choked flow conditions. Substantial vapor has been formed in the flow stream, and significant levels of cavitation can exist. As discussed elsewhere, using F_L in this manner is not a universally correct solution and is, in general, valid only for specially designed valve trims. The vast majority of valves cannot operate problem free under this condition.

5.10 A modified single parameter will be adopted for use in this document. A specific parameter, as defined in Equation 1, is recommended. Adjustments to this parameter are supplied wherever they are known to account for the variations associated with different application conditions. While it is recognized that even this technique may have limitations, it is believed that it will provide a justifiable blend of ease of use and meaningful predictions.

5.11 The parameter chosen for use in this document is the cavitation index

$$\sigma = (P_1 - P_v)/(P_1 - P_2) \tag{1}$$

where P_1 is the absolute pressure upstream of the valve, P_2 is the absolute pressure downstream of the valve, and P_v is the absolute vapor pressure of the fluid at the inlet temperature.

5.12 As noted above, other parameters have been used and probably will continue to be used in the future for essentially the same purpose. In many instances, well defined mathematical relationships exist between these parameters and the index defined in Equation 1. Several other parameters, relationships to σ, and the associated advantages or disadvantages are discussed in Annex B.

5.13 The σ index, in effect, quantifies only the service conditions. By itself it does not convey any information about the performance of a particular valve in that particular application. Different valves can tolerate different levels of cavitation, and different applications are concerned about different aspects of cavitation (for instance, noise versus damage). Therefore, σ must be evaluated at the service conditions and then compared to some benchmark.

5.14 The benchmark σ value for any specific application obviously will depend on both the problem of concern (e.g., noise) as well as the valve style selected. Various limits have been suggested and used in the control valve industry in the past. The following benchmarks, hereafter referred to as levels, are used throughout this document:

a) incipient cavitation;

b) constant cavitation;

c) incipient damage;

d) choking cavitation; and

e) maximum vibration cavitation.

Definitions of these various levels are given in Section 3. More complete descriptions of their significance are provided in Annex B. A discussion of the methods of determination is presented in Section 8.

5.15 The different levels of cavitation listed in 5.14 merely define different significant cavitation conditions that exist. No specific levels can be universally recommended. The appropriate level to use for a given application is not always self-evident and will usually embrace a degree of subjectivity. In addition to the service conditions, factors such as valve style and opening, duty cycle, location, desired life, and past experience should be considered. The valve manufacturer always should be consulted. A manufacturer may recommend an application dependent valve operating limit called "manufacturer's recommended limit" or σ_{mr}. Additional discussion may be found in Section 7 and Annex B of this document.

5.16 These various levels are a strong function of the internal geometry of the control valve. It can be expected that different values of any given cavitation coefficient will be associated with different valve styles or even different openings of the same valve.

5.17 Furthermore, the numerical values of these cavitation coefficients must be adjusted if a reduced scale model was used to determine them. Any factor that changes the numerical value of a cavitation coefficient as that factor is varied is known as a "scale effect." The numerical values of σ coefficients can be corrected for "size scale effect" (SSE) through the use of scaling equations presented in Section 6.

5.18 Research has shown that many factors contribute to the total nature of cavitation and to the resulting problems. Some of these factors are associated with the valve geometry as noted in 5.16. Others are associated with the service environment. The foremost service condition scale effect is the "pressure scale effect" (PSE). The numerical values of the various cavitation coefficients for a particular valve will change as a function of the pressures at which they are evaluated. Consequently, an adjustment in the values is necessary if the service pressures are different from the test pressures. Equations to calculate these adjustments are presented in Section 6.

5.19 At this point it is helpful to introduce another category of effects other than "scale effects" as defined in 5.17. Application "influences" include factors that do not change the numerical value of cavitation coefficients as in scale effects, but that do affect the intensity of the cavitation. The list of influences is long, but for engineering purposes can be pared down to the following primary effects (Knapp, ref. 1; Barnes and Cain, ref. 13):

a) viscosity;

b) velocity;

c) dissolved and undissolved gases in the liquid;

d) thermal properties of the liquid; and

e) duty cycle.

More detailed discussion regarding the nature of these effects and methods of accounting for them are presented in Annexes B and C.

5.20 The pressure drop, ΔP, measured in testing valves for cavitation is the pressure difference between upstream and downstream pressure taps of the test manifold. For higher recovery valves ($C_v/N_1d^2>20$) and critical applications, it may be necessary to adjust this pressure drop to account for actual piping configuration in evaluating C_v, ΔP, valve opening, and σ. Annex D discusses this in more detail.

5.21 Cavitation will continue to be a major problem in industrial process control. An understanding of the nature of the subject and utilization of current quantitative information will aid in formulating effective problem abatement. However, the ultimate benefit of analyzing valves and service conditions for cavitation depends upon the quality and completeness of service and valve information available. Valve users and manufacturers should make every reasonable effort to share clear and accurate data (see ISA-RP75.21, Process Data Presentation). The data will make possible the comparisons between the service conditions and valve capabilities.

5.22 The procedures contained in this document are intended to provide the best practical knowledge currently available on the subject. However, practitioners always should avail themselves of proven new technology as it becomes available.

6 Cavitation index and valve scale effects

6.1 Application dependencies

6.1.1 The cavitation index σ (Equation 1) is based on the assumption that the size of the valve and the process fluid properties (other than vapor pressure) would have little effect on the value of the index. (Knapp et. al., ref. 1). In actuality, the cavitation behavior and cavitation coefficients are not independent or constant with either different upstream pressures or valve size. The change in the value of a cavitation coefficient associated with change in pressure is known as the "pressure scale effect" (PSE) (Tullis, refs. 2, 4; Rahmeyer, ref. 3; Stripling, ref. 6). Likewise, the change in the value of a cavitation coefficient associated with valve size is known as the "size scale effect" (SSE) (Tullis, refs. 2, 4; Rahmeyer, ref. 3; Stripling, ref. 6). In addition to direct valve size changes, the presence of pipe reducers or increasers can also affect the value of a coefficient.

6.1.2 Methods are presented in this section to account for these effects. Equations are available that adjust the value of the coefficients for the difference between the actual size or pressure (operating conditions) and the test size or pressure (reference conditions) (Tullis, ref. 2). Modifications to the coefficients for pipe fittings are also provided. It is important to note that the equations for scale effect adjustments were developed using water as the test fluid. They are assumed valid for fluids other than water.

6.1.3 Other factors besides pressure and valve size also can influence the nature of the cavitation. Fluid properties such as viscosity, density, and surface tension are recognized effects (Knapp, et al., ref. 1; Barnes and Cain, ref. 13). Effects such as these do not tend to change the value of the coefficients, but can change the degree of cavitation associated with any particular cavitation level. These are referred to as "influences", which are further discussed in Annex B.

6.2 Equations for scaling the cavitation coefficients

6.2.1 Scaling equations and exponents have been derived to adjust or extrapolate cavitation coefficients from one system pressure and size to another. Equation 2 gives the relationship by which a coefficient for the valve, σ_v, can be calculated from a reference coefficient, σ_R. The value of σ_R may be chosen as the value of a cavitation coefficient or a manufacturer's limit σ_{mr}. Equation 2 reflects the correlation of published data (Rahmeyer, ref. 3 and Tullis, ref. 4).

$$\sigma_v = (\sigma_R SSE - 1)PSE + 1 \quad (2)$$

After σ_v has been calculated, it can be compared to σ(service) calculated by Equation 1. If σ(service) is greater than σ_v, the valve will operate at a level of cavitation less severe than that for which the valve's σ_R was determined by the manufacturer. If the valve will be installed in a larger diameter pipe, σ_p must be calculated from σ_v by Equation 7. The piping adjusted σ_p is then compared to σ(service).

6.2.2 Intensity and level of cavitation increase with increasing (P_1-P_v). The pressure scale effect or scaling correction PSE can be calculated from the power relationship of Equation 3.

$$PSE = [(P_1-P_v)/(P_1-P_v)_R]^{a} \qquad (3)$$

The subscript R refers to reference pressures. In most cases, the actual test pressure difference (P_1-P_v) will be less than 100 psi (690 kPa). For high recovery valves with $C_v/N_1d^2 > 20$, refer to Annex D for suggested piping loss corrections. For the purpose of uniformity in data presentation, it is recommended that the test data be scaled up to and presented at a reference value of $(P_1-P_v)_R$ is equal to or less than 100 psi (690 kPa). For example, for reasons of system capabilities and concerns for safety, a rotary valve might be tested at (P_1-P_v) equal to 40 psi (276 kPa).

The exponent *a* of Equation 3 is found by measuring the slope of a log-log plot of σ_n (n=i, c, mv, id...) versus P_1-P_v. Table 2 shows a list of typical values of the coefficient for different valve types and the different levels of cavitation. The value of zero for the coefficient *a* indicates there is no pressure scale effect for choking (Tullis, ref. 4; Rahmeyer & Odeh, ref. 5).

Table 2 — Pressure scale effect exponent

Valve type	Cavitation level	Exponent a
Quarter-turn valves (e.g., ball, butterfly)	Incipient	0.22 - 0.30
	Constant	0.22 - 0.30
	Incipient Damage	0.10 - 0.18
	Choking	0
Segmented ball and eccentric plug	Incipient	0.30 - 0.40
	Constant	0.30 - 0.40
	Incipient Damage	N/A
	Choking	0
Single-stage globe	Incipient	0.10 - 0.14
	Constant	0.10 - 0.14
	Incipient Damage	0.08 - 0.11
	Choking	0
Multi-stage globe	Incipient	0.00 - 0.10
	Constant	0.00 - 0.10
	Incipient Damage	N/A
	Choking	0
Orifice	Incipient	0
	Constant	0
	Incipient Damage	0.20
	Choking	0

N/A = Not Available

6.2.3 Intensity and level of cavitation also increase with valve size. The size scale effect correction SSE can be calculated from the power relationship of Equation 4. The exponent *b* of Equation 5 was derived from limited testing for the size scale effect for the cavitation levels of incipient, constant, and incipient damage. Note that there is no size scale effect for the cavitation level of choking. Therefore, the coefficient *b* has a value of zero for the level of choking cavitation (Rahmeyer & Odeh, ref. 5; Stripling, ref. 6).

$$SSE = (d/d_R)^b \tag{4}$$

$$b = 0.068\left(\frac{C_v}{N_1 d^2}\right)^{1/4} \tag{5}$$

where C_{vR} and d_R refer to the reference valve.

It also must be noted that this equation for scaling cavitation coefficients is rigorously valid only when the valves are geometrically similar. Most valves usually are not geometrically similar for different sizes. However, limited testing has suggested that these equations may be used whenever two valves are of the same style (e.g., globe, butterfly, or ball), flow is in the same direction, <u>and</u>

$$C_v/d^2 = C_{vR}/d_R^2 \tag{6}$$

> **NOTE —** It is difficult to apply size scale effects to specialty, multi-stage, anti-cavitation valve designs. For this reason, the valve manufacturer should be consulted for applications of this type of product.

6.2.4 Another important effect on valve cavitation is the installation of a valve in a larger sized pipeline — for example, a 6-inch (150 mm) valve in an 8-inch (200 mm) pipeline. The upstream pipe reducer and downstream expansion cause a variation in cavitation levels and coefficients. Equation 7 can be used to calculate the corrected cavitation coefficient σ_P for the installation of a valve in a larger sized pipeline. The coefficient σ_P (for the proposed valve and reducers) is then compared to the σ (Equation 1) of the service system. The coefficient σ_v (for the valve without reducers) first should be calculated from Equation 2, before it is used in Equation 7. Equation 7 can be used with all of the cavitation levels (Rahmeyer, ref. 3, (&Odeh), 5; Cain, ref. 19).

$$\sigma_P = F_P^2\,[\sigma_v + (K_1+K_{B1})C_v^2/(N_2 d^4)] \tag{7}$$

$$F_P = [1+(\Sigma K)\,Cv^2/(N_2 d^4)]^{-1/2} \tag{8}$$

$$K_{B1} = 1 - d^4/D_1^4 \tag{9}$$

$$K_{B2} = 1 - d^4/D_2^4 \tag{10}$$

$$K_1 = 0.5\,(1 - d^2/D_1^2)^2 \tag{11}$$

$$K_2 = 1.0\,(1 - d^2/D_2^2)^2 \tag{12}$$

$$\Sigma K = K_{B1} - K_{B2} + K_1 + K_2 \tag{13}$$

Equations 7 through 13 are theoretical expressions that account for the combined head losses associated with the upstream and downstream reducers. Other potential effects of close coupled reducers are not included. Although these equations have been supported by testing, the accuracy of the method becomes less certain as the relative capacity (C_v/d^2) increases. Actual σ_P should be used for maximum accuracy.

7 Applications

7.1 Method

The method of using σ for determining the level of cavitation in a valve is a straightforward procedure. A value of σ is calculated for the service conditions; σ is then compared to a selected cavitation coefficient for the valve. If the σ(service) is greater than the σ of the selected limit, the valve will experience an intensity of cavitation less than that associated with the selected limit. This assumes that the service pressures and valve size are the same as those used in testing for the different levels or regimes of cavitation. (See Annex B.7 and B.8) Section 6 explains how data from cavitation tests may be "scaled" for comparison with other service pressures and valve sizes.

7.2 Valve information

Evaluation or selection of a valve for cavitation applications can be reasonably objective after certain decisions have been made and the necessary information is available. This section will describe how to use cavitation coefficients discussed in Sections 5 and 6 to evaluate a valve in specific service conditions. The following steps are prerequisites to the evaluation:

a) Gather system and service data for the valve. Pressures P_1 and P_2 should be determined at the valve inlet and outlet, respectively. For valves with $C_v/N_1d^2 > 20$ (e.g., ball, butterfly, and plug valves), see Annex D.

b) Make a selection of the general type valve (i.e., select globe, ball, butterfly, plug, etc.) unless the valve type is already known.

c) Gather data from cavitation tests for the type and size of valve being considered. See Section 8 on cavitation testing. Consult the manufacturer for available coefficients (i.e., σ_c, σ_{id}, etc.) or the recommended operating σ limit (i.e., σ_{mr}).

7.3 Operating levels

The various levels of cavitation are introduced in 5.14 and described in Annex B. The recommended cavitation coefficient for a particular valve depends on the application requirements, valve performance capabilities, and cost considerations. A valve's reference cavitation index σ_R usually is set at a value of σ_c, σ_{id}, or σ_{mr} depending on the tolerable level of cavitation. Certain designs may not exhibit cavitation parameter plots with clear, meaningful σ_i, σ_c, or other characteristics. Some valves may not have been tested due to test facility limitations. In these circumstances, a manufacturer may provide only a σ_{mr} based on field experience, damage testing, or other criteria agreeable to the user.

7.4 Considerations for selecting cavitation limits

7.4.1 Manufacturer's recommended limit, σ_{mr}

The manufacturer's recommended limit for cavitation, σ_{mr}, is the limit suggested by the manufacturer for a given valve. It may or may not coincide with other cavitation coefficients such as incipient damage or constant cavitation. Published values of this limit are based on experience and on the normal type of application for the valve. Published values may not be suitable for all applications. The manufacturer also should publish the criteria for the selection of σ_{mr}. The manufacturer always should be contacted to verify the recommended limit for each type of valve application.

7.4.2 Incipient cavitation, σ_i

Selecting incipient cavitation, σ_i, as a limit restricts all operation to a cavitation-free regime. This regime exists only with relatively low pressure drops. For instance, typical values of σ_i for a high recovery valve and a low recovery valve are approximately 15 and 8, respectively. Moderate to high pressure drop applications require larger, more costly, multi-stage trim to maintain cavitation-free operation. This regime of operation may be necessary in sensitive laboratory test applications or in certain biochemical processes where all vaporization must be prevented.

7.4.3 Constant cavitation, σ_c

Constant cavitation, σ_c, is a mild level of cavitation that produces low levels of vibration and cavity formation and poses no danger to valve equipment. This is considered a conservative application limit; generally, no objectionable noise, vibration, or damage occurs at this condition.

7.4.4 Incipient damage cavitation, σ_{id}

Operating at the incipient damage level of cavitation, σ_{id}, may produce objectionable noise and vibration in a control valve; minor indications of pitting begin on softer materials. Testing for σ_{id} is a complex process beyond the scope of this document. Additional guidance for damage testing is available in the references (refs. 1,4,6,7,10,15). Many control valves can operate at this level in moderate pressure drops if minor trim erosion is tolerable. For smaller valve sizes, stainless steels or hardened materials often provide enough resistance for economical operation without additional design enhancements. Special materials, multi-stage, or tortuous-path designs often are required to operate at this limit in very high pressure drop applications. When operation beyond this level cannot be avoided, evaluation of the intensity index per Annex C is suggested.

7.4.5 Choking cavitation, σ_{ch}

The choking cavitation coefficient, σ_{ch}, is not determined from a cavitation parameter plot (see Figure 1), but it is calculated (Equation B.4) to correspond to the liquid pressure recovery factor, F_L. It is important to note that scale effects for pressure and size differences do not apply to F_L or σ_{ch}. Most control valve applications should not use σ_{ch} as a reference limit for operation. When liquid flow becomes choked, the cavitation has become severe enough to develop very large volumes of vapor in the valve and piping until pressure recovery causes the vapor cavities to collapse. Flashing, on the other hand, does not produce cavity collapse because the downstream pressure remains below the vapor pressure. The increase in volume develops extremely high velocities. Most materials are subject to severe damage when exposed to choked conditions for any significant length of time. Valves designed for this service are intended for infrequent operation (e.g., pressure relief valves); some use sacrificial components (e.g., chokes and liners); and they usually exhaust into tanks, headers, or vents to protect downstream piping and equipment from erosion. Infrequent upset or start-up conditions may justify operating briefly in the choked condition, but this should be done with great caution. Annex C on intensity and service life presents an evaluation procedure for quantifying these situations.

7.4.6 Maximum intensity of vibration, σ_{mv}

As with choking, the conditions at the point of maximum intensity of vibration, σ_{mv}, can subject valve and piping to severe damage when exposed for a significant duration. Valves in this service are designed for infrequent operation, sacrificial components, and for exhausting into vessels without impingement on vessel walls. Annex C presents an evaluation procedure for estimating damage intensity.

7.5 Cavitation-resistant valve designs

Frequently, it is necessary to use special designs to reduce the effects of cavitation or to eliminate cavitation altogether. Manufacturers produce valves or special trim for cavitation applications. Designs exist for rotary valves and for linear motion valves. Cavitation resistant control valves vary widely in cost and performance capabilities. Understanding the application requirements and accurate process data from valve users is essential in applying cavitation-resistant valve designs.

7.6 Examples

The following examples illustrate the application of this cavitation evaluation method.

7.6.1 Rotary valve application (US Customary units)

Service data	Fluid: Water T = 74 °F Line Size: 10-inch Sch. 40 P_v = 0.41 psia P_1 = 82 psia G_f = 0.998 P_2 = 70 psia q = 3500 gpm
Results of valve sizing calculations	C_v = 1009
Preliminary valve selection	8-inch, ANSI Class 150 throttling rotary disk valve Approximate opening = 75%
Calculate σ (Equation 1)	σ = (82-0.41)/12 = 6.80
Compare with manufacturer's cavitation data. (σ_{mr} and *a* are based on incipient damage for this application.)	σ_i = 12.5 σ_c = 7.0 σ_{id} = 4.0 σ_{mr} = 4.1 $(P_1\text{-}P_v)_R$ = 100 psi; a = 0.12; d_R = 6-inch C_{vR}/d_R^2 = 16 at 75% opening
Calculate PSE, *b*, SSE (Equations 3, 4, 5)	PSE = $[(82 -.41)/100]^{0.12}$ = 0.976 $b = 0.068[1009/(1)(8^2)]^{1/4}$ = 0.14 SSE = $(8/6)^{0.14}$ = 1.04
Calculate σ_v (Equation 2)	Let $\sigma_R = \sigma_{mr}$ = 4.1 σ_v = [(4.1)(1.04)-1](0.976)+1 = 4.186

Calculate effects of pipe reducers:	Valve inlet d = 8.00 in.
	Pipe inside diameter D = 10.0 in.
(Equation 9)	$K_{B1} = 1 - (8.0)^4/(10.0)^4 = 0.59$
(Equation 10)	$K_{B2} = 1 - (8.0)^4/(10.0)^4 = 0.59$
(Equation 11)	$K_1 = 0.5[1-(8.0)^2/(10.0)^2]^2 = 0.065$
(Equation 12)	$K_2 = [1 - (8.0)^2/(10.0)^2]^2 = 0.13$
(Equation 13)	$\Sigma K = 0.59 - 0.59 + 0.065 + 0.13 = 0.195$
(Equation 8)	$F_p = \{1+(0.195)(1009)^2/[890(8)^4]\}^{-1/2} = 0.974$
(Equation 7)	$\sigma_P = (0.974)^2\{4.186+(0.065+0.59)(1009)^2/[890(8)^4]\}$
	$\sigma_P = 4.14$
Evaluation	Valve has an acceptable σ_P (i.e., $\sigma \geq \sigma_P$). Minor cavitation may be present, because σ is slightly less than σ_c.

7.6.2 Globe valve in ammonia service (US Customary units)

Service data	Fluid: Ammonia	T = 20 °F
	Line Size: 3 inch Sch 40	P_v = 48.2 psia
	P_1 = 149.7 psia	G_f = 0.65
	P_2 = 64.7 psia	q = 850 gpm
	ΔP = 85 psi	

Results of valve sizing calculations	C_v = 74.3
Preliminary valve selection	3-inch, ANSI Class 300 globe valve Full Area trim, Linear Characteristic
Calculate σ (Equation 1)	$\sigma = (149.7-48.2)/85 = 1.19$
Compare with manufacturer's recommended σ based on incipient damage for this application.	<u>Trim Style σ_{mr}</u> Standard 2.0 Trim A 1.15 Trim B 1.002
Trim A is checked for size and pressure scale effects	Manufacturer's data for Trim A $(P_1 - P_v)_R$ = 90 psi; a = .20; d_R=3.0 inch $PSE = (101.5/90)^{0.20} = 1.02$ $SSE = (d/d_R)^b = (3/3)^b = 1.00$
Calculate σ_v for Trim A (Equation 2)	Let $\sigma_R = \sigma_{mr} = 1.15$ $\sigma_v = [(1.15)(1.00)-1](1.02)+1 = 1.153$
Evaluation	Trim A in a 3-inch globe has an acceptable σ_v (i.e., $\sigma \geq \sigma_v$).

7.6.3 Boiler feedwater application for globe valves (US Customary Units)

Service data	Fluid: Water	T = 350°F
	Line Size: 8-inch Sch. 80	
	P_v = 135 psia	
	P_1 = 1600 psia	G_f = 0.89
	P_2 = 1500 psia	q = 1800 gpm

Results of valve sizing calculations	$C_v = 170$
Preliminary valve selection	6-inch, ANSI Class 900, globe valve, d = 5.75 in. Reduced Trim, Equal Percentage Characteristic
Calculate σ (Equation 1)	$\sigma = (1600 - 135)/100 = 14.6$
Manufacturer's recommended operating σ and scaling data for incipient damage	$\sigma_{mr} = 2.5$ $(P_1 - P_v)_R = 100$ psi; $a = 0.11$; $d_R = 3.0$ inch; $C_{vR}/d_R^2 = 5.5$
Calculate PSE (Equation 3)	$PSE = [(1600-135)/100]^{0.11} = 1.34$
Calculate b and SSE (Equation 4, 5)	$b = 0.068\ (5.5)^{1/4} = 0.102$ $SSE = (5.75/3.0)^{0.102} = 1.07$
Calculate σ_v (Equation 2)	Let $\sigma_R = \sigma_{mr} = 2.5$ $\sigma_v = [(2.5)(1.07)-1](1.34)+1 = 3.24$
Calculate effects of pipe reducers.	Valve inlet d = 5.75 in. Pipe inside diameter D = 7.62 in.
(Equation 9)	$K_{B1} = 1 - (5.75)^4/(7.62)^4 = 0.68$
(Equation 10)	$K_{B2} = 1 - (5.75)^4/(7.62)^4 = 0.68$
(Equation 11)	$K_1 = 0.5[1 - (5.75)^2/(7.62)^2]^2 = 0.093$
(Equation 12)	$K_2 = [1 - (5.75)^2/(7.62)^2]^2 = 0.185$
(Equation 13)	$\Sigma K = 0.68 - 0.68 + 0.093 + 0.185 = 0.278$
(Equation 8)	$F_p = \{1+(0.278)(170)^2/[890(5.74)^4]\}^{-1/2}$ $F_p = 0.996$
(Equation 7)	$\sigma_p = (0.996)^2\{3.24+(0.093+0.68)(170)^2/[890(5.75)^4]\}$ $\sigma_p = 3.24$
Evaluation	Preliminary selection is acceptable, because σ_p (3.24) is less than the operating σ (14.6). Also note that piping effects are negligible for small values of C_v/N_1d^2.

7.6.4 Rotary Valve Application (SI units)

Service data	Fluid: Water T = 23.3 °C Line Size: NPS 10, Sch. 40, D = 254 mm $P_v = 2.83$ kPa $P_1 = 565.39$ kPa $P_2 = 482.65$ kPa $G_f = 0.998$ $q = 795.6\ m^3/h$
Results of valve sizing calculations	$C_v = 1009$
Preliminary valve selection	NPS 8, ANSI Class 150 throttling rotary disk valve Approximate opening = 75%, d = 203 mm
Calculate σ (Equation 1)	$\sigma = (565.39-2.83)/82.74 = 6.80$

Compare with manufacturer's cavitation data. (σ_{mr} and *a* are based on incipient damage for this application.)	$\sigma_i = 12.5$ $\sigma_c = 7.0$ $\sigma_{id} = 4.0$ $\sigma_{mr} = 4.1$ $(P_1\text{-}P_v)_R = 690$ kPa; $a = 0.12$; $d_R = 152$ mm $C_{vR}/N_1 d_R^2 = 16$ at 75% opening
Calculate PSE, *b*, SSE (Equation 3, 4, 5)	$PSE = [(565.39\text{-}2.83)/690]^{0.12} = 0.976$ $b = 0.068[1009/(0.00155)(203^2)]^{1/4} = 0.14$ $SSE = (203/152)^{0.14} = 1.04$
Calculate σ_v (Equation 2)	Let $\sigma_R = \sigma_{mr} = 4.1$ $\sigma_v = [(4.1)(1.04)\text{-}1](0.976)+1 = 4.186$
Calculate effects of pipe reducers:	Valve inlet d = 203 mm Pipe inside diameter D = 254 mm
(Equation 9)	$K_{B1} = 1 - (203)^4/(254)^4 = 0.59$
(Equation 10)	$K_{B2} = 1 - (203)^4/(254)^4 = 0.59$
(Equation 11)	$K_1 = 0.5[1\text{-}(203)^2/(254)^2]^2 = 0.065$
(Equation 12)	$K_2 = [1 - (203)^2/(254)^2]^2 = 0.13$
(Equation 13)	$\Sigma K = 0.59 - 0.59 + 0.065 + 0.13 = 0.195$
(Equation 8)	$F_p = \{1+(0.195)(1009)^2/[0.00214(203)^4]\}^{-1/2} = 0.974$
(Equation 7)	$\sigma_P =$ $(0.974)^2\{4.186+(0.065+0.59)(1009)^2/[0.00214(203)^4]\}$ $\sigma_P = 4.14$
Evaluation	Valve has an acceptable σ_P (i.e., $\sigma \geq \sigma_P$). Minor cavitation may be present, because σ is slightly less than σ_c.

7.6.5 Globe valve in ammonia service (SI units)

Service data	Fluid: Ammonia T = -6.67 °C Line Size: NPS 3, Sch 40, D = 76 mm $P_v = 332.4$ kPa $P_1 = 1032.4$ kPa $P_2 = 446.2$ kPa $\Delta P = 586.2$ kPa $G_f = 0.65$ $q = 193.2\ m^3/h$
Results of valve sizing calculations	$C_v = 74.3$
Preliminary valve selection	NPS 3, ANSI Class 300 globe valve, d = 76 mm Full Area trim, Linear Characteristic
Calculate σ (Equation 1)	$\sigma = (1032.4\text{-}332.4)/586.2 = 1.19$
Compare with manufacturer's recommended σ based on incipient damage for this application.	<u>Trim Style σ_{mr}</u> Standard 2.0 Trim A 1.15 Trim B 1.002

Trim A is checked for size and pressure scale effects	Manufacturer's data for Trim A $(P_1 - P_v)_R$ = 620 psi; a = .20; d_R=76 mm PSE = $(700.3/620)^{0.20}$ = 1.02 SSE = $(d/d_R)^b = (76/76)^b$ = 1.00
Calculate σ_v for Trim A (Equation 2)	Let $\sigma_R = \sigma_{mr}$ = 1.15 σ_v = [(1.15)(1.00)-1](1.02)+1 = 1.153
Evaluation	Trim A in a 3-inch globe has an acceptable σ_v (i.e., $\sigma \geq \sigma_v$).

7.6.6 Boiler feedwater application for globe valves (SI units)

Service data	Fluid: Water	T = 176.7 °C
	Line Size: NPS 8, Sch. 80,	D = 193.5 mm
	P_v = 931.0 kPa	G_f = 0.89
	P_1 = 11 034 kPa	q = 409.1 m^3/h
	P_2 = 10 344 kPa	

Results of valve sizing calculations	C_v = 170
Preliminary valve selection	NPS 6, ANSI Class 900, globe valve, d = 146 mm Reduced Trim, Equal Percentage Characteristic
Calculate σ (Equation 1)	σ = (11034 - 931)/690 = 14.6
Manufacturer's recommended operating σ and scaling data for incipient damage	σ_{mr} = 2.5 $(P_1 - P_v)_R$ = 690 kPa; a = 0.11; d_R = 76 mm; $C_{vR}/N_1 d_R^2$ = 5.1
Calculate PSE (Equation 3)	PSE = $[(11034-931)/690]^{0.11}$ = 1.34
Calculate b and SSE (Equation 4, 5)	$b = 0.068[170/(0.00155)(146)^2]^{1/4}$ = 0.102 SSE = $(146/76)^{0.102}$ = 1.07
Calculate σ_v (Equation 2)	Let $\sigma_R = \sigma_{mr}$ = 2.5 σ_v = [(2.5)(1.07)-1](1.34)+1 = 3.24
Calculate effects of pipe reducers:	Valve inlet d = 146 mm Pipe inside diameter D = 194 mm
(Equation 9)	$K_{B1} = 1 - (146)^4/(193.5)^4 = 0.68$
(Equation 10)	$K_{B2} = 1 - (146)^4/(193.5)^4 = 0.68$
(Equation 11)	$K_1 = 0.5[1 - (146)^2/(193.5)^2]^2 = 0.093$
(Equation 12)	$K_2 = [1 - (146)^2/(193.5)^2]^2 = 0.185$
(Equation 13)	$\Sigma K = 0.68 - 0.68 + 0.093 + 0.185 = 0.278$
(Equation 8)	$F_p = \{1+(0.278)(170)^2/[0.00214(146)^4]\}^{-1/2}$ $F_p = 0.996$
(Equation 7)	$\sigma_p =$ $(0.996)^2\{3.24+(0.093+0.68)(170)^2/[0.00214(146)^4]\}$ $\sigma_p = 3.24$

Evaluation	Preliminary selection is acceptable, because σ_p (3.24) is less than the operating σ (14.6). Also note that piping effects are negligible for small values of C_v/N_1d^2.

8 Testing

8.1 Scope

This section provides a method of testing to determine the following control valve performance characteristics in a cavitating fluid service. Section 6.2 describes how the test results of one valve may be scaled to larger or smaller valves and other pressure conditions.

a) σ_i (end of Regime I and beginning of Regime II), incipient cavitation coefficient;

b) σ_c (end of Regime II and beginning of Regime III), constant cavitation coefficient; and

c) σ_{mv} (end of Regime III and beginning of Regime IV), point of maximum vibration cavitation coefficient .

NOTE — The above cavitation coefficients are not intended to identify a point of unacceptable or damaging cavitation. Sections 5, 6, 7, and Annex B of this recommended practice explain in detail their use and description.

8.2 Test system

8.2.1 General description

The flow test system shall be as shown in Figure 2. (Except for the cavitation detection equipment, the flow test system is the same as that required for testing to ISA-S75.02, "Control Valve Capacity Test Procedures.") It includes:

a) a test specimen;

b) a test section;

c) upstream and downstream throttling valves;

d) a flow measurement device;

e) pressure taps and measuring devices (upstream and downstream);

f) a temperature sensor; and

g) cavitation detection instrumentation shown in Figure 3.

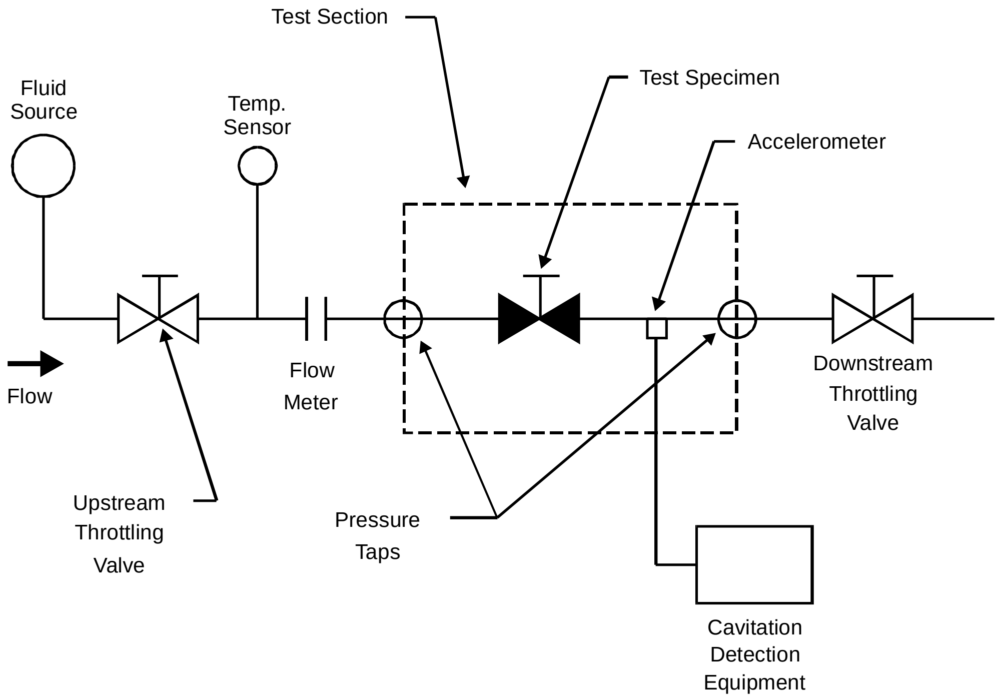

Figure 2 — Flow test system

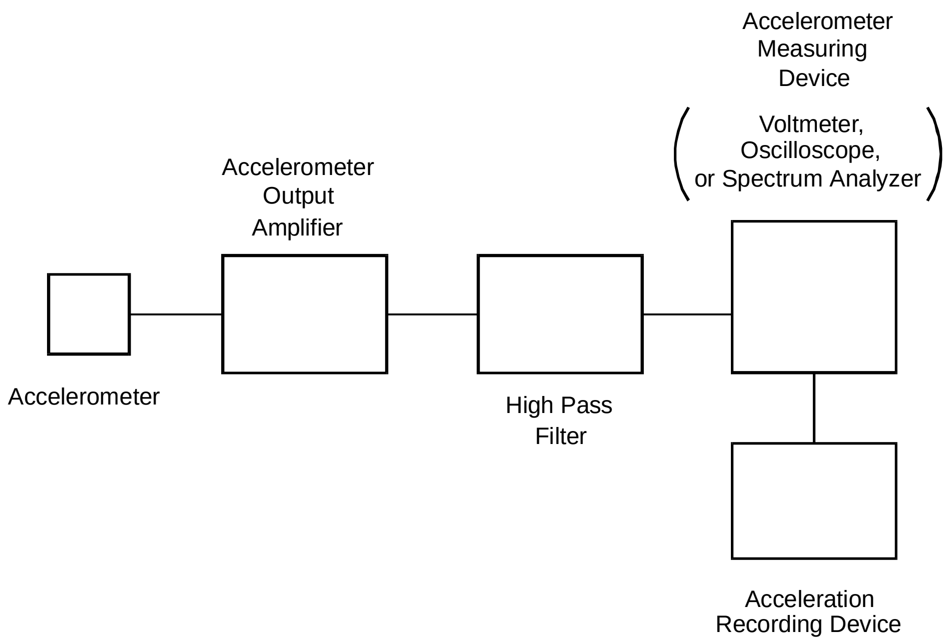

Figure 3 — Cavitation detection equipment

8.2.2 Test specimen

The test specimen can be any valve or test apparatus for which test data are required. The initial test specimen for any laboratory should be the calibration test section shown in Figure 4. The test results from the test orifice plate shall be recorded and shall be within the calibration limits in Section 8.6 to qualify the laboratory for testing to this recommended practice.

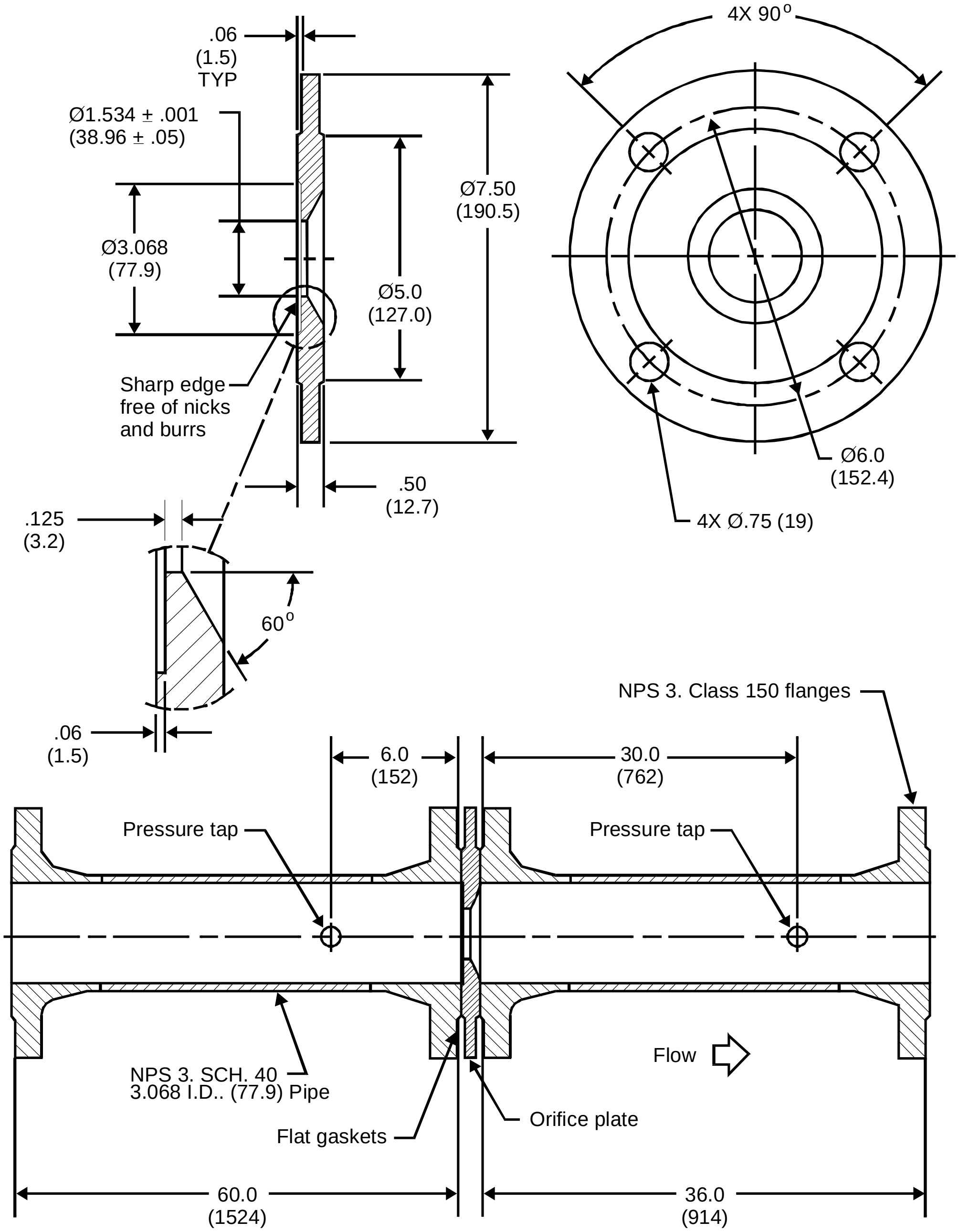

Figure 4 — Cavitation calibration test manifold

8.2.3 Test section

The test specimen upstream and downstream piping shall conform to the nominal size of the test specimen connection and to the following length requirements.

The upstream pressure tap shall be two nominal pipe diameters from the test specimen connection, while the downstream pressure tap shall be six nominal pipe diameters from the test specimen connection. There shall be at least 18 nominal pipe diameters of straight pipe (eight if straightening vanes are used) upstream of the upstream pressure tap, and at least one pipe diameter of straight pipe downstream of the downstream pressure tap.

An effort should be made to match the inside diameter at the inlet and outlet of the test specimen with the inside diameter of the adjacent piping.

> **NOTE —** For valves with $C_v/N_1d^2 > 20$, see Annex D.

8.2.4 Throttling valves

The upstream and downstream throttling valves are used to control the pressure differential across the test section pressure taps and to maintain a specific downstream pressure. The downstream valve should be of sufficient capacity to ensure that choked flow (in the case of the standard calibration test manifold) and the other desired cavitation coefficients (σ_i, σ_c, σ_{mv}) can be achieved. Care should be taken to assure that noise or cavitation from these valves during the testing does not influence the test specimen results.

8.2.5 Flow measurement

The flow measuring instrument may be any device that meets the required accuracy. This instrument shall measure the true time average flow rate within an error not exceeding ± 2% of the actual value. The resolution and repeatability of the instrument shall be within ±0.5%.

8.2.6 Pressure taps

Pressure taps shall be provided on the test section piping in accordance with the requirements of 8.2.3 and shall conform to the construction described in ISA-S75.02.

8.2.7 Pressure measurement

All pressure and pressure differential measurements shall be made to an error not exceeding ±2%.

8.2.8 Temperature measurement

All temperature measurements shall be made to an error not exceeding ± 2 °F (1.1 °C).

8.2.9 Accelerometer installation

An accelerometer shall be rigidly mounted on the test specimen downstream pipe wall. The main sensitivity axis of the accelerometer shall be perpendicular to the pipe axis. The exact location on the downstream pipe should be determined by test (to obtain maximum vibration sensing); however, one nominal pipe diameter downstream of the test specimen connection is a good starting point. As the higher frequency vibrations (5-50 kHz) are of interest in this testing, mounting on the lower mass pipe wall (as compared to the flange mounting) will provide improved sensitivity from a high frequency accelerometer.

8.2.10 Vibration measurement

Although this testing does not require accurate quantitative acceleration results (±5% required), a piezoelectric or other accelerometer that has a high enough resonant frequency (> 100 kHz) should be used. This will assure consistent results for the full frequency range being analyzed (5-50 kHz). A vibration preamplifier should be used as recommended by the accelerometer manufacturer. For ease of data analysis, an optional high pass filter can be used to differentiate the low frequency noise (< 5 kHz) resulting from background and turbulent flow and the high frequency noise resulting from cavitation. Figure 3 depicts suggested instrumentation.

The exact frequency range of interest for each valve type may vary, and it is the responsibility of the valve tester to determine the optimum range of frequencies for evaluation.

8.2.11 Installation of test specimen

The alignment between the centerline of the test section piping and the centerline of the inlet and outlet of the test specimen shall be within 1/16-inch (1.6 mm) for pipe sizes up to NPS 6 (150 mm), and within 1% of the diameter for pipe sizes NPS 8 (200 mm) and larger.

When rotary valves are being tested, the valve shaft shall be aligned with the test section pressure taps. All gaskets should be positioned so that they do not protrude into the flow stream.

8.3 Test fluid

Water at relatively constant temperature shall be the basic fluid used in this test procedure. The water shall be sufficiently free of suspended particles, air, or other gases so as not to affect the test results. Successful completion of the laboratory qualification testing per Section 8.6 of this procedure will verify the suitability of the test fluid. Inhibitors may be used to prevent or retard corrosion or to prevent the growth of organic matter. The effect of additives on density, vapor pressure, and viscosity shall be verified.

8.4 Test procedure

The procedure for evaluation of the data collected in these tests is given in 8.5.

8.4.1 The test section shown in Figure 2 shall be used with the test specimen (valve) set at a specified travel. Travel positions of 20%, 40%, 60%, 80%, and 100% rated travel, as a minimum, shall be used for a valve, and the following test should be run for each travel position. Since the parameters being tested are a direct function of valve geometry, additional travel positions for some valves may be needed. (Figure 1 only applies to a single open position.)

8.4.2 The downstream throttling valve shall be in the fully open position, or in a travel position that provides for cavitating conditions in the test valve. Then, with a preselected upstream pressure, the flow rate, upstream pressure, downstream or differential pressure across the test valve, and accelerometer levels shall be recorded. The use of differential pressure measuring instruments is preferred whenever possible to ensure accuracy. During a test run, P_1 shall be held constant to within ± 5%. This establishes a maximum pressure differential for the test valve in a test system at a particular travel setting.

> **CAUTION — DO NOT EXCEED THE MAXIMUM PRESSURE DROP RATING OF THE VALVE BEING TESTED. CARE SHOULD BE TAKEN TO ASSURE THAT GAS IS NOT TRAPPED IN THE DOWNSTREAM PRESSURE TAPS.**

8.4.3 A series of additional tests shall be made at subsequently decreasing pressure drops by throttling the downstream throttling valve. Each test should decrease the test specimen pressure drop by suitable increments to detect inflection points in the vibration curve; and the same recordings of flow, pressure, and acceleration shall be made.

8.4.4 The following data shall be recorded:

a) Valve travel — measurement error shall not exceed ± 0.5% of rated travel;

b) Upstream pressure (P_1) — instrument measurement error shall not exceed ± 2% of the actual value;

c) Pressure drop (ΔP) (preferred) or downstream pressure (P_2) — instrument measurement error shall not exceed ± 2% of the actual value;

d) Measured flow rate (q) — measurement error shall not exceed ± 2% of the actual value;

e) Fluid temperature (T) — measurement error shall not exceed ± 2 °F (1.1 °C). Care should be taken to monitor and record changes in fluid temperature so that vapor pressure can be properly determined for all test points. This is especially important for recirculating test systems in which fluid temperature may increase during a test.

f) Downstream pipe wall vibration — measurement error shall not exceed ±5% of full scale. Instrument full scale should be selected for optimum measurement of the range of vibration amplitudes.

g) Barometric pressure — measurement error shall not exceed ± 2% of the actual value. Convert to absolute pressure (P_a), psia (kPa).

8.5 Data evaluation

For each test:

a) Evaluate C_v and F_L in accordance with ISA-S75.02 or Annex D, as applicable.

b) Calculate sigma cavitation index for each test point, preferably using differential pressure measurement data.

$$\sigma = (P_1 - P_v) / (\Delta P) \tag{14}$$

c) Calculate the average acceleration in G, ft/s^2 (m/s^2), or equivalent to as high a frequency as permitted by the instrumentation within the frequency range 5-50 kHz. These data can be obtained from a frequency analyzer, oscilloscope, or voltmeter (with a high-pass filter).

d) Obtain cavitation coefficients by making a log/log (or semi-log) plot of acceleration versus sigma (σ) for each test made at a particular travel. A separate plot should be made for each travel position tested.

From these plots, determine cavitation curve inflection points σ_i, σ_c, and σ_{mv} where possible by intersection of the straight line segments, Regimes I, II, III, and IV, as shown in Figure 1. Note that some valves will not exhibit all inflection points in the test data which are always subject to expert interpretation; and some multistage valves may not provide meaningful data by this test method alone.

8.6 Laboratory qualification

8.6.1 To assure that a particular laboratory can successfully test a piece of equipment to this procedure, a qualification test is recommended. The test procedure and data evaluation of 8.4 and 8.5 shall be performed on the calibration test manifold shown in Figure 4, using 3-inch test section piping. A P_1 of 100 psig (690 kPa) is recommended as a test pressure. The resultant sigma values shall be within the following acceptable ranges to qualify the laboratory and equipment.

$$\sigma_i = 2.7 \pm 5\%$$

$$\sigma_c = 2.3 \pm 5\%$$

$$\sigma_{mv} = 1.4 \pm 25\%$$

$$C_v = 52 \pm 5\%$$

$$F_L = 0.86 \pm 5\%$$

8.6.2 When use of the 3-inch piping manifold is impractical, alternate pipe sizes (having inside diameter D_1) may be used for calibration test manifolds if the following manifold design conditions are met:

a) the ratio of orifice diameter to pipe inside diameter is 0.50 ± 1%;

b) a minimum of twenty (20) nominal pipe diameters of straight pipe is provided upstream of the orifice;

c) a minimum of twelve (12) nominal pipe diameters of straight pipe is provided downstream of the orifice;

d) upstream and downstream pressure taps are located two (2) nominal pipe diameters upstream and ten (10) nominal pipe diameters downstream of the orifice; and

e) the orifice is a conventional sharp-edged design, (similar in construction to the orifice in Figure 4, to match with manifold piping), thick enough to resist vibration and pressure forces from cavitation testing, and approximately centered in the pipe.

When other calibration manifold sizes are used for testing, size scale effects must be considered (Equations 2 through 5), assuming PSE = 1.00. The alternate piping test values shall be within the following acceptable ranges (which include adjustment for size scale effect) to qualify the laboratory and equipment.

$$\sigma_i = 2.7[(D_1/3.068N_3)^{0.104}] \pm 5\% \tag{15}$$

$$\sigma_c = 2.3[(D_1/3.068N_3)^{0.104}] \pm 5\% \tag{16}$$

$$\sigma_{mv} = 1.4 \pm 25\%$$

$$C_v/N_1D_1^2 = 5.52 \pm 5\%$$

$$F_L = 0.86 \pm 5\%$$

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Annex B — Cavitation fundamentals

B.1 Introduction

B.1.1 Cavitation is a subject that has been of interest to both the theorist and the industrial practitioner for nearly a hundred years. Unfortunately, many aspects of the cavitation process remain a frustrating mystery in spite of intense study during this time. Theory has supplied much understanding of the behavior of single cavities as well as the nature of the influence of many variables. Likewise, contending with cavitation in pumps, valves, propellers and hydrofoils has spawned many "rules-of-thumb" governing prediction and control practices. However, the state-of-the-art does not currently offer a satisfactory technology that bridges the gap between theory and practice.

B.1.2 The purpose of this section is to provide more insight into the cavitation events from both a theoretical and a practical perspective. This level of familiarity provides a broad foundation for dealing with cavitation in the process control industry. Not only does it supply the generally agreed upon quantitative methods, it also establishes a background on which to base the inevitable engineering judgments that characterize this type of application.

B.2 Cavity behavior

B.2.1 Cavity dynamics are governed by changes in fluid pressure and can be very complex. An analysis of even a single cavity in a well defined environment requires methods of thermodynamics, heat transfer, mass diffusion, fluid mechanics, statics, and dynamics. A detailed discussion is obviously not in keeping with the overview nature of this document. However, some of the salient points on the subject of cavity inception, growth, and collapse are presented below.

B.2.2 The pressure level conventionally associated with the onset of vaporization of a liquid is the thermodynamic vapor pressure. This model holds that a portion of the liquid will vaporize when the prevailing fluid pressure decreases below the vapor pressure. This vaporization occurs at a "weak" spot within the liquid continuum. This weak spot is usually associated with a free surface within the liquid that was created by an entrained foreign particle, microscopic gas bubble or other such "nucleus." In the absence of such nuclei, vaporization would not occur at the thermodynamic vapor pressure of the fluid. Very large fluid forces would be required to overcome the effects of surface tension at the radius of an infinitely small bubble within a pure liquid.

B.2.3 Since nuclei are required to promote vaporization at the vapor pressure of the liquid, the subject of nuclei content naturally arises. In most industrial applications an ample supply of such nuclei are present in the form of various contaminants. Even in the case of highly purified liquids, small amounts of gas can be stably entrained in the liquid and the surrounding solid surfaces. Thus, for practical intents and purposes the fluid can be assumed to vaporize at the vapor pressure of the liquid.

B.2.4 This model assumes thermodynamic equilibrium and, therefore, is accurate only when the pressure change rate is slow enough that sufficient time exists for exchange of the various thermal energy quantities. When the pressure changes are very large and occur very rapidly, it is possible for non-equilibrium or "meta-stable" thermodynamic states to exist. For the specific case at hand, this would result in the fluid existing in the liquid state under pressure and temperature conditions that ordinarily correspond to a saturated state.

B.2.5 When non-equilibrium conditions occurs, the fluid pressure may actually drop below the vapor pressure of the fluid with little or no vaporization occurring. As the thermal dynamics "catch up" to the inertial dynamics, vaporization occurs very rapidly (physically displaced slightly downstream from the location where the local pressure actually drops below the vapor pressure).

B.2.6 A forming cavity experiences a period of stable initial growth. During this growth phase the force balance on the cavity is such that energy is required for sustained growth. However, when the cavity attains a "critical" size, this growth is no longer stable. Now the force balance is such that energy is liberated on additional growth. Under these circumstances the growth is "explosive" (Knapp, et al., ref. 1).

B.2.7 If the fluid pressure should rise above the vapor pressure, the reverse process will occur; i.e., cavity growth will cease and collapse of the cavity and condensation of the vapor will occur. This latter event occurs very rapidly and is the primary source of the noise, vibration and material damage. In some cases additional growth and collapse cycles, known as rebound, occur.

B.2.8 At this point it is appropriate to mention another phenomenon, which is outwardly similar to cavitation but requires distinction. This is the phenomenon of "out-gassing." Pockets of gas may form when dissolved gases (and possibly the fluid's own vapors) come out of solution upon depressurization of the fluid. Sometimes this is referred to as "gaseous cavitation." It may occur separately from the aforementioned type (i.e., vaporous cavitation) or in combination with it. The behavior of this type of cavity formation is markedly different. It generally is considered less detrimental, because the growth, and particularly the collapse rates, are significantly reduced.

B.3 Damage mechanisms

B.3.1 Material damage is a side effect of cavitation that receives considerable attention for obvious reasons. Several theories have been offered to explain the observed levels of damage; however, no single mechanism is currently considered universal. It appears that all of the identified theories apply to varying degrees.

B.3.2 The microjet theory asserts that vapor cavities collapse asymmetrically and, in doing so, set up a very small, high velocity liquid jet (Knapp & Hollander, ref. 8). This jet, if it impinges on a material surface, imparts damage by eroding away a portion of the material. This theory has been substantiated with both high speed photographic observations of collapsing cavities, and also numerical studies of individual cavity collapses. Interestingly, the preferred orientation of this jet is influenced by the nature of the various boundaries in close proximity to the jet. Rigid surfaces tend to attract the microjet, while compliant surfaces appear to repel the microjet (Knapp, et al., ref. 1). This appears to be a dominant damage mechanism on the initial collapse of travelling cavities.

B.3.3 Another theory, the shock wave theory, advances the notion that shock waves are established by the rapid movement of the vapor-liquid interface during collapse. The compressive loading from shock wave impingement on the material surface can result in failure by either fatigue (multiple loadings) or plastic deformation (single loading). This appears to be characteristic of the symmetrical collapse mode, typical of rebound collapses.

B.3.4 These two forms of cavitation attack constitute the mechanical component of attack. The damage resulting from either of these types of mechanical attack possesses some similar attributes.

The physical appearance of cavitation damage varies from a "frosted glass" appearance to a rough, cinder-like surface texture. Also, the collapsing cavity must be near the material surface to cause any damage, regardless of the exact mechanism of damage. Both the shock wave intensity and the microjet dissipate to a harmless level within a fairly short propagation distance.

B.3.5 Finally, chemical attack or corrosion can occur simultaneously with mechanical attack. This form of attack usually is considered secondary. It is important in that it tends to accelerate or reinforce the mechanical attack process. The extent to which this form of attack is a factor depends on the chemical compatibility of the process fluid and the materials of valve construction.

B.3.6 The combined effects of mechanical attack, chemical attack, and the particular material of construction give rise to an "incubation period" in some circumstances. This is a period of time (that varies with changing conditions) during which no material loss is apparent. Plastic deformation or mass loss then commences, even though the cavitation intensity remains constant.

B.4 Pressure dynamics that cause cavitation

B.4.1 The pressure changes that cause cavitation arise from the flow of the liquid through the control valve. However, pressures sufficiently low to cause cavitation may result from different sources. The conventional industrial treatise usually accounts for only the mean fluid pressure. Pressure changes associated with various boundary layer phenomena also play an important role in industrial cavitation.

B.4.2 Cavitation in control valves begins from highly localized pressure reductions typically associated with boundary layer separation. Flow streams in control valves are characterized by irregularly shaped passages, abrupt cross-sectional area changes and various protrusions. These features subject the fluid to adverse pressure gradients that in turn cause the flow stream to separate from the bounding surface. The resulting high turbulence and re-entrant vortex formation produce very low pressures that initiate cavitation. This can occur even though the mean fluid pressure at the vena contracta is greater than the fluid vapor pressure.

B.4.3 Mean fluid pressure changes are associated with changes in the kinetic energy of the fluid as it passes through the valve. The fluid velocity increases as the liquid moves into the reduced area of the throttling restriction. This causes a corresponding decrease in the mean fluid pressure. Conversely, when the fluid exits the throat region into a larger cross-sectional flow area, the velocity decreases and the mean fluid pressure is partially restored. Thus, a minimum pressure exists in the valve in the general vicinity of the minimum flow area. If this mean pressure falls below the vapor pressure of the fluid, the degree and extent of cavitation increases.

B.4.4 Both of these sources of cavitation (i.e., mean pressure changes and localized pressure changes) are important. It cannot be categorically stated that one dominates the other throughout the cavitating flow regime. The total behavior of the cavitating liquid is heavily dependent on valve style and other factors. This document will not distinguish between the origins of cavitation in further discussion of the subject matter.

B.5 Cavitation parameters

B.5.1 As noted in Section 5, the complexity of cavitation and its sensitivity to a variety of factors (discussed later) render it difficult to quantify cavitation in terms of a single or few parameters. Nonetheless, it is necessary to construct a mathematical representation of the application in order to meaningfully deal with it during process design. In the case of control valves it is conventional to describe the application in terms of a single parameter and then to compare this parameter to different limits of operation for a given control element.

B.5.2 The parameter selected for use in this document was given in Section 5 as

$$\sigma = (P_1 - P_v)/(P_1 - P_2) \tag{B.1}$$

where, P_1 is the upstream absolute pressure of the valve, P_2 is the downstream absolute pressure of the valve, and P_v is the absolute vapor pressure of the fluid at the inlet temperature. For high recovery valves with C_v/N_1d^2 > 20, see Annex D.

B.5.3 The cavitation index sigma (σ) is a form of another dimensionless parameter, the Euler Number. Sigma is constant for either SI or US Customary units as long as the same units of pressure are used throughout the equations for the parameter. This index (σ) is the ratio of fluid forces trying to prevent cavitation (the system or service pressure) to the forces trying to cause cavitation (the pressure drop). The smaller the value of the cavitation index for a flow system, the more likely or the more severe cavitation will be.

B.5.4 Different cavitation parameters have been used for defining cavitation in control valves (Hutchison, ref. 9; IEC, ref. 21). Parameters such as K or x_F may be defined as

$$K = x_F = 1/\sigma = (P_1 - P_2)/(P_1 - P_v) \tag{B.2}$$

The operational limiting value of K is often referred to as K_c (Hutchison, ref. 9). This limit was originally defined as the set of conditions where cavitation will begin to affect the flow pattern in the valve. This manifests itself as apparent decrease in the flow coefficient of the valve. Some publications have referred to this as the incipient cavitation point. However, for many valves, cavitation damage, heavy vibration, and noise could occur for flows and pressure drops less than that for this definition of K_c (NRC, ref. 22). Another parameter used to indicate incipient cavitation for the purpose of predicting noise levels is x_{Fz} (IEC, ref. 21). While x_{Fz} is approximately equivalent to $1/\sigma_i$ evaluated for the same valve, size, opening, and test pressure, equations contained in this document are rigorously developed only for σ_i. However, for purposes of evaluating control valves for cavitation, the value of $1/x_{Fz}$ may be used for σ_i. Users are advised to check with the valve manufacturer for the definition and proper use of any cavitation parameter.

B.5.5 Sigma (σ) is recommended because of numerous performance correlations published using σ (refs. 1-7, 11-13, 16, 20, 22) and because of the confusion that exists both nationally and internationally over the use and definitions of the K-type parameters. Sigma will typically range from 1 to 15. As a rule of thumb, a valve with service conditions corresponding to a sigma value greater than 15 for low pressure loss valves or greater than 8 for high pressure loss valves usually will not experience cavitation.

B.5.6 Alternate forms of the sigma index also exist. One particular expression that has been used in the past is based on the relationship of the vapor pressure to the downstream pressure (as opposed to the upstream pressure). The following equation can be used to convert between these different forms of σ:

$$\sigma = 1 + \sigma_2 \tag{B.3}$$

where

$$\sigma_2 = \frac{P_2 - P_v}{P_1 - P_2}$$

B.5.7 The pressure recovery factor, F_L, is a coefficient that is useful for predicting the choked flow rate through a control valve. However, because of its relationship to the pressure recovery characteristics of the valve, it also has been used as a cavitation limit. This does not constitute a correct use of this factor. In fact, when the pressure drop meets or exceeds the pressure drop calculated from F_L, substantial levels of cavitation can already exist. The significance of this parameter to the analysis of control valve cavitation is discussed further in B.6.3 and B.8.4.

B.6 Levels of control valve cavitation

B.6.1 The cavitation index, σ, by itself does not convey any information about the performance of a particular valve in a particular application. Different valves can tolerate different levels of cavitation, and different applications are concerned about different aspects of cavitation (for instance noise versus vibration versus damage). Therefore, σ must be evaluated at the service conditions (service σ) and then compared to some benchmark for the valve that reflects the permissible degree of cavitation (limiting σ) for the application.

B.6.2 The degree and extent of cavitation in a control valve will be a function of many things. Clearly, it is a function of the valve style and valve opening since these factors directly affect the character of the flow through the valve. It follows that the desired benchmarks (limiting σ coefficients) will be unique to a given valve style and opening, if not a particular valve. A "good" benchmark will delineate a meaningful cavitation threshold as well as be measurable. However, in the quest for practical benchmarks one or the other of these two characteristics should not be overemphasized. It is possible to define a level of cavitation that is convenient to measure but has little meaning to valve selection. On the other hand, it may be desirable to have a benchmark that predicts a particular event, such as damage, but is difficult to accurately measure or predict.

B.6.3 The following levels of cavitation, which often serve as benchmarks for the valve selection process, are believed to represent a good blend of measurability and meaning. An additional parameter, which can be based on valve manufacturers experience with a particular valve, is also included. The corresponding σ coefficients are shown in parenthesis.

B.6.3.1 Incipient cavitation (σ_i)

The first level of cavitation is incipient cavitation, σ_i. This level is associated with the flow conditions for which cavitation can first be detected. A typical means of establishing this condition is by measuring increased SPL (IEC, ref. 21) or vibration (Tullis, ref. 2; Riveland, ref. 10). Incipient cavitation is intermittent and usually is the result of a few isolated collapses of vapor pockets very near the source of the flow separation. Incipient cavitation is extremely mild, and often cannot be heard over the flow noise and vibration produced by other components in a piping system.

B.6.3.2 Constant cavitation (σ_c)

Constant cavitation, σ_c, is one of the earliest levels of cavitation in which the intensity has noticeably increased above the incipient level (Ball and Tullis, ref. 11). At this point the cavitation involves a sufficiently large volume of vapor to produce a uniform and constant level of cavitation that is readily detected. The collapse region is beginning to move downstream from the point of cavitation inception within the valve. The observable effects of constant cavitation are light vibration and mild noise levels. Usually there is no undesirable damage associated with this level of cavitation. However, as the flow rate is increased, cavitation increases above this level at a steady and moderate rate. Constant cavitation can be determined in a flow laboratory by comparing the relationship of the acceleration or vibration produced by cavitation with different flow rates.

B.6.3.3 Incipient damage (σ_{id})

Incipient damage, σ_{id}, is the level of cavitation in which an increase in cavitation intensity first produces any detectable damage to either the valve or the downstream piping. This level is much more difficult to detect. It cannot be determined from an acceleration or vibration curve as other points might be. A suggested method is to measure the pitting rate of damage on samples of soft materials such as aluminum (Rahmeyer, ref. 12). The pitting rate of one pit per minute per square inch in aluminum is a good limit since it represents the most mild and conservative stage of cavitation damage. The rate and magnitude of damage have been found to rapidly increase with velocity after the level of incipient damage in aluminum.

B.6.3.4 Choking cavitation (σ_{ch})

"Choking" cavitation, σ_{ch}, is a most severe level of cavitation. Under "fully choked" flow conditions an additional decrease in the downstream pressure will not increase the flow rate through the valve at a given inlet pressure. The value of σ associated with this condition may be estimated from the following equation:

$$\sigma_{ch} = \frac{(P_1 - P_v)}{F_L^2\,(P_1 - F_F P_v)} \tag{B.4}$$

This relationship is only an approximation of the fully choked condition. In control valve sizing this condition usually is associated with the following pressure differential:

$$\Delta P_{\text{choked}} = (F_L)^2\,(P_1 - F_F P_v)$$

However, the choking "process" actually occurs over a range of pressure drops as a consequence of the compressibility changes accompanying vapor formation. The <u>fully choked</u> condition may actually occur at a pressure differential slightly larger than that calculated by this equation.

The maximum levels of noise, vibration, and material damage have been observed to occur at or just prior to this condition. This level of cavitation usually is avoided in most applications. However, consideration of other offsetting factors such as trim design or reduced flow rate, size or pressure may justify operation at this level without significant problems. These factors are discussed later.

B.6.3.5 Maximum vibration cavitation (σ_{mv})

Maximum vibration cavitation, σ_{mv}, is a level of cavitation corresponding to the "peak" of the characterizing vibration curve. As noted in the preceding paragraph, maximum vibration, noise, material damage, and flow rate all have been observed to occur very close to the same value of σ. However, this is not to imply that they are necessarily coincident points. Use of a noise-based laboratory cavitation detection method may yield a slightly different maximum noise coefficient. Categorically, these points may be utilized in somewhat similar fashion; they all represent a severe level of cavitation that is ordinarily avoided. On occasion, however, certain mitigating factors may justify operation in this region. As noted above, these factors are discussed later.

B.6.3.6 Manufacturer's recommended limit (σ_{mr})

Manufacturer's recommended limit, σ_{mr}, is an operational limit supplied by the valve manufacturer. The evaluation of this point may be based on factors other than a single laboratory test, such as accumulated experience with a particular design in a particular application or understanding of specific design features. It may or may not coincide with one of the levels of cavitation already discussed. This σ_{mr} limit concept has been introduced to allow the most effective use of available control valve hardware and knowledge. The valve user should seek clarifications from the valve manufacturer of the information used in this evaluation.

B.6.4 The general evaluation of these different levels of cavitation is the subject of other sections of this document. (See Sections 6, 7, and 8 for more information.)

B.7 Control valve design and application factors

B.7.1 Ostensibly, the sizing procedure would appear complete with the introduction of the various cavitation levels and their coefficients. However, in actuality, several issues still confront the potential user. Examples of these issues follow:

a) What factors affect the choice of σ_{mr} limits?

b) What factors might affect the actual value of cavitation coefficients?

c) What factors might affect the value of the apparent operating σ index?

B.7.2 The different levels of cavitation described in the preceding paragraphs merely define different significant cavitation conditions that exist. The appropriate coefficient to use for a given application is not always self-evident and usually embraces a degree of subjectivity. In addition to the service conditions, factors such as valve style and opening, duty cycle, location, desired life and past experience should be considered (Barnes and Cain, ref. 13). There are no firm guidelines for this decision, and the valve manufacturer should be consulted in this matter. Additional discussion may be found in Section 7 and Annex C of this document.

B.7.3 Factors that affect the numerical values of the coefficients established by test, and subsequently the level of cavitation, may be broadly grouped into valve factors (associated with the style of the control valve) and service factors (e.g., pressure, temperature, fluid, and flow rate).

B.7.4 Accounting for all of these various effects can be difficult and controversial. The extent of cavitation-related problems is a function of the total amount and intensity of the cavitation formed in the flow stream, and of the response of the surroundings (material surfaces, acoustic field, etc.) to that cavitation. Research has shown that many factors contribute to the total nature of cavitation and to the resulting problems.

B.7.5 To aid in understanding the nature of some of these effects, it is helpful to divide them into two categories that will be labeled "scale effects" and "influences." Scale effects are factors that change the numerical value of any of the defined cavitation coefficients as that factor is varied. "Influences," on the other hand, are those factors that do not change the numerical value of the defined cavitation coefficients, but may affect the intensity of cavitation or cavitation damage. The differences in these two types of effects are depicted in Figures B.1 and B.2. Knowledge of potential factors in either category is necessary in both the testing and application phases. Thus, it is necessary to sufficiently standardize the testing procedure and supply correcting equations for application purposes where possible. In spite of the efforts and advancements of research in this area, a degree of subjectivity still is present in the application process.

B.7.6 Two very important scale effects are associated with the cavitation parameter σ: the pressure scale effect, PSE, and the size scale effect, SSE. The various cavitation coefficients are not independent of or constant with either different upstream pressures or the valve size. Failure to account for the difference between the actual service pressure and size and the test values (reference values) can introduce significant errors. Thus, it becomes important to conduct all σ evaluation tests at a prescribed or known reference condition and scale the values to the service conditions with a known scaling equation.

B.8 Effects on σ by size and other control valve geometry

B.8.1 The various cavitation coefficients, as well as the suitability of the valve to operate at a given level, are strong functions of the geometry of the control valve. It can be expected that different values of any given coefficient will be associated with different valve styles, or even different openings of the same valve. Further, it can be expected that different levels of cavitation are appropriate limits for different valve styles. Some of the major factors that have an effect on σ values are discussed below.

B.8.2 Perhaps one of the most self-evident geometric features is the physical size of the valve. There are strong size scale effects associated with most of the various cavitation levels discussed earlier. In general, as the size of the valve increases, the numerical value of the cavitation coefficient increases for a true scale model. An understanding of the general behavior of broad classes of control valves has lead to the development of scaling equations. Thus, if cavitation coefficient information is available from test data for a given valve, application limits can be conditionally estimated for different size valves of the same style (Tullis, ref. 2).

B.8.3 Size scale effects need to be accounted for only when a reduced scale prototype has been used to establish the various cavitation coefficients for the full size valve. It is important that scaling is done from an actual scale model of the valve in question. Use of a smaller size valve model of a particular valve style line will introduce geometric dissimilarities. These may have different effects in the installed valve performance, which are not reflected in the scaled results. Additional information on these considerations is supplied in Section 6.

B.8.4 Another characteristic that is different from valve to valve is the pressure recovery characteristic. As noted previously, this often is associated with cavitation. This is not altogether inappropriate, but too much emphasis often has been placed on this one feature. Qualitatively, pressure recovery characteristics help distinguish valves inherently better suited for cavitating applications. Low recovery valves maintain higher overall pressures within the valve body under otherwise equal conditions. Therefore, the tendency for the valve to cavitate, and the intensity of any cavitation that may occur are both reduced.

B.8.5 The degree of pressure recovery is heavily dependent on the valve geometry and is characterized by the pressure recovery factor, F_L. Note, however, that this is not a cavitation index. The singular quantitative use of this parameter is to determine the maximum flow (i.e., choked flow) for the valve under a given set of conditions.

B.8.6 Trends in the pressure recovery characteristic usually are reflected in the other σ values accordingly. High recovery valves have the effect of relatively increasing the numerical value of corresponding levels of cavitation compared to a low recovery valve. However, no verified functional relationship between the cavitation coefficients and F_L exists, with the exception of that between σ_{ch} and F_L. Actual σ coefficients should be obtained for each valve and should not be predicted from the pressure recovery factor.

B.8.7 Another valve geometry consideration, which has a pronounced effect on the magnitude of the various σ values, is the shape and number of primary restrictions in the control valve. Highly irregular cross-sectional flow areas (e.g., such as the "lens shape" associated with certain rotary valves) may result in substantial amounts of cavitation at lower operating sigmas. Cavitation may begin in one localized region (for the reasons cited earlier) and gradually develop. Ultimately the cavitation will extend throughout most of the entire flow area. The compressibility effects of the forming vapor on the overall flow characteristics may be seen in the associated flow curve

(Figure B.3). Typically such valves begin departing from the "straight line" portion of the curve at much lower pressure differentials relative to the choked flow point.

B.8.8 The typical butterfly valve geometry creates two flow paths of differing geometries. Each flow path possesses individual flow parameters and passes flow accordingly. One path may choke prior to another and give rise to a "segmented" flow curve as shown in Figure B.4. The slope of the flow curve is proportional to the magnitude of the C_V value associated with the valve. The steep slope of the first segment represents the full flowing capability of both flow paths. The second segment has a slightly lower slope as a result of one flow path choking, and subsequently flowing a fixed amount regardless of the pressure drop. The third segment has zero slope, since both flow paths are choked and flowing fixed volumes of liquid. "Safe" operating limits may be tied to the flow path that cavitates first and may consequently be lower than anticipated when considering the choked flow limit. The degree of this effect will be a function of the valve opening.

B.8.9 Certain control valves capitalize on particular geometric effects and are designed especially for cavitation service. These designs may include special or proprietary features that enhance the numerical σ values of the various cavitation levels and the acceptable operating limit. Other times they control where and to what extent the cavitation occurs. This allows a less conservative limit to be used in the valve selection process. For instance, consider the case when σ_c is chosen as an appropriate operating limit when considering a butterfly valve for a particular application. However, the use of a valve specially designed to control cavitation may allow the choice of σ_{ch} as the limit for that same application.

B.8.10 A note of caution is appropriate at this point. The σ values of the various cavitation levels are clearly a function of the valve style and valve opening. To a certain extent this can be exploited when selecting valves for specific applications. However, the user must resist the temptation to arbitrarily change styles and sizes to effect favorable numbers. For example, the use of a large butterfly valve at small openings in lieu of smaller valves near wide open could result in additional damage to seating and shutoff surfaces from throttling too close to the seat. More factors to consider are discussed in the sections that follow.

B.9 Pressure scale effects and other application influences on σ

B.9.1 The foremost scale effect associated with specific application service conditions is the pressure scale effect (PSE). Similar to the size scale effect, the actual σ value of the various cavitation levels will change as the operating pressure changes. In general, as the service pressures increase, the numerical σ value of the cavitation level also increases for a given valve. Empirical scaling equations based on general behavior of broad classes of control valves are available. Thus, if σ values are available from test data at a specific pressure, application limits can be estimated for other service conditions.

B.9.2 Other factors, particularly fluid properties, can have an effect on the degree and extent of cavitation, even though the numerical σ value of a cavitation level is unchanged. Several prominent considerations follow:

a) viscosity
b) dissolved gas volume
c) surface tension
d) "thermal" properties
e) fluid composition

B.9.3 The influence of viscosity usually is considered to be one of damping, although for most practical applications the effect is negligible (Ivany & Hammitt, ref. 14). Again, the numerical value of a particular σ will not change. No analytical or empirical method for accounting for viscosity has been developed. However, high viscosity fluids are not likely to present significant cavitation problems.

B.9.4 Undissolved gases can have a pronounced influence on the behavior of cavitation. Small additional amounts of undissolved gases provide additional nuclei for the inception process and, in theory, thus result in "additional" cavitation (Knapp et al., ref. 1). However, larger amounts (i.e., a few percent by volume) tend to suppress the effects of cavitation by disrupting the collapse process (Mousson, ref. 15). As emphasized in the section on testing, it is necessary to assure minimal air content so that the test results are not obscured by this influence.

B.9.5 The surface tension term appears in the theoretical equations that describe the cavity behavior. It acts to restrain nucleation and growth, yet promotes cavity collapse. The former effect tends to diminish the total of cavitation that occurs and is a mitigating effect. The latter has the opposite effect by intensifying the collapse. It is likely that the magnitude of the restraining effect would dominate (Hammitt, ref. 16); however, for most fluids of practical interest, the total effect probably is negligible.

B.9.6 Most thermal property influences are accounted for by utilizing the absolute vapor pressure (at service condition temperature) in the sizing and scaling equations. There are some additional effects that are less quantifiable. When the thermal energy terms are negligible compared to the inertial terms, the pressure dynamics govern the rate at which vaporization and collapse occur. When the thermal terms become significant, the time associated with the rate of heat transfer can become the controlling factor, and the process becomes more like boiling (Stepanoff, ref. 17). However, most fluids of industrial interest usually do not fall into this category.

B.9.7 The discussion to this point has concerned fluids that are comprised of a single chemical species. Many process fluids are actually mixtures of different fluids. The fluid composition of such mixtures can have a direct bearing on the degree to which cavitation-related problems occur. When the process fluid consists of a mixture of fluids with widely varying vapor pressures, the classical "single fluid" model breaks down. Vaporization occurs over a range of pressures—as opposed to the constant pressure, constant temperature vaporization of a "pure" liquid. Furthermore, the compositions of both the vapor and liquid phases change as a result of this fractionation. The combined effect is, in general, to reduce the severity of cavitation-related problems. A good example of a mixture of fluids is a broad mixture of hydrocarbons.

B.10 Closure

B.10.1 Cavitation will continue to be a major problem in industrial process control. An understanding of the nature of the subject and utilization of current quantitative information will aid in formulating effective problem abatement. However, the ultimate benefit of analyzing valves and service conditions for cavitation depends upon the quality and completeness of service and valve information available. Valve users and manufacturers should make every reasonable effort to share clear and accurate data (see ISA-RP75.21, Process Data Presentation). These data will make possible the comparisons between the service conditions and valve capabilities.

B.10.2 The procedures contained in this document are intended to provide the best, practical knowledge currently available on the subject. However, practitioners should always avail themselves of proven new technology as it becomes available.

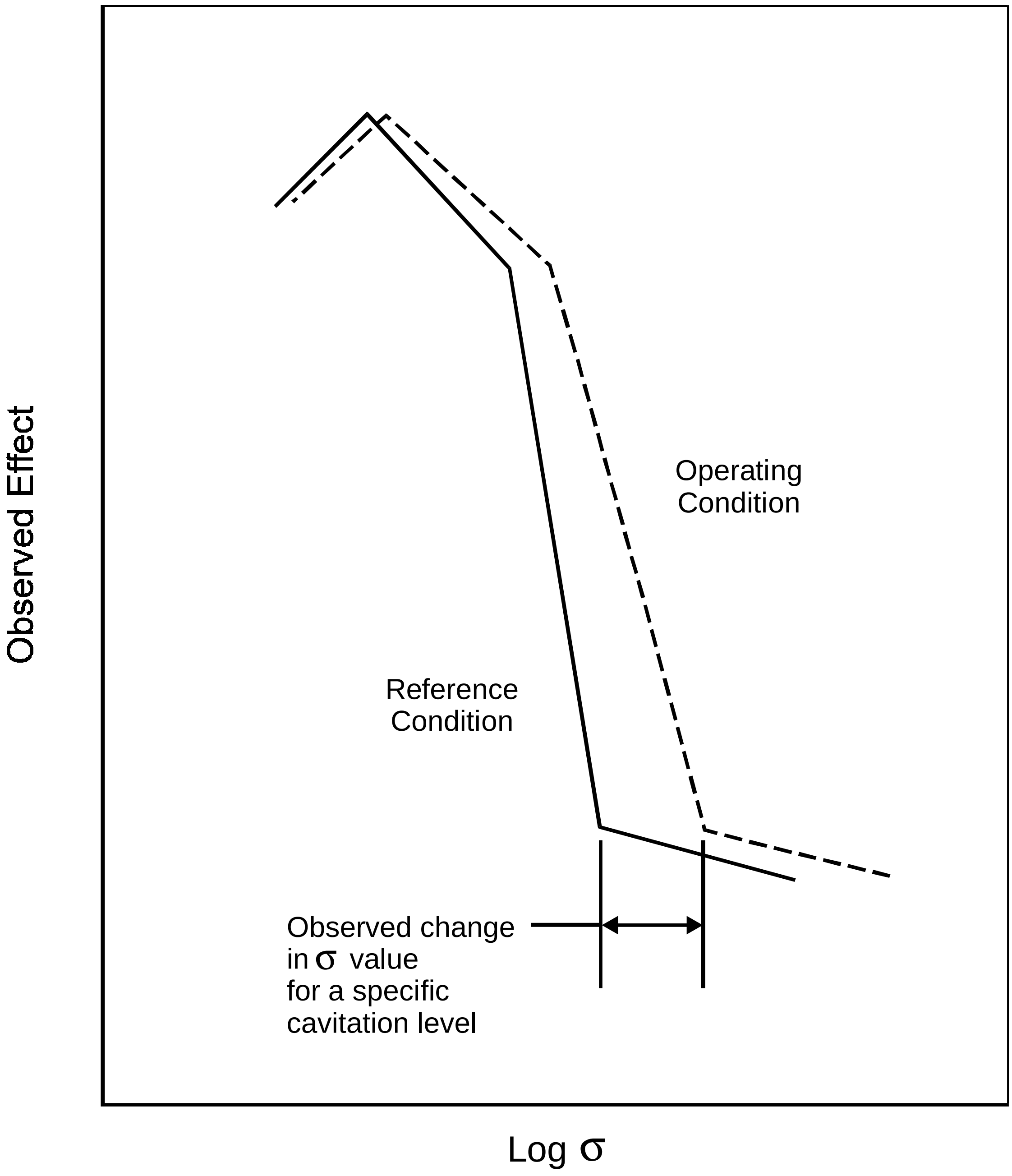

Figure B.1 — Cavitation scale effects

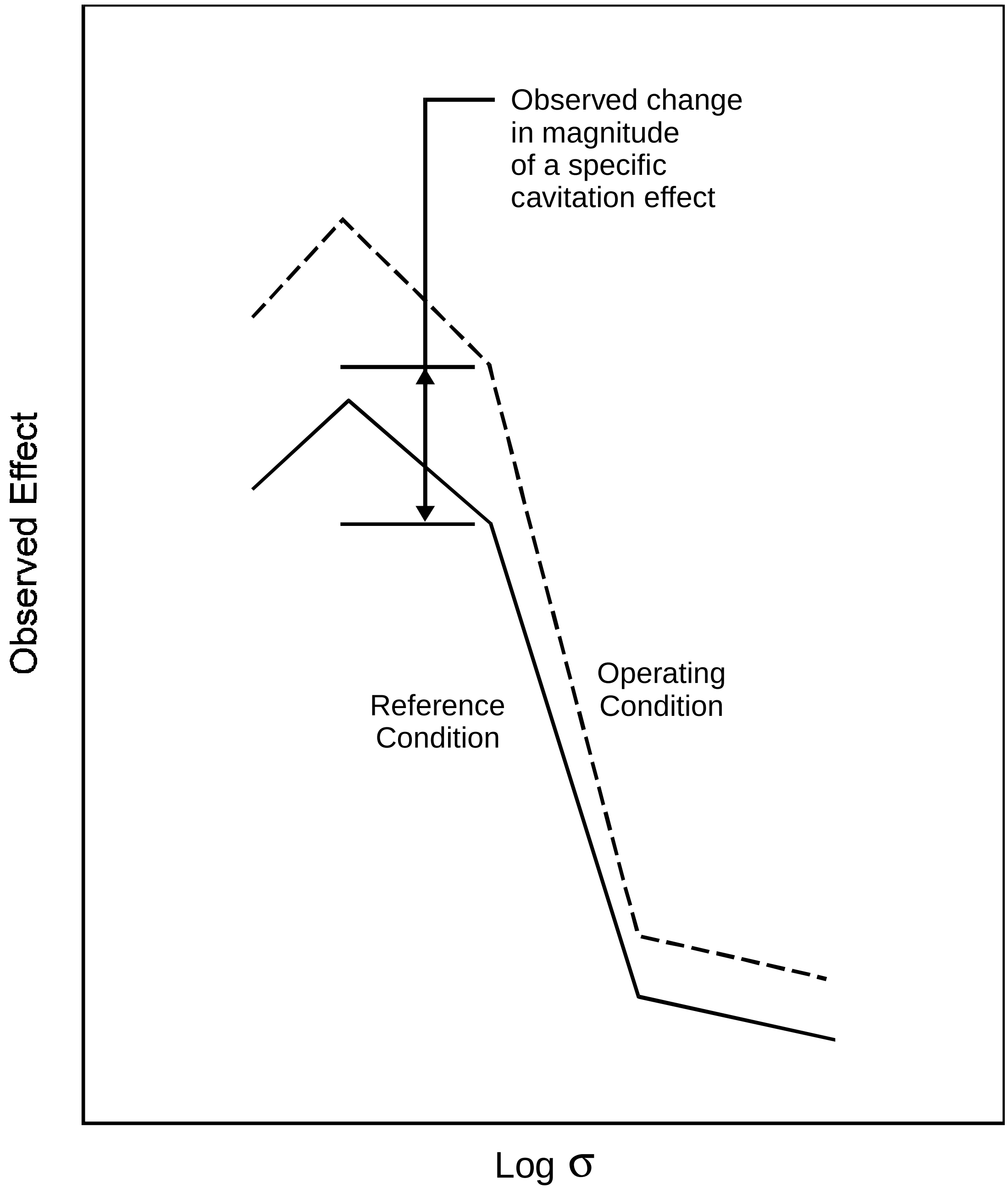

Figure B.2 — Cavitation influences

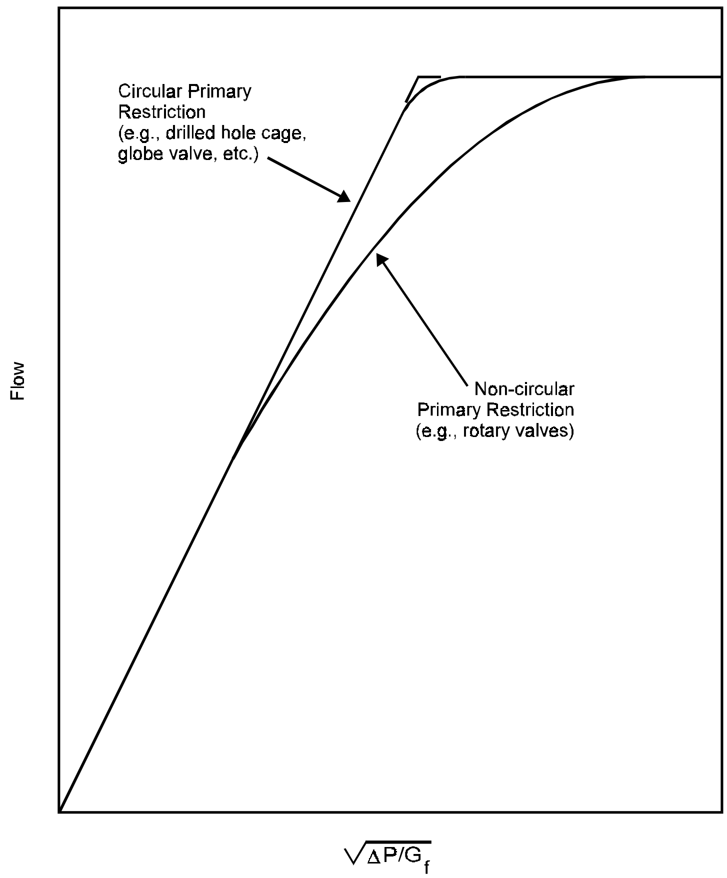

Figure B.3 — Typical flow curve appearance

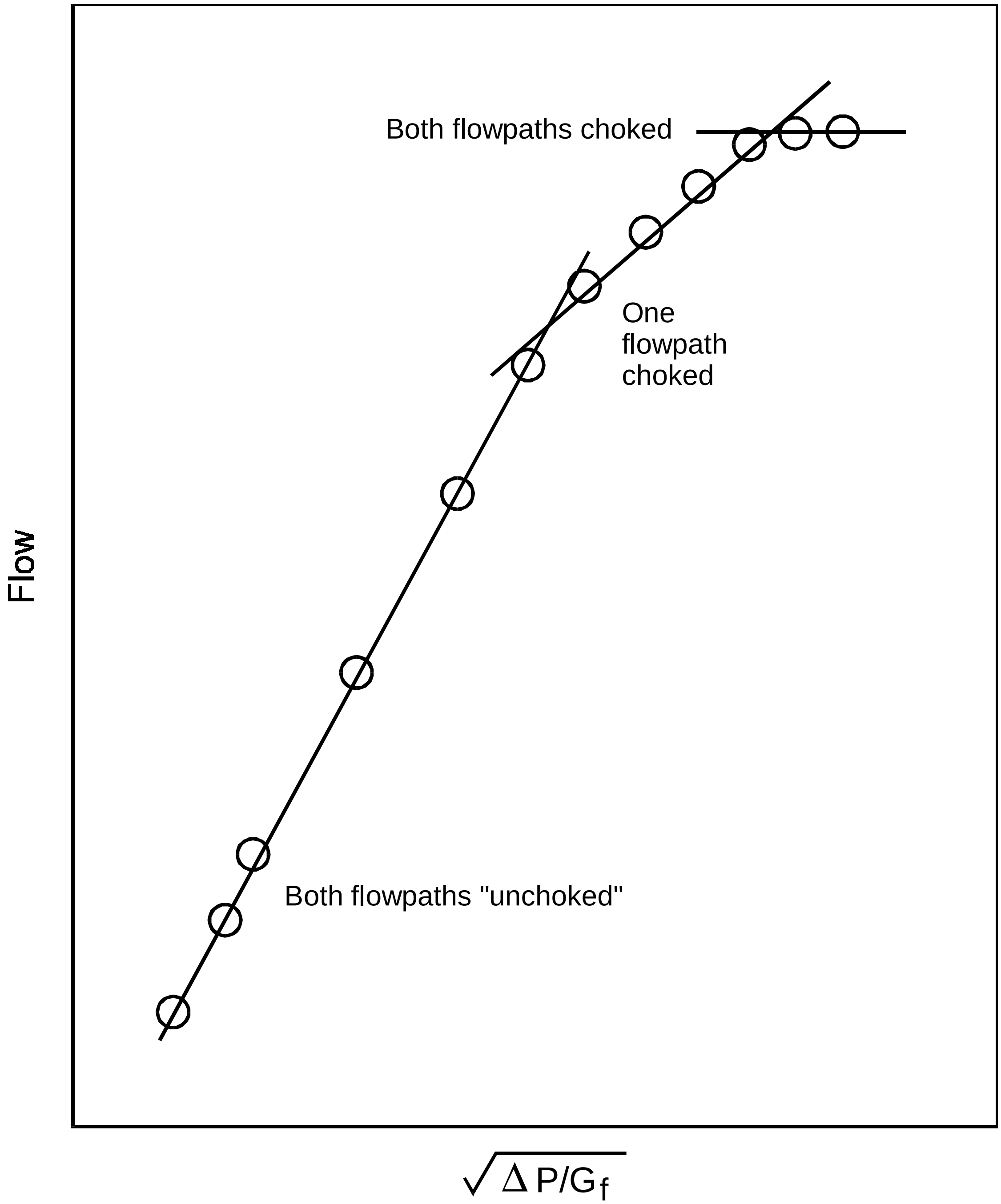

Figure B.4 — Flow curve appearance: two flowpath butterfly valve

Annex C — Cavitation damage intensity and service life*

C.1 Purpose

The safety of personnel, environment, and equipment must be ensured by proper monitoring, inspection, and maintenance programs in addition to proper valve selection. Within safety constraints, the desire to eliminate all cavitation damage sometimes meets with other limitations such as physical space, process variables, infrequent use, and cost. The following method is proposed in order to quantify cavitation damage intensity or relative service life reduction of control valves. In the past, this has been a judgment based on "guess work" and personal experience. Although based on theory and experience, the method and its results have not been substantiated by as broad a database as, for example, the scaling equations for σ. The presentation of the method acknowledges the need to better quantify the severity of cavitating service and the need to promote research, testing, and exchange of experience data.

C.2 Intensity index

The intensity index "I" (Equation C.1) can be considered as a damage intensity or life reduction factor. The magnitude represents an approximation of how many times faster erosion will occur over the threshold damage rate for the given operating conditions and service environment.

> **NOTE —** The intensity index can be used only when a valve must operate in cavitation levels more severe than the incipient damage level, i.e., at levels that are not recommended for the valve because of potential damage to the valve. It cannot be used where σ_{id} for the valve is unknown. This method must be approached with caution with some types of valves, e.g., rotary butterfly valves, for which σ_{id} is difficult to determine.

Once a numerical measure of severity of the service is calculated using σ_{SS} (Equation C.2), valve styles and trim can be selected. If the recommended application guidelines for any style of valve are exceeded, damage and vibration can be expected to be approximately proportional to the intensity index, defined as Equation C.1.

$$I = F_U F_T F_{DC}\left(\frac{\sigma_{id} - 1}{\sigma_{SS} - 1}\right) \tag{C.1}$$

$$\sigma_{SS} = \left(\frac{\frac{\sigma}{SSE} - 1}{PSE}\right) + 1 \tag{C.2}$$

*The procedure outlined in Annex C was adapted from a work by Barnes and Cain (ref. 13) wherein the authors theorized a method for evaluating valve damage potential in the cases where damaging cavitation for short durations cannot be totally eliminated. The method proposes some relationships from experience and the general literature, but it has not been rigorously validated by testing. It is presented here for information and to encourage further investigation. The method is not presented to justify the operation of valves under damaging cavitation conditions, but rather to illustrate the consequences of doing so.

where F_U, F_T, and F_{DC} are intensity modifiers defined below. Calculate the application σ from Equation 2. Adjust σ by using Equation C.2 to calculate σ_{ss}. This "normalizes" σ to the reference data (i.e., test data or manufacturer's reference values for σ_c, σ_{id}, etc.) Compare the selected σ_R to σ_{ss}. If σ_R is less than σ_{ss}, then the valve will operate at an intensity level less than the level corresponding to the selected σ_R. The basis for interpreting values of I is that a value of 1 (one) indicates "normal" wear, noise, and vibration for a valve operating at conditions for incipient damage. Values between zero and one mean proportionally less wear, noise, and vibration. Values greater than one indicate greater than normal wear, damage, vibration and noise. The intensity index is not defined for values less than or equal to zero.

C.3 Intensity modifiers

The Intensity is a function of influences in addition to σ. These factors should be determined by appropriate damage testing, but the following approximations may be used for estimating purposes. The incipient damage reference sigma (σ_{id}) can be determined from model tests or tests on valves in service with means provided to inspect damage coupons. Reference values of σ_{id} for different valve styles are provided by valve manufacturers.

C.3.1 Velocity factor, F_U

The velocity factor, F_U, may be the most important factor, because it is a function of the difference between the pitting threshold velocity (U_o) and the actual velocity (U). It can be shown that once damage or pitting has commenced at the incipient damage sigma or velocity, the rate of pitting is a strong function of the difference U-U_o. The following velocity correction factor for intensity appears to be in general agreement with published data (Knapp, et al., ref. 1) on pitting rates versus velocity.

$$F_U = 1.0 \qquad \text{for} \qquad U - U_0 < 0$$

$$F_U = 0.18 + 0.82e^{N_4(U-U_0)} \qquad \text{for} \qquad U - U_0 \geq 0 \tag{C.3}$$

where:

U is the average fluid velocity through valve outlet area or other characteristic area,

U_o is the damage threshold velocity (average) through the valve's outlet or characteristic area, and

N_4 is a constant based on units used for U and U_o (see Table 1).

From Equation C.3, it can be observed that values of F_U approach the value 1.0 as fluid velocity approaches U_o. As U exceeds U_o, values of F_U increase exponentially, which reflects the generally observed trends of pitting rate and material removal rate in the cavitation zone with sigma values less than σ_{id}. Due to the sensitivity of F_U to the velocity difference (U-U_o), the value of U_o should be verified by testing. Care should be taken in such tests to allow sufficient incubation time for pitting to be observed in the test materials. The importance of testing each valve style for its cavitation coefficients (i.e., σ_c and σ_{id}) and U_o cannot be over-emphasized where intensity evaluations are concerned. Minor differences in design geometry can result in significant differences in valve cavitation parameters.

C.3.2 Fluid temperature factor, F_T

F_T is the fluid temperature factor. Research on water at low pressures shows that, at fluid temperatures roughly half-way between the freezing point and boiling point (based on upstream pressure), cavitation damage is approximately three times more severe; that is, the rate of material removal is three times faster than the removal rate near the freezing or boiling temperatures (Knapp, et al., ref. 1). Equation C.4 can be used as an approximate factor to take this into account. This temperature effect may vary at higher pressures and with liquids other than water (Hammitt, ref. 16). Effects of temperature on damage rate are not well known as pressure nears the critical pressure; F_T should be ignored (i.e., set to 1) for pressures approaching or exceeding critical pressure. This factor applies only to liquids of a single chemical species.

$$F_T = 3 - 2\left(\frac{|T - T_{\text{ave}}|}{T_B - T_{\text{ave}}}\right) \tag{C.4}$$

where:

T_B = boiling temperature at upstream pressure

T_F = freezing temperature at upstream pressure

T = fluid service temperature

$T_{ave} = (T_B + T_F)/2$

$|T - T_{ave}|$ = absolute value of $(T-T_{ave})$

C.3.3 Duty cycle factor, F_{DC}

The duty cycle of the control valve within the cavitating condition (i.e., continuous, intermittent, rare) can also be taken into account with the duty cycle factor F_{DC}. If the valve will experience the cavitating condition only during a rare upset, short operation in this condition may not jeopardize the overall performance of the valve, since the damage effects are time dependent. If the valve cavitates only during certain service conditions, such as start-up, a more intense cavitating condition might be withstood than in a continuously throttling valve. If the valve is continuously throttling, or is in critical service, or both, a more conservative factor must be taken. Table C.1 represents some possible values of F_{DC} that might be used for estimating purposes if service history data are lacking. Values for F_{DC} are highly application dependent, and estimated values may introduce large uncertainties in the resulting calculation of intensity.

Table C.1 — Range of estimated values of duty cycle factor, F_{DC}

Frequency of occurrence	F_{DC}
Rare upset	0.1 - 0.3
Start-up	0.5 - 0.8
Throttling	1.0 - 1.5
Continuous or critical duty	2.0 - 3.0

C.4 Example

Calculations of the intensity index were originally proposed (Barnes and Cain, Ref. 13) for a cavitation index based on P_2 (see Equation B.3). The conversion of equations from the P_2-based index to a P_1-based index involves adding one (1.0). For very small values of σ, this changes the apparent precision of the calculations and magnifies the effects of cumulative round-off errors. Therefore, determination of significant figures and round-off should be postponed until the final calculation of intensity index (I).

C.4.1 Boiler feedwater start-up application (US Customary units)

Service data (same valve and piping as example 7.6.3)	Fluid: Water T = 90°F P_1 = 1600 psia P_v = 0.70 psia P_2 = 150 psia G_f = 0.995 ΔP = 1450 psi q = 400 gpm
Results of valve sizing	$C_v = 10.5$
Preliminary valve selection	Same valve as in example 7.6.3 Body outlet velocity = 4.9 ft/s.
Calculate σ (Equation 1)	$\sigma = (1600 - 0.70)/1450 = 1.103$
Compare manufacturer's recommended operating σ for possible alternative trim styles	see table below

Trim Style	σ_{mr}	U_o	σ_{id}
Standard	2.5	30 ft/s	2.3
Trim A	1.2	33 ft/s	1.2
Trim B	1.02	--	--

Evaluate Trim A, a multi-hole trim. Use exponent *a* for an orifice, because multi-hole geometry determines the pressure scaling behavior.	Manufacturer's cavitation reference data $(P_1\text{-}P_v)_R$ = 100 psi; a = 0.20
Calculate PSE, (Equation 3) Assume SSE = 1 for small, multi-hole type trim.	$PSE = (1599/100)^{0.20} = 1.741$
Calculate σ_v (Equation 2)	$\sigma_v = [(1.20)(1.00)-1](1.741)+1 = 1.348$
Calculate effect of reducers. (Equations 8, 7)	$F_p = \{1+(0.278)(10.5)^2/[890(5.75)^4]\} = 1.00$ $\sigma_p = (1)^2\{1.348+(0.093+0.68)(10.5)^2/[890(5.75)^4]\}$ $\sigma_p = 1.348$
Evaluation	σ is not $\geq \sigma_P$ for Trim A. Evaluate damage potential per Annex C or select another trim type.
Use intensity index evaluation. Calculate σ_{ss} (Equation C.2)	$\sigma_{ss} = [(1.103)/(1)-1]/1.741+1 = 1.059$
Determine velocity factor, F_u (Equation C.3)	Since $U\text{-}U_o = 4.9 - 33 < 0$, then $F_u = 1$.

Calculate temperature factor, F_T (Equation C.4)	$F_T = 3\text{-}2[\lvert 90\text{-}318 \rvert/(605\text{-}318)]$ $F_T = 1.411$
Estimate duty cycle factor F_{DC} (Table C.1)	For start-up, assume $F_{DC} = 0.5$.
Determine intensity index (Equation C.1) Let $\sigma_{id} = \sigma_{mr} = 1.2$	$I = (1)(1.411)(0.5)[(1.2\text{-}1)/(1.059\text{-}1)]$ $I = 2.4$
Evaluation	Trim A will wear (erode) approximately 2½ times faster than at the incipient damage level. Other trim options (Trim B) should be checked, e.g., multi-stage trim.

CAUTION — RAPID TRIM EROSION MAY BE ACCOMPANIED BY BODY WALL EROSION.

NOTE — The results of intensity calculations are sensitive to the accuracy and precision of the terms. Comparisons of intensity indexes should consider the relative precision of the terms and their accuracy.

C.4.2 Boiler feedwater start-up application (SI units)

Service data (same valve and piping as example 7.6.6)	Fluid: Water T = 32.2 °C $P_v = 4.83$ kPa $P_1 = 11\,034$ kPa $P_2 = 1034$ kPa $\Delta P = 10\,000$ kPa $G_f = 0.995$ $q = 90.85\ m^3/h$
Results of valve sizing	$C_v = 10.5$
Preliminary valve selection	Same valve as in example 7.6.6 Body outlet velocity = 1.49 m/s.
Calculate σ (Equation 1)	$\sigma = (11034 - 4.83)/10000 = 1.103$
Compare manufacturer's recommended operating σ for possible alternative trim styles	See table below

Trim Style	σ_{mr}	U_o	σ_{id}
Standard	2.5	9.14 m/s	2.3
Trim A	1.2	10.06 m/s	1.2
Trim B	1.02	--	--

Evaluate Trim A, a multi-hole trim. Use exponent *a* for an orifice, because multi-hole geometry determines the pressure scaling behavior.	Manufacturer's cavitation reference data $(P_1\text{-}P_v)_R = 690$ kPa; $a = 0.20$

Calculate PSE, (Equation 3) Assume SSE = 1 for small, multi-hole type trim.	$PSE = (11029/690)^{0.20} = 1.741$
Calculate σ_v (Equation 2) Calculate effect of reducers. (Equations 8, 7)	$\sigma_v = [(1.20)(1.00)-1](1.741)+1 = 1.348$ $F_p = \{1+(0.278)(10.5)^2/[0.00214(146)^4]\} = 1.00$ $\sigma_p = (1)^2\{1.348+(0.093+0.68)(10.5)^2/[0.00214(146)^4]\}$ $\sigma_p = 1.348$
Evaluation	σ is not $\geq \sigma_P$ for Trim A. Evaluate damage potential per Annex C or select another trim type.
Use intensity index evaluation. Calculate σ_{ss} (Equation C.2)	$\sigma_{ss} = [(1.103)/(1)-1]/1.741+1 = 1.059$
Determine velocity factor, F_u (Equation C.3)	Since $U-U_o = 1.49 - 10.06 < 0$, then $F_u = 1$.
Calculate temperature factor, F_T (Equation C.4)	$F_T = 3-2[\lvert 32.2-158.9 \rvert/(318.3-158.9)]$ $F_T = 1.41$
Estimate duty cycle factor F_{DC} (Table C.1)	For start-up, assume $F_{DC} = 0.5$.
Determine intensity index (Equation C.1) Let $\sigma_{id} = \sigma_{mr} = 1.2$	$I = (1)(1.41)(0.5)[(1.2-1)/(1.059-1)]$ $I = 2.4$
Evaluation	Trim A will wear (erode) approximately 2½ times faster than at the incipient damage level. Other trim options (Trim B) should be checked, e.g., multi-stage trim.

CAUTION — RAPID TRIM EROSION MAY BE ACCOMPANIED BY BODY WALL EROSION.

NOTE — The results of intensity calculations are sensitive to the accuracy and precision of the terms. Comparisons of intensity indexes should consider the relative precision of the terms and their accuracy.

Annex D — Net pressure drop corrections

D.1 Nomenclature

$C_{v\ meas}$	Valve flow coefficient calculated from the measured pressure drop ΔP during testing per SP75.02.
$C_{v\ net}$	Valve flow coefficient calculated from the net pressure loss through the valve only.
d	Valve inlet inside diameter, inches (mm).
f	The Darcy-Weisbach pipe friction factor.
G_f	Specific gravity of the liquid at inlet flowing conditions.
ΔP_{meas}	Measured valve pressure drop which included pipe friction loss, psi (kPa).
ΔP_{net}	Net valve differential/pressure drop, psi (kPa).
σ_{meas}	Cavitation index calculated with the measured pressure drop.
σ_{net}	Cavitation index calculated with the net pressure drop.

D.2 Measured and net pressure drop

D.2.1 The specific pressure drop utilized in testing valves versus that specified when applying them in a cavitating service is very important. Ordinarily, the pressure drop used in valve flow tests is the pressure difference between upstream and downstream pressure taps within the test section. However, when sizing and selecting a control valve, the upstream and downstream pressures are often specified at the upstream face and downstream face of the valve, respectively. Failure to account for this can lead to unexpected results.

D.2.2 The differences as described above give rise to the concept of net pressure difference versus total (or measured) pressure difference. The net pressure drop is the net effect on system pressure of installing a control valve into a piping system. The measured pressure drop is the total of the net drop and the additional loss associated with the length of pipe between the pressure taps. The flow coefficient, C_v, and the cavitation index, σ, calculated from the measured pressure drop can be significantly different from the values of C_v and σ calculated from the net pressure drop ($C_{v\ net}$ and σ_{net}).

D.2.3 The difference in calculated values of C_v, valve opening, and σ, is generally within the accuracy of the method itself for low recovery valves, and is therefore negligible. Low recovery valves typically are considered to be valves where $C_v/N_1d^2 < 20$, as in the case of many globe valves.

D.2.4 For higher recovery valves (i.e., when $C_v/N_1d^2 > 20$), the differences may be significant, and it becomes necessary to ensure consistency between test methods and application practices. Differences between measured and net pressure drop can be as large as 50% for low loss or high pressure recovery valves (Rahmeyer and Driskell, ref. 18). The difference between measured and net values can be even more important for cavitation calculations, because the cavitation index can vary considerably with small changes in valve opening. Small differences in calculating the C_v and pressure drop can produce significant differences in valve opening, actuator requirements, and valve cavitation coefficients.

D.2.5 If the net measured pressure drop is to be determined directly from testing, the downstream pressure taps can be located 10 diameters downstream (instead of 6 diameters) to ensure more complete pressure recovery. The downstream pressure of a control valve with a C_v/N_1d^2 greater than 20 may not be fully recovered at the distance of the 6 diameters as specified by ISA-S75.02.

D.2.6 As implied in the above discussion, the effect of adjacent piping should be considered when applying high recovery valves in cavitating service. The following equations describe methods to convert the various coefficients between total (measured) and net values. This technology is still evolving and is subject to certain restrictions. The methods described herein are limited to straight pipe installations. As of this writing, no inference is made as to the nature of the net effect of a valve on a system other than when straight pipe is immediately adjacent to the valve both upstream and downstream of the valve.

D.3 Equations

For control valves with a C_v/N_1d^2 greater than 20, the following equations can be used to convert pressure drops, C_v, and σ from measured values to net values. Equations D.1 and D.2 were originally published by Rahmeyer and Driskell (ref. 18) to convert values of C_v. Equations D.3 through D.6 were then derived from the original equations.

$$\frac{C_{v\,\mathrm{net}}}{N_1 d^2} = \left[\left(\frac{C_{v\,\mathrm{meas}}}{N_1 d^2}\right)^{-2} - 0.008986 f G_f\right]^{-\frac{1}{2}} \quad \text{(D.1)}$$

$$\frac{C_{v\,\mathrm{meas}}}{N_1 d^2} = \left[\left(\frac{C_{v\,\mathrm{net}}}{N_1 d^2}\right)^{-2} + 0.008986 f G_f\right]^{-\frac{1}{2}} \quad \text{(D.2)}$$

$$\Delta P_{net} = \Delta P_{meas}\left[1 - 0.008986\ f G_f \left(\frac{C_{v\,\mathrm{meas}}}{N_1 d^2}\right)^2\right] \quad \text{(D.3)}$$

$$\Delta P_{meas} = \Delta P_{net}\left[1 + 0.008986\ f G_f \left(\frac{C_{v\,\mathrm{net}}}{N_1 d^2}\right)^2\right] \quad \text{(D.4)}$$

$$\sigma_{net} = \frac{\sigma_{meas}}{\left[1 - 0.008986 f G_f \left(\frac{C_{v\,\mathrm{meas}}}{N_1 d^2}\right)^2\right]} \quad \text{(D.5)}$$

$$\sigma_{meas} = \frac{\sigma_{net}}{\left[1 + 0.008986\ f G_f \left(\frac{C_{v\,\mathrm{net}}}{N_1 d^2}\right)^2\right]} \quad \text{(D.6)}$$

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