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Cavitation Guide for Control Valves

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Prepared by
J. Paul Tullis

Tullis Engineering Consultants

Prepared for
U.S. Nuclear Regulatory Commission

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Cavitation Guide for Control Valves

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ABSTRACT

This guide teaches the basic fundamentals of cavitation to provide the reader with an understanding of what causes cavitation, when it occurs, and the potential problems cavitation can cause to a valve and piping system. The document provides guidelines for understanding how to reduce the cavitation and/or select control valves for a cavitating system. The guide provides a method for predicting the cavitation intensity of control valves, and how the effect of cavitation on a system will vary with valve type, valve function, valve size, operating pressure, duration of operation and details of the piping installation. The guide defines six cavitation limits identifying cavitation intensities ranging from inception to the maximum intensity possible. The intensity of the cavitation at each limit is described, including a brief discussion of how each level of cavitation influences the valve and system. Examples are included to demonstrate how to apply the method, including making both size and pressure scale effects corrections. Methods of controlling cavitation are discussed providing information on various techniques which can be used to design a new system or modify an existing one so it can operate at a desired level of cavitation.

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NOMENCLATURE

A	=	Area of the pipe	
A_j	=	Area of the contracted jet	
A_o	=	Area of the orifice hole	
C_c	=	Ratio of contracted jet area to the pipe area	
C_d	=	Discharge coefficient	(equ. 3.1)
C_{df}	=	Free discharge coefficient	(equ. 3.5)
C_v	=	Flow coefficient	(equ. 3.3)
$C_{v(ISA)}$	=	Flow coef. based on measured ΔP (not net ΔP)	(equ. 3.13)
d	=	Diameter of valve or pipe	
d_o	=	Orifice diameter	
D	=	SSE valve diameter adjustment	
ΔH	=	Head loss	
ΔH_{gross}	=	Measured head loss (uncorrected for friction)	
ΔH_{net}	=	Net head loss	(equ. 3.12)
ΔP	=	Pressure drop	
ΔP_{ch}	=	Choking pressure drop	(equ. 4.5)
ΔP_{gross}	=	Gross pressure drop	
ΔP_{kc}	=	Net pressure drop at incipient choking	
ΔP_{net}	=	Net pressure drop	
ΔP_{sys}	=	System pressure drop	
f	=	Darcy Weisbach friction factor	
F_f	=	Liquid critical pressure ratio factor	
F_L	=	Liquid pressure recovery factor	(equ. 4.9)
g	=	Acceleration of gravity	
H_p	=	Rated total dynamic pump head	
H_u	=	Total head at valve inlet	
K	=	Valve loss coefficient	(equ. 3.2)
k_c	=	Cavitation index	(equ. 2.7)
L	=	Distance between the pressure taps	
$NPSH_a$	=	Available net positive suction head for pumps	
$NPSH_r$	=	Required net positive suction head for pumps	
P_1	=	Absolute upstream pressure	
P_2	=	Absolute downstream pressure	
P_b	=	Barometric pressure	
P_d	=	Downstream pressure	
P_j	=	Absolute pressure at the vena contracta	(Fig. 4.3)
P_u	=	Upstream pressure	
P_v	=	Absolute vapor pressure	
$P_{v \text{ ref}}$	=	Reference absolute vapor pressure	

PSE	=	Pressure scale effect adjustments	(equ. 5.2 - 5.5)
Q	=	Flow rate	
Q _{th}	=	Choked flow	
Re	=	Reynolds number	(equ. 5.1)
SG	=	Specific gravity	
SSE	=	Size scale effect adjustments	(equ. 5.6 - 5.8)
V	=	Average velocity at the inlet to the valve	
V _{id}	=	Velocity at incipient damage	
V _u	=	Average velocity in upstream pipe	
V _d	=	Average velocity in downstream pipe	
X	=	Experimental PSE exponent	
Y	=	SSE exponent	

GREEK SYMBOLS

β	=	d_o/d = Area ratio for an orifice
γ	=	Fluid specific gravity
ρ	=	Fluid density
σ	=	Cavitation parameter
ν	=	Kinematic viscosity

SUBSCRIPTS FOR σ (Equation 2.1)

1	=	Sigma calculation based on ΔP_{net} and P_d	(equ. 2.2)
2	=	Sigma calculation base on velocity head	(equ. 2.4)
i	=	Incipient cavitation	
c	=	Critical cavitation	
id	=	Incipient damage cavitation	
ich	=	Incipient choking	
ch	=	Choked flow	
max	=	maximum noise and vibration	
super	=	Super cavitation	
ref	=	Reference cavitation limit	
m r	=	Manufacturer recommended	
sys	=	Cavitation index of the system	

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INTRODUCTION

This guide provides a method for predicting the cavitation intensity of control valves. It was prepared specifically for the nuclear power industry but the material applies equally well to any hydraulic system. The method can be used both as an analysis tool to determine the cavitation potential of existing valves and as a method for selecting and sizing valves for replacement or the design of new valve installations. It teaches the basic fundamentals of cavitation to provide the reader with an understanding of what causes cavitation, what cavitation can do to a valve and piping system and gives guidelines for understanding how to reduce the cavitation and its negative effect.

The first step in applying the method requires determining an appropriate level of cavitation for a given valve installation. Cavitation can result in undesirable noise, vibration, erosion damage and a decrease in valve performance. The guide defines six limits identifying cavitation intensities ranging from inception to the maximum intensity possible, including fully choked flow. The method used to experimentally determine each limit is described and suggestions given as to when each limit might be appropriate for design. Each of these effects should be considered and an appropriate operational cavitation limit must be selected when designing a valve in a system. The next step in the design process is to determine the cavitation potential for the valve by analyzing its performance over the full range of expected system operating conditions. The cavitation potential for the valve is then compared with experimental data for the six cavitation limits defined in the guide. If the valve is operating at an intensity greater than the selected operating limit, the valve is not suited to the application and system modifications may be needed or a different valve selected.

Comparing the valves performance to reference cavitation data frequently requires scale effects adjustments. Laboratory data for valves clearly show that at some levels of cavitation, there are scale effects associated with changes in system pressure and with the size of the valve. The guide clearly defines the scale effects and identifies for which cavitation limits scale effects adjustments are required. Equations and a procedure are introduced for making these corrections. Examples are included to demonstrate how to apply the design/analysis method, including how to make scale effects corrections.

The final section of the guide discusses methods of controlling cavitation and the principles used to design valves that are less susceptible to cavitation. It provides guidance for reducing the cavitation intensity for an existing valve, options for selecting replacement valves or selecting valves for new valve installations. Options include selecting a different type of valve, using multiple valves in series, using valves in parallel pipelines, using multiple orifices in series with the valve, using a slightly larger valve, plating parts of the valve and piping with cavitation resistant materials, using free discharge valves, using sudden enlargements and injecting air to suppress the cavitation.

1.0 FUNDAMENTALS OF CAVITATION

1.1 GENERAL DESCRIPTION OF CAVITATION

Cavitation is the process of rapid vaporization and condensation of a liquid. When vaporization occurs at constant pressure, due to a temperature increase, the process is referred to as boiling. When the process occurs at constant temperature, due to a pressure reduction, it is called cavitation. The vaporization process requires a liquid/gas interface generally provided by small gas bubbles (or nuclei) trapped on the boundary or on particles suspended in the liquid.

In the boiling process, as the water temperature increases and the pressure remains constant, vapor cavities eventually form at the interface of the nuclei trapped on the boundary of the heated surface. As these vapor cavities grow in size, they eventually detach from the heated boundary and rise to the free surface. As they rise, they expand due to decreasing pressure and increased vaporization. At the surface, the vapor cavities explode, releasing the vapor into the atmosphere. There is no violent cavity collapse or damage associated with the boiling process.

Cavitation will occur when a liquid at constant temperature is subjected to vapor pressure. In a flowing system, the pressure reduction is caused by a combination of a reduction in the average pressure due to local velocity increases and by local pressure reductions inside turbulent eddies created in the shear layers associated with flow separation and boundary layers. When the local pressure inside these eddies drops to vapor pressure, and if a nuclei is trapped inside the eddie, a vapor cavity can form. For valves, cavitation always begins before the average pressure drops to vapor pressure. If the local pressure inside the eddie stays at vapor pressure long enough, the cavity can reach a critical diameter and will grow rapidly. Since the ambient, or average, pressure surrounding the cavity is above vapor pressure, the bubble becomes unstable and collapses. The collapse is violent and is usually accompanied by noise, vibrations, potential erosion damage to solid surfaces and in advanced stages, a reduction in capacity or efficiency.

1.1.1 Types of Cavitation The literature describes two types of cavitation, one is referred to as gaseous cavitation and the other as vaporous cavitation. Gaseous cavitation occurs when a significant amount of free air is present in the liquid. The air can either be ingested into the system from some external source, if the pressure in the pipe is negative, or it can come out of solution if the fluid is subjected to fluctuating pressures below the liquid saturation pressure. When free air is present, the rate of growth of the cavity is slower and the collapse process is not as violent and generally does not cause erosion damage. Consequently, this type of cavitation is usually of little concern since it does not influence the safe operation of control valves. Sometimes air is injected into a system intentionally to reduce the cavitation. In systems where free air can be tolerated, it is one of the most effective methods of controlling cavitation provided it is

injected into the proper region. Aeration is generally only practical when the pressure in the cavitating region is negative so the air can be forced in with barometric pressure.

If there is little or no free air in the liquid, the cavities will consist almost exclusively of vapor making the growth and collapse rates, as well as the pressures generated upon cavity collapse, extremely high. This can cause erosion damage if the collapse occurs close to the boundary. This is called vaporous cavitation and is the type of cavitation responsible for damage to valves, pumps, turbines and other devices. This type of cavitation is of concern to the design engineer and to system operators.

There are three fundamental requirements for vaporous cavitation to occur: 1) there must be nuclei in the system which serve as a basis for vaporization of the liquid, 2) the pressure somewhere in the liquid must drop, at least momentarily, to or below vapor pressure, 3) the ambient pressure around the vapor cavity must be greater than vapor pressure in order for it to collapse. All subsequent reference to cavitation in this guide refers to vaporous cavitation (unless stated otherwise).

1.1.2 Effects of Cavitation There are a few beneficial applications for cavitation. The high degree of turbulence associated with cavitation can be used to increase mixing, accelerate chemical reactions, and is the basis for ultrasonic cleaning devices. In most hydraulic systems, however, the effects of cavitation are detrimental. There are at least six basic problems created by cavitation: noise, vibrations, pressure fluctuations, erosion damage, accelerated corrosion and loss of efficiency or flow capacity.

In its earliest stages, cavitation is usually heard as light intermittent crackling sounds similar to bacon frying or popcorn popping. These sounds are just slightly louder than the basic turbulence noise generated by the flowing liquid. As the cavitation level increases, the intensity and the frequency of the events increase so that it is easy to hear the cavitation above the operating level of the system. When the valve is operating at a moderate level of cavitation, the sound can sometimes be characterized as noise generated by gravel flowing through a pipeline. As the cavitation level approaches what would be called heavy cavitation, it consists of a continual loud roar and sometimes intermittent loud bursts similar to small explosions. For a valve operating in heavy cavitation, the audible sound level can exceed 100 db. This will result in hearing loss if an observer is exposed to that level for prolonged periods of time.

The intensity of the cavitation noise depends upon the type and size of the valve. For example, in a small valve, and for large valves that divide the flow into many small passages, the cavitation sound can be described as a hissing or a light crackling sound even at advanced stages of cavitation.

Even though cavitation is generated at the valve, the noise is radiated in both directions through the pipe so the sound is more like a line-source. If there is a conical expansion the noise may be louder downstream from the valve than it is right at the valve.

The shock waves generated by collapsing cavities produce pressure fluctuations that can cause the system to vibrate. As cavitation increases, the magnitude of the vibrations increase by several orders of magnitude. Even for large valves securely tied down, the pipe and valve can be visually seen to move when operating at heavy levels of cavitation. Such vibrations can loosen bolts, cause fatigue connections, loosen or break tie downs and lead to structural failure.

If the cavities collapse close to a solid boundary, erosion damage can occur. This is perhaps the most serious cavitation problem and is responsible for repairs to or replacement of many valves, pipes, pumps, turbines, etc.

At advanced stages of cavitation, large vapor cavities form which can change the hydrodynamics of the flow through the system and reduce the efficiency of the device. Pumps produce less head, turbines generate less power, and valves no longer pass the predicted flow if they operate in heavy cavitation. For example, the required net positive suction head for a pump represents the suction pressure just above the pressure at which the efficiency of the pump begins to decrease because of cavitation. For a valve, a similar loss of efficiency occurs at onset of choking cavitation. This corresponds to the condition where the average pressure downstream from the valve drops to vapor pressure. The efficiency loss is expressed by a decrease in the flow and discharge coefficients because of the large amount of vapor present downstream from the valve.

1.2 ROLE OF NUCLEI

The term nuclei is merely another word for a gas bubble or void in the liquid. In order for either boiling or cavitation to occur, there must be an interface for vaporization. The nuclei or void provides such an interface. If a liquid were completely de-aerated and the container cleaned so that there were no contaminants, voids, or entrapped air either in the water or on the boundary, the liquid could sustain tension. This means that the liquid would not boil at normal temperatures or cavitate until the pressure dropped far below the normal liquid vapor pressure. Therefore, the existence of nuclei is one of the primary requirements for cavitation to occur. The primary sources of nuclei are from free air bubbles, air bubbles trapped in crevasses of suspended material, or on the boundary.

Industrial liquids contain enough nuclei and contamination that cavitation generally occurs when the local pressure drops to vapor pressure. Only under controlled laboratory conditions can the nuclei content be reduced to the point where the critical pressure for cavitation is significantly less than vapor pressure. Therefore, attempting to control cavitation by reducing the nuclei content is not practical. However, controlling cavitation by artificially increasing the air content is a valid method of control. Air can be artificially introduced into the water, or if the water is recirculated in a water tunnel or in a closed loop pumping system, air can come out of solution and reduce the cavitation. Experience has shown that aeration is one of the most effective methods of controlling cavitation. However, not all piping systems can operate

satisfactorily with free air. Before considering using air to suppress cavitation it is necessary to determine if the piping system can tolerate it.

1.3 ROLE OF AMBIENT PRESSURE

The most prevailing misconception about cavitation is the frequently quoted statement that "cavitation begins when the average pressure downstream from the valve drops to vapor pressure." This concept is wrong and has caused much confusion about the cavitation process. It has been the cause of some valves being misapplied. Cavitation always begins when the downstream average pressure is higher than vapor pressure. By the time the average pressure reaches vapor pressure, the valve is choking and the cavitation intensity is near its maximum. Only under exceptional conditions is a valve and piping system able to operate at choked cavitation without operational problems.

Cavitation begins when the minimum value of the local fluctuating pressure (not the average pressure) drops to vapor pressure. The local pressure is the average pressure, which is uniform over a certain region of the flow, minus the fluctuating pressure, which depends upon fluid momentum and especially on the formation and dissipation of eddies or vortices in turbulent shear zones. Both of these play an important role in the cavitation process. The eddies are generated in zones of separation and in boundary layers. The local pressures at the center of these eddies can be significantly below the average pressure of the system. When the pressure inside the eddies drops to vapor pressure, cavitation begins.

There is some uncertainty from past literature about the actual pressure in the eddie when cavitation begins. Since it is virtually impossible to measure the pressure inside a traveling eddie the controversy may never be resolved. For this guide it is assumed that the critical pressure is vapor pressure.

The average pressure varies as the liquid passes through a valve, or any flow restrictive device, primarily due to local accelerations caused by a change in the cross sectional area of flow. The role of average pressure can be demonstrated by describing the flow through a venturi. Experimental studies have documented that cavitation in a venturi first occurs inside the eddies near the center of the boundary layer when the average pressure at the throat is slightly above vapor pressure.

In a well designed venturi there is no flow separation and the only turbulence is that generated in the boundary layer. Because the pressure variations in these boundary layer eddies is significantly smaller than that caused by flow separation (such as occurs in valves and orifices), the influence of the turbulent eddies on cavitation is less in a venturi. Consequently, the average pressure at the venturi throat must be lowered closer to the liquid vapor pressure (compared to valves and orifices) before cavitation begins. For venturies, the reduction of the average pressure caused by the high throat velocity is the predominant factor influencing onset of cavitation. For valves and orifices the pressure reduction inside the eddies is much greater and cavitation can occur when the average pressure is significantly above vapor pressure.

When designing or analyzing a piping system, consideration should be given to the location of valves, pumps, or turbines, and to the effect of average pressure on cavitation. Since the potential for cavitation reduces as the average pressure increases, it is desirable to place any device, which might be subject to cavitation, at a location of higher pressure. This can be done, for example, by placing a valve or pump at the lowest point of the pipeline or lowering a turbine relative to the downstream water surface elevation.

If an adequate pressure is not possible by adjusting the elevation of the installation, it may be desirable or necessary to increase the pressure by some other control device. For example, if a valve is required to produce a large pressure drop and there is concern about cavitation, it may be desirable to consider placing an orifice or second valve downstream to create a moderate pressure drop and increase the pressure at the main control valve. For a pump, it may be desirable to increase the size of the suction piping or change the pipe configuration to reduce intake losses.

1.4 ROLE OF TURBULENCE

Time dependent pressures generated by the formation and decay of turbulent eddies is an important part of the cavitation process. These time dependent pressure fluctuations are also the primary reason why cavitation is so difficult to predict and calculations of cavitation depend almost exclusively on experimental data. Average pressures can be predicted by applying the energy equation and properly estimating losses, flow areas and contraction coefficients in the system, but the pressure inside the eddies can only be estimated from empirical data.

To demonstrate the role of eddies (or turbulence) in the cavitation process, the flow characteristics of the sudden enlargement shown in Figure 1.1 will be described. The flow in the approach pipe is at relatively low velocity and high pressure. As the flow approaches the orifice, the velocity increases and the pressure reduces. When the jet enters the downstream enlargement, an intense shear layer is created at the boundary between the jet and the surrounding separation region. This region is shown between the two sets of broken lines. The high velocity gradients in this shear layer create eddies depicted in Figure 1.1.

The average pressure in the separation region surrounding the jet is approximately the same as the pressure inside the jet. However, the pressure inside the eddies is significantly less because of its high rotational speed. If nuclei (air bubbles) are entrapped in some of these eddies and the pressure inside drops to vapor pressure, the nucleus will begin to grow by vaporization. The force resisting growth of the nucleus is surface tension. Growth is controlled primarily by vaporization at the bubble interface. If the pressure remains near vapor pressure long enough for the nucleus to reach a critical diameter (where vaporization dominates the surface tension), the bubble begins to grow rapidly by vaporization.

As the size of the vapor cavity increases, the strength of the eddy is rapidly destroyed, the rotational speed is reduced and it can no longer sustain vapor pressure in

the cavity. Since the ambient pressure surrounding the cavity is significantly above vapor pressure, it becomes unstable and collapses inward. The collapse can be very violent since there is usually little or no air trapped inside the cavity to provide a cushion.

Figure 1.1 Formation of Eddies and Cavitation at an Orifice

The length of time that a nucleus is subjected to the low pressure inside the eddie is important. If the time is so short that the bubble cannot reach its critical diameter, it will not become a cavitation event. The time of exposure of the nucleus to vapor pressure depends on the time of travel of the eddie through the separation zone. This travel time through the low pressure region time will increase with valve size. The flow rate and the number of nuclei will also increase. Therefore, the probability of cavitation should significantly increase as the size of the system increases. As a result, cavitation would be expected to be more severe in larger valves. This concept will be addressed again in the section entitled "Size Scale Effects."

To further understand the role of the eddies, compare the cavitation potential of a venturi described in the preceding section, to that of a submerged jet. Consider that both are installed in the same size pipe and that the throat diameter of the venturi is the same as the contracted jet diameter for the submerged jet. This means that the maximum velocity will be the same in both systems. If both are operating at the same upstream pressure and flow rate, the minimum average pressure of the contracted jet, at its smallest area, will be the same for both. However, the submerged jet will be much more susceptible to cavitation because of the higher intensity of the turbulence generated in the shear layer around the jet.

In some systems it may be desirable to provide streamlining to avoid separation and cavitation. A typical example would be the design of pump impellers and turbine blades. The concept of streamlining can also be used in the design of high recovery valves. Such valves are designed to minimize pressure loss by streamlining the flow passage. This allows the valve to operate somewhat like a venturi where most of the

energy contained in the high velocity at the throat is converted back into pressure energy in the diverging section of the valve.

However, if the primary function of the valve is to dissipate energy by creating a pressure drop, it is generally not practical or possible to use the concept of streamlining to avoid separation and suppress cavitation. By so doing, the energy dissipation or pressure reducing capability of the device is compromised. Therefore, if a valve is required to produce both a large pressure drop and avoid cavitation, the preferred approach is to create a well defined separation region rather than trying to eliminate it. By removing the boundary farther from the jet, it reduces the intensity of the shear stress and causes the cavitation to collapse remote from the pipe wall. Potential damage is reduced but noise and vibration may not be effected.

1.5 ROLE OF PRESSURE RECOVERY

The final phase of cavitation is collapse of the cavity. If the local pressure surrounding the cavity is higher than vapor pressure, the cavity eventually becomes unstable and can collapse violently. For systems like a venturi, cavitation begins near the throat section where the average pressure is only slightly above vapor pressure. As the cavities move away from the throat and into the diffuser, the average pressure increases due to reduction in the velocity. This pressure recovery increases the local pressure tending to collapse the cavities. There is also some local pressure recovery as the cavity trapped inside the tiny vortices in the boundary layer are dissipated and lose their rotational speed.

For an orifice, the pressure recovery occurs primarily as a result of the eddies being dissipated because of a reduction in their rotational speed and the vapor cavity being subject to the local pressure (which is well above vapor pressure). There is also some pressure recovery associated with diffusion of the jet beyond the "vena contracta", as shown by the rise in the pressure gradient in Figure 1.1 between points 2 and 3.

One method of controlling the damaging effects of cavitation is to prevent or control the pressure recovery. This concept is applicable to choked cavitation for certain valve designs and for orifices and venturies. When a valve chokes, the average pressure at the minimum flow cross section (vena contracta) is at vapor pressure. If the pressure can be maintained at vapor pressure for some distance downstream, the zone where the cavities collapse can be removed from the valve. This allows the valve to operate in heavy cavitation and not experience the damaging effects of the collapse. However, in the region where the collapse occurs, damage can be severe.

1.6 DAMAGE MECHANISM

If the vapor cavities are transported close to a solid boundary before they collapse, erosion damage can occur. Prior research has indicated that collapse must occur within approximately one bubble diameter from the boundary in order to cause erosion damage (Ref. 21). Since the bubbles are generally small, this means that only those cavities that collapse very near or on the surface will cause erosion damage.

Two mechanisms have been identified in the literature by which damage occurs at solid boundaries. One is the high pressure shock waves generated by collapse of the cavities. With vaporous cavitation there is essentially no free air trapped inside the cavity. As the cavity collapses, the vapor condenses and the cavity tends to go to zero volume. The pressure differential causing acceleration of the liquid is relatively high and the mass of fluid involved is very small. The result is a very large acceleration and high velocities of the cavity surface as the cavity disappears. The pressure generated as the cavity disappears is directly proportional to that velocity. The pressures generated by these shock waves have been estimated by some researchers to be on the order of 10^6 psi (Table 4.4 in Ref. 20) which is sufficient to damage even the most cavitation resistant materials. Erosion of the solid boundary can result from material fatigue and plastic deformation.

The other source of potential damage is caused by micro jets. When a bubble collapses near the boundary, the pressure distribution around the bubble is unsymmetrical due to the presence of the boundary. As the bubble collapses, the boundary causes a resistance to the flow trying to fill the void from the boundary side and allows the liquid on the side of the bubble away from the wall to attain a higher velocity. This causes the bubble to collapse faster on that side forming a jet which shoots through the center of the bubble. This jet attains high velocities and creates a local pit when it impacts the wall. This is schematically shown in Figure 1.2. Reference 32 contains a comprehensive discussion of the formation and collapse of the cavities and the proposed damage mechanisms.

Figure 1.2 Formation of a Micro Jet Near a Solid Boundary

The micro jet concept may also explain a case of cavitation damage observed by the author where the 1-inch thick wall of a valve was penetrated by numerous tiny holes. The wall looked like a honeycomb. Most of the wall material was still in place, but the wall was perforated by a labyrinth of small diameter with deep holes. A possible explanation is shown graphically in Figure 1.3. Small air bubbles trapped in these crevasses would respond to the shock waves generated by cavitation implosions near the boundary and create micro jets. Surface tension caused by the wall would retard the fluid attempting to compress the bubble. The liquid near the center will accelerate faster and create a micro jet as shown in the figure. Repeated blows of the jet would act like a jack hammer and continue eroding through the valve wall.

Figure 1.3 Micro Jet in Boundary Crevice

Once a system reaches a point at which cavitation erosion damage occurs, damage increases rapidly as the velocity (V) of the system is further increased. Past research has shown that the rate of cavitation erosion damage varies exponentially with velocity. This relationship can be expressed as:

$$\text{Damage} \propto (V - V_{id})^n \quad (1.1)$$

where V_{id} is the velocity at incipient damage and n varies, depending upon the experiment, between 4 and 8 (Ref. 19, 20, 21, 35, 36). This strong dependence of damage rate on velocity serves as a warning that one should be conservative in selecting conditions corresponding to the onset of erosion damage since only a small increase in the velocity causes such a large increase in the damage rate. For example, assume for a given valve that the exponent n in Equation 1.1 is 7. Operating that valve at a velocity 2.3 fps beyond the incipient damage velocity would increase the damage rate by 340 times.

1.7 CAVITATION DAMAGE

The typical appearance of a surface exposed to cavitation damage for an extended period of time can be characterized as having a "crater-like" surface or a "honeycombed" appearance. Under magnification the surface is seen to have deep pits with sharp edges. Some of the holes can extend to significant depths.

In early stages, the surface will either look etched (like frosted glass) or will contain well defined individual craters visible under low power magnification. If the surface is initially rough, the cavitation can actually give the surface a polished appearance.

The damage discussed so far can be called mechanical damage. Another aspect of cavitation erosion damage is the relationship between cavitation and chemical corrosion. This process has been referred to as cavitation corrosion. The combined effect of the two processes may be greater than the sum of each acting alone (Ref. 20). Even when the cavitation is not intensive enough to cause mechanical damage, it may be heavy enough to remove the oxidation film which normally retards the corrosion process. The removal of the oxidation layer by cavitation continuously exposes fresh material and significantly accelerates the corrosion process. The cavitation process can also release dissolved gasses that can accelerate the corrosion process. In some cases a significant percentage of material removed by cavitation may actually be due to corrosion.

Cavitation can also accelerate abrasion damage caused by high velocity jet impingement especially when there is sediment in the liquid. A surface weakened by cavitation is more easily eroded by the abrasive action of the jet or sediment. Damage caused by a cavitating high velocity jet impinging on a surface (containing sediment) produces a damaged surface that does not always look like classical cavitation damage. The jet and sediment can abrade the surface removing the sharp edged craters formed by the cavitation and produce a relatively smooth surface.

2.0 QUANTIFYING CAVITATION POTENTIAL

Cavitation is frequently an important consideration in the selection and operation of valves and piping systems. The intensity of cavitation and its effect on the system varies with valve type, size, operating pressure and details of the piping installation. To include cavitation in the design process, it is necessary to determine if cavitation will exist, evaluate its intensity, characterize the flow conditions and estimate its effect on the system and environment. Possible effects include noise, vibration, erosion damage and a decrease in performance.

Determining the existence and intensity of cavitation depends upon the availability and proper application of experimental data. It is helpful to establish acceptable cavitation limits that identify the flow conditions for cavitation levels varying from no cavitation, to fully choked flow. This assumes that the laboratory has adequate capacity and can safely test the valve at choked cavitation. Selection of the appropriate cavitation level for a given valve in a particular system is a critical part of the analysis. This section discusses the parameters used to quantify the cavitation intensity and the valve's hydraulic performance. Subsequent sections describe how the value of the cavitation parameter is experimentally evaluated at the different levels of intensity. Cavitation limits to be used for design and analysis are discussed in Section 4.

To characterize the flow conditions corresponding to a selected level of cavitation requires defining a cavitation index derived from dimensional analysis. The process involves determining the fluid properties, geometric characteristics and fluid parameters that influence the cavitation process. The main variables influencing the cavitation process are: 1) the geometric configuration of the device, 2) absolute pressure in the cavitating region, 3) the critical pressure, generally assumed to be the liquid vapor pressure and 4) velocity or pressure drop.

A dimensionless cavitation parameter is needed to quantify the intensity of cavitation at different flow conditions. This parameter is usually defined as the ratio of force trying to suppress cavitation to the force trying to cause it. For a valve, the force suppressing cavitation is proportional to the magnitude of the average pressure relative to absolute vapor pressure i.e. $(P - P_v)$. The force causing cavitation is proportional to the velocity head $\rho V^2/2$ or to the pressure drop across the valve ΔP . The resulting cavitation parameter sigma can be expressed as:

$$\sigma = \frac{(P_1 - P_v)}{\Delta P_{\text{net}}} \quad (\text{use absolute units}) \quad (2.1)$$

in which P_1 is the absolute pressure just upstream from the valve ($P_1 = P_u + P_b$), P_u is the gauge pressure just upstream from the valve, P_b is the atmospheric or barometric pressure, P_v is the absolute vapor pressure and ΔP_{net} the net pressure drop across the valve. This is the definition of the cavitation parameter that will be used in this guide.

Section 3.2 contains a discussion about ΔP_{net} and the reason why it is used in the sigma equation and in the valve coefficient equations. The minimum value of sigma is 1.0 since the maximum ΔP_{net} across the valve is $P_1 - P_v$.

Numerous other forms of the cavitation parameter have been used. Several will be discussed and conversion equations provided so that cavitation data found in other references can be converted to the sigma Equation 2.1. The choice of reference pressures used in the sigma equation is arbitrary. P_1 is generally the most convenient choice and will be used in this guide.

The preferred form of the cavitation index varies with the type of device being considered. For example, for a pump, cavitation potential is normally expressed by the value of the required net positive suction head (NPSH_r) or in terms of a dimensionless cavitation parameter equal to NPSH_r/H_p . Where H_p is the rated total dynamic pump head. For valves and other devices that create a pressure drop, the cavitation parameter can be defined in several ways. The reference pressure in the numerator of Equation 2.1 can be either the upstream or the downstream pressure. Both have been used and there are valid reasons supporting either choice. Using the downstream pressure P_d , the equation becomes:

$$\sigma_1 = \frac{(P_2 - P_v)}{\Delta P_{\text{net}}} \quad (2.2)$$

in which P_2 is the absolute downstream pressure ($P_2 = P_d + P_b$), where $P_d = P_u - \Delta P_{\text{net}}$ and P_b is the barometric pressure. This is the form originally preferred by the author (Ref. 43) because the downstream pressure is the pressure closer to the zone where the cavitation actually occurs. Therefore, the downstream pressure more directly influences the cavitation. However, it is more convenient to use the upstream pressure for most applications and the two sigma values differ by a constant value of 1.0. The two equations are directly related by the following simple equation:

$$\sigma = \sigma_1 + 1.0 \quad (2.3)$$

For cavitation caused by surface roughness, an isolated roughness, an offset in the boundary or by any device for which it is not possible or convenient to evaluate a pressure differential, the velocity head can be used in place of ΔP in Equation 2.1. The resulting equation is:

$$\sigma_2 = \frac{2(P_1 - P_v)}{\rho V^2} \quad (2.4)$$

in which ρ is the fluid density and V the average velocity. Equations 2.1 and 2.4 are directly related by the valve loss coefficient K . The relationship is:

$$\sigma = \frac{\sigma_2}{K} \quad (2.5)$$

where

$$K = \frac{2 \Delta P}{\rho V^2} = \frac{2 g \Delta H}{V^2} \quad (2.6)$$

Some authors (Ref. 33) have chosen to use the reciprocal of Equation 2.1 and have defined the cavitation index as k_c instead of sigma (Ref. 33). The equation defining k_c is:

$$k_c = \frac{\Delta P_{kc}}{(P_1 - P_v)} \quad (2.7)$$

in which ΔP_{kc} is the net pressure drop at incipient choking. This equation is discussed again in Section 4.3.1. In the past k_c has been referred to as "incipient cavitation" which is inconsistent with the normal understanding of the term "incipient." Additional comments on this are in Section 4.3.1. It is sufficient to say at this point that k_c should not be used to predict early stages of cavitation.

The choice of definitions for the cavitation index is somewhat a matter of choice. For this guide, Equation 2.1 will be used in all subsequent discussions and examples. If the reader has data based on another definition such as σ_1 , σ_2 , k_c or any other form found in the literature, it is easy to convert them to Equation 2.1.

Subsequent sections will describe in detail the procedure for making a cavitation analysis for a valve. In brief, the procedure includes:

1. Determining the required K , C_d or C_v and sigma (σ_{sys}) for the system at a specific flow condition.
2. Selecting the appropriate operating limit for the valve (incipient, critical, etc.)
3. Locating experimental cavitation data for that limit to determine the allowable σ limit.
4. Making necessary size and pressure scale effects adjustments to the experimental σ values.
5. If $\sigma_{sys} \geq \sigma$ limit, the valve will operate within the required cavitation intensity.
6. Repeat the calculations for the full range of operating conditions and valve openings.

3.0 FLOW AND LOSS COEFFICIENTS

The cavitation potential of a valve generally varies with the valve opening. This relationship can be expressed by plotting the value of sigma versus valve opening for a chosen level of cavitation (several levels of cavitation are presented in Section 4). Such a relationship will be unique for a given valve. However, if one wishes to compare the cavitation performance of similar valves, it is better to make the comparison based on a flow coefficient rather than valve opening.

The relationship between flow and pressure drop at any valve opening can be expressed by any of a number of coefficients. Like the cavitation index, these coefficients are derived from a ratio of forces. In this case, it is the ratio of inertial forces to pressure forces. Numerous forms of these coefficients have been proposed. The coefficients used in this guide are:

Discharge Coefficient

$$C_d = \frac{V}{\sqrt{\frac{2\Delta P_{net}}{\rho} + V^2}} \quad (3.1a)$$

or

$$C_d = \frac{V}{\sqrt{2g \Delta H_{net} + V^2}} \quad (3.1b)$$

Loss Coefficient

$$K = \frac{2 \Delta P_{net}}{\rho V^2} \quad (3.2a)$$

or

$$K = \frac{2 g \Delta H_{net}}{V^2} \quad (3.2b)$$

Flow Coefficient

$$C_v = \frac{Q}{\sqrt{\frac{\Delta P_{net}}{SG}}} \quad (3.3)$$

$$\frac{C_v}{d^2} = \frac{Q}{d^2 \sqrt{\frac{\Delta P_{net}}{SG}}} \quad (3.4)$$

Free Discharge Coefficient

$$C_{df} = \frac{V}{\sqrt{(2gH_u + V^2)}} \quad (3.5)$$

in which V is the average velocity at the inlet to the valve, g the acceleration of gravity, ρ is the fluid density (specific weight/gravity), ΔP_{net} the net pressure drop across the valve, ΔH_{net} the net head loss across the valve, SG is specific gravity of the liquid and H_u is the pressure head at the inlet to the valve. In Equations 3.3 and 3.4, ΔP_{net} is in psi, Q is in US gallons per minute (gpm) and d the inside diameter of the valve (inches). In the other equations the coefficients are dimensionless and any system of units can be used.

Equation 3.5 is the form used for free discharge valves where there is no downstream pressure recovery. It also applies to a valve that is choking.

C_d is defined as the discharge coefficient. It is dimensionless and varies from zero when the valve is closed to a maximum of 1.0 for valves that have no pressure loss in the full open position. The variation of cavitation potential with valve position in this guide will be expressed using the variation of sigma with C_d . This allows a direct comparison of cavitation data for similar types of valves. It is recognized that many in the valve industry use C_v , and C_v/d^2 . Since neither of these coefficients are dimensionless C_d is preferred. Its use avoids confusion caused by choices of units such as whether to use the flow rate in US gpm or British gpm and the dimensions of d .

The dimensionless coefficient K is conventionally referred to as the valve loss coefficient and is used in the application of the energy equation to a piping system to calculate the head loss (or pressure drop) across a valve or any other local loss.

The flow coefficient C_v is not a dimensionless coefficient but is one that has gained wide acceptance because it is an easy method of quantifying the flow capacity of different valves. It physically represents the flow rate in US gpm that a valve will pass at a differential pressure of one psi. Since C_v has units and varies with valve diameter, C_v/d^2 is frequently used to make it into a pseudo dimensionless coefficient so meaningful comparisons can be made between similar valves of different sizes.

When comparing the flow coefficients of identical valves of different sizes, K , C_d and C_v/d^2 will be the same (ignoring any Reynolds number effects). C_v can be scaled from a valve of diameter d_2 , to one of diameter d_1 using:

$$C_{v1} = C_{v2} \left(\frac{d_1}{d_2} \right)^2 \quad (3.6)$$

One of the difficulties in scaling any of the coefficients to valves of different size is that valve manufacturers seldom maintain geometric similarity. Manufacturing and structural constraints generally require that smaller valves have relatively more blockage and hence a smaller discharge coefficient. Therefore, it is usually necessary to make some adjustment in the coefficients to account for size variation.

For full ported valves, like ball, cone and gate valves, where the net head drop for the valve fully open is almost zero, the value of C_v becomes very large and is difficult to experimentally evaluate. For such valves it is better to use C_d because it approaches a value of 1.0 for a low loss valve.

3.1 CONVERSION EQUATIONS

Since there are several coefficients in use, the following conversions are included to aid in transferring from one definition to another. Sigma conversion equations were presented previously in section 2.0.

$$K = \frac{1}{C_d^2} - 1 \quad (3.7)$$

$$C_d = \frac{1}{\sqrt{K + 1}} \quad (3.8)$$

$$C_d = \frac{1}{\sqrt{\frac{890 d^4}{C_v^2} + 1}} \quad d \text{ in inches} \quad (3.9)$$

$$C_v = \frac{\sqrt{890 d^4 C_d^2}}{\sqrt{1 - C_d^2}} \quad d \text{ in inches} \quad (3.10)$$

$$\frac{C_v}{d^2} = \frac{29.8}{\sqrt{K}} \quad (3.11)$$

For any who do not feel comfortable using the conversion equations, Figures 3.1-3.4 have been included. The conversions from C_v to any of the other coefficients is only valid for a specific valve diameter.

3.2 NET VERSUS GROSS PRESSURE DROP

In all of the preceding equations, ΔP_{net} and ΔH_{net} have been defined as the net loss. It is important to understand the difference between the gross and net loss and the influence of the choice on the magnitudes of the various coefficients and on the results of system calculations. Because there is not a universal agreement on whether to use the

gross or net loss, the author will explain why this guide uses the net pressure drop in the equations for the cavitation index and for the various flow and loss coefficients.

The pressure drop and flow required to evaluate the various coefficients are measured experimentally. The pressure taps are typically located one or two pipe diameters upstream and six or more diameters downstream (Ref. 2). Placing the downstream pressure tap several pipe diameters from the valve is necessary to insure that the measured downstream pressure is not effected by the non-uniform flow conditions near the valve. This means that the measured pressure drop includes the friction loss caused by at least seven or eight diameters of pipe. For valves that have a small loss in the full open position, the friction loss can be a significant percentage of the total measured loss. For high loss valves, the difference is small and may be neglected.

The net loss is the measured (or gross) loss minus the measured or calculated friction (or manifold) loss. Experimentally, this is done by first determining the manifold loss at different flow rates by testing without the valve installed. This loss can be expressed as a friction coefficient for the pipe. Then, with the valve installed, the manifold loss for the test pipe section is calculated at the measured flow and its value is subtracted from the measured pressure drop. The same pressure taps should be used for the friction or manifold loss measurements and the valve pressure drop measurements. The net loss, expressed in feet of fluid, can be calculated with the following equation:

$$\Delta H_{\text{net}} = \Delta H_{\text{gross}} - \frac{fLV^2}{2gd} \quad (3.12)$$

in which f is the Darcy-Weisbach friction factor and L the distance between the pressure taps. Subtracting the friction loss from the measurement and using the net loss is the only thing that really makes sense when applying the data to a system because the friction loss in the pipe is always included as a separate calculation when analyzing the system. Including friction loss as part of the valve loss calculation would mean adding it twice. As defined, the net loss represents the net effect of installing a valve in a pipeline. This is a simple concept but there is still disagreement in the valve industry ΔH_{net} versus ΔH_{gross} on whether to use the gross (Refs. 2, 16) or net pressure loss (Ref. 17) in capacity and sigma calculations.

Figure 3.1 Conversion Graph for C_d and K

Figure 3.2 Conversion Graph for C_v/d^2 and K

Figure 3.4 Conversion Graph for C_v and K

Figure 3.3 Conversion Graph for C_v and C_d

The magnitude of the effect of ΔH_{net} versus ΔH_{gross} on C_v for a low loss valve is best demonstrated by an example. Consider the following experimental data taken on a 24-inch butterfly valve fully open. The valve was installed in a straight section of 24-inch diameter pipe with piezometers located two diameters upstream and 6.0 diameters downstream. For one test point the measured data were: $\Delta P_{\text{gross}} = 0.355$ psi, $Q = 21,000$ gpm. The friction factor for the pipe had been previously measured as 0.0128. The net pressure drop was calculated to be 0.227 psi. The flow coefficient C_v for the ΔP_{gross} and ΔP_{net} values are:

$$C_v = \frac{21000}{(0.355)^{0.5}} = 35,200 \quad \text{(based on } \Delta P_{\text{gross}} \text{)}$$

$$C_v = \frac{21000}{(0.227)^{0.5}} = 44,100 \quad \text{(based on } \Delta P_{\text{net}} \text{)}$$

The difference is about 20 percent. For higher loss valves, such as most globe valves (especially those with special cavitation trim) the effect on C_v is much less. When obtaining flow coefficients and sigma values from manufacturers or laboratory studies, it is important to know if they are based on ΔP_{net} or ΔP_{gross} . Many valve manufacturers use the measured pressure drop as specified by the Instrument Society of America (Ref. 2) in the flow coefficient equations for valves rather than the ΔP_{net} . To convert from C_v based on the measured pressure loss, the following equation can be used (Ref. 26). The conversion equation assumes that the friction loss (manifold loss) is based on 8 pipe diameters and the equation requires a friction factor for the pipe. The equation is:

$$C_v^2 = \frac{C_{v(\text{ISA})}^2}{d^4 - 0.009 \text{ SG } f C_v^2} \quad (3.13)$$

in which $C_{v(\text{ISA})}$ is the flow coefficient based on the measured pressure drop rather than the net loss.

When a valve is super cavitating no correction to the ΔP_{net} is made. Since the pipe between the valve and the downstream pressure tap has a large amount of vapor surrounding the jet discharging from the valve, the friction loss in that section of pipe is negligible; so $\Delta P_{\text{net}} = \Delta P_{\text{gross}}$.

3.3 INFLUENCE OF PIPE DIAMETER

To understand and properly use the flow and loss coefficients, some discussion might be helpful about situations where corrections to the coefficients are necessary. These coefficients are experimentally determined generally with the valve installed in a pipe that has the same inside diameter as the valve inlet diameter. If the valve is to be

used in a pipe that has a diameter different from the valve diameter, or if the system has a different diameter upstream and downstream, corrections will be necessary to the coefficients. The corrections are needed to account for the contraction and expansion losses and the difference in the velocity heads.

When the valve is used in a pipe that has a constant diameter upstream and down, the ΔP_{net} or ΔH_{net} in the equations is both the net pressure drop and net loss of total energy. The coefficients should be calculated using the net loss of total energy.

If the upstream and downstream pipes have different diameters, the ΔH_{net} or ΔP_{net} in the equations must be corrected for the difference in the velocity heads between the upstream and downstream pipes using the following equation.

$$\Delta H_{\text{net}} = \Delta H_{\text{gross}} + \left(\frac{V_u^2}{2g} - \frac{V_d^2}{2g} \right) - \text{friction loss} \quad (3.14)$$

For a valve installed in a larger pipe (same diameter upstream and downstream) with smooth transitions, the experimental coefficients apply as long as the velocity is based on the valve inlet diameter rather than the pipe diameter. If the valve is installed in a larger pipe with reducing flanges, the pressure drop will be larger and will reduce the flow coefficients.

3.4 NON LINE SIZED VALVES

To improve controllability, it is common to select a control valve that has a diameter smaller than the pipe diameter. This section discusses the influence of a reduced valve size on the flow coefficients and its cavitation performance.

All calculations related to the valve flow characteristics, such as flow coefficients, the cavitation parameter (σ) and torque calculations should be based on the diameter of the pipe at the inlet to the valve. The velocity used with the C_d and K should be the average velocity of the flow at the inlet to the valve not on the velocity in the main pipe. One way to satisfy this constraint is to use Q/A in place of V in the energy equation and in the equations for the loss coefficient K and C_d . One must still be careful to use the area based on the reduced pipe diameter at the inlet to the valve and not the area of the main pipe.

The pressure loss created by the reducers and expansions should be treated just like any other minor or local loss by including their loss coefficients with the summation of all the other minor loss coefficients. The pressure drop of the valve included in the various coefficients (K , C_d and σ) is the net loss caused by the valve only and does not include the loss caused by the contraction and expansion.

The pressure to be used in making pressure scale effects adjustments to the cavitation limits is the pressure in the reduced pipe right at the inlet to the valve. The pressure at the valve inlet is lower than the pressure in the main pipe upstream from

the contraction by the difference in the two velocity heads. Pressure scale effects are discussed in Section 5.3.

Another approach to accounting for non-line sized valves is used by ISA (Ref. 26). Their procedure calculates adjusted values of C_v , K and σ that include the effect of the difference in pipe and valve diameters plus the losses caused by the contraction and expansion. One limitation to this method is that it does not provide for calculating the proper upstream pressure to be used in the pressure scale effects calculations.

Both methods assume that the transitions are contoured with either a standard bell reducer or a tapered conical section. The methods do not apply to a sudden change in pipe diameter such as that caused by a reducing flange. This means that the influence of the expansion and contraction is to only slightly increase the pressure drop and not cause any significant variation to the flow patterns through the valve that would change its flow or cavitation performance. In reality, the reduction in pipe diameter can slightly alter the loss and cavitation characteristics of the valve but this effect is considered small and is ignored.

3.5 INFLUENCE OF UPSTREAM DISTURBANCES

Another factor influencing the flow coefficients and the cavitation performance of valves is disturbances located close to the valve in the upstream piping. If an elbow, tee, or other disturbance is located too close, the approach velocity profile and turbulence can change the capacity, torque, cavitation and stability of the valve. The effect of a downstream disturbance is much less. It can have a small effect on the pressure recovery and, therefore, a small effect on the coefficients. One should be careful in using any experimental data obtained on a valve tested with straight pipe for an installation where there is a disturbance located near the valve.

3.6 INFLUENCE OF FREE DISCHARGE

Free discharge means that the valve either has no pipe or a very short pipe attached to the downstream side so that it discharges directly into the atmosphere or a tank. When a valve is used in this manner, the discharge coefficients C_d and C_v are reduced and the loss coefficient K increased because there is no downstream pressure recovery. Equation 3.5 is the equation traditionally used for free discharge valves where the H_u is the total head upstream which is the sum of the pressure head and the velocity head at the valve inlet. When a valve operates in choked cavitation, there is no normal pressure recovery downstream and the valve operates as if it were free discharging (but into a vacuum). The values of C_d , C_v and K are the same for free discharging and choked cavitation. These values must be obtained experimentally, independent of the normal C_d , C_v and K values.

4.0 EVALUATING CAVITATION LIMITS

Cavitation causes noise, pressure fluctuations, vibrations, erosion damage, and in advanced stages, can reduce the valve's capacity. The acceptable cavitation level for a valve in a given system varies with valve type, valve function, details of the piping layout and duration of operation. For example, for a control valve required to operate continuously, it would be best to select a light level of cavitation for design so there would be no erosion damage and the noise and vibrations would not be excessive. In contrast, a valve intended for only intermittent use as a pressure relief valve could be designed to operate at a very high cavitation level. Economics is also an important consideration. It may be that purchasing a cheap valve, allowing it to cavitate and replacing it periodically is the lowest cost solution unless it results in an unsafe operating condition or a costly shutdown.

To have adequate information for analysis and design, it is necessary to identify and provide experimental data for several levels of cavitation intensity. This section defines six different cavitation design limits covering the full range of possible cavitation intensities. The intensity of the cavitation at each limit is described, including a brief discussion of how each level of cavitation influences the valve and system. The methods used to experimentally determine each limit are described and suggestions are given as to when each limit might be appropriate. These limits are:

1. Incipient cavitation σ_i
2. Critical (or constant) cavitation σ_c
3. Incipient damage σ_{id}
4. Incipient choking σ_{ich} (or K_c)
5. Choked flow σ_{ch}
6. Maximum noise and vibration level σ_{max}

An analytical solution to the problem of cavitation has not been found. Emphasis has, by necessity, been concentrated on laboratory experiments to explore the physics, develop instrumentation to measure cavitation and develop empirical relationships for predicting various levels of cavitation. Since cavitation causes noise, pressure fluctuations, vibrations, pitting, and loss of efficiency, any procedure or device which is sensitive to one or more of these can be used to detect at least one of the cavitation levels. However, no single technique can determine all of the levels. Several electronic instruments including sound level meters, pressure transducers, accelerometers and hydrophones have been used in the laboratory to measure cavitation.

Based on laboratory and/or field experience, valve manufacturers should be able to recommend which cavitation level is appropriate for their valve for a specific application. The choice depends on valve style and opening, piping configuration, duty cycle, desired life, economics, environmental considerations (allowable noise and vibrations) and the consequence of valve failure. ISA (See footnote 1) has defined a

manufacturers recommended cavitation limit (σ_{mr}) which can be any of the limits defined in this section or another limit selected for a given application.

4.1 INCIPIENT AND CRITICAL CAVITATION

The intensity of cavitation at "incipient", consists of light intermittent popping sounds which are barely audible above the background turbulent noise of the flowing liquid. Incipient cavitation is therefore a very conservative design limit and is suggested for use only when no cavitation noise can be tolerated. Since it is so conservative, it has been evaluated only for a few valves and is not generally recommended as a design or operational limit. It is primarily of academic interest to identify the flow conditions where cavitation first occurs. Comparing it to the next higher limit (critical), identifies the range of sigma values where the cavitation intensity is light. The procedure used to evaluate incipient cavitation is similar to that used for critical cavitation and is described later in this section.

The next higher cavitation level is "critical" or "constant" cavitation. This limit was originally defined by the author as "critical cavitation". An ISA committee presently working on a cavitation standard ISA RP75.23 (See footnote 1) decided that a more appropriate and descriptive name for this limit would be "constant" cavitation. One of the reasons for this choice by the ISA committee was to avoid confusion with other limits that have used the term critical. For example, the limit usually referred to as k_c , which corresponds to incipient choking σ_{ich} , has at times been called a critical limit. Since the term "critical" has been used to refer to more than one limit, there is no way to avoid some confusion. Rather than introduce another definition, this guide will retain the name of "critical" for this limit and hope that all future references to k_c and σ_{ich} will accurately recognize that it represents an incipient choking condition and not suggest that it is near incipient or critical cavitation.

At critical cavitation, the intensity is about the same as for incipient, but the cavitation is constant rather than intermittent. The sound is similar to bacon frying or popcorn popping. Critical cavitation is generally not considered to be objectionable and should not cause damage or reduce the service life of a valve. For most applications the critical condition is recommended for what might be termed "cavitation free operation." It would be the design limit for a valve required for continuous operation in an environment where a slight amount of cavitation noise can be tolerated, but loud noise, intense vibrations, and erosion damage must be avoided. Considerable experimental data are available identifying critical cavitation for different types of valves.

Cavitation can be detected both visually and audibly. If viewed through a transparent section, cavitation events can be seen either by the naked eye or with the aid of stroboscopic lighting. Cavitation can also be easily detected by ear except in a system where there is an extremely high background noise level. A trained observer can be quite precise in detecting early stages of cavitation without the aid of electronic instruments. Even when using electronic means to measure the cavitation, it is helpful

to record the audible level of the noise to help interpret the electronic measurements of cavitation.

The accelerometer is probably the most widely used instrument used to detect the onset of cavitation (Ref. 40). It is simple to use, rugged, self contained, portable and has a wide frequency response. It is sensitive to the lightest cavitation and can measure the heaviest levels possible. It is the instrument normally preferred by the author and numerous other researchers to quantify incipient and critical (or constant) cavitation. Typically, laboratory data are obtained by mounting an accelerometer on the valve or pipe just downstream from the discharge flange and recording the flow conditions and the accelerometer output at a number of test conditions ranging from no cavitation to heavy cavitation. Normally, the upstream pressure is maintained constant for each series of tests. Several published articles provide cavitation test procedures (Refs. 40, 43).

For many valves, the values of σ at incipient σ_i and critical σ_c can be evaluated by plotting the accelerometer output versus σ on log-log coordinates as shown in Figure 4.1. The data in the figure are from tests on a butterfly valve.

Region a-b corresponds to very light or no cavitation. The variation in accelerometer output with sigma in this region is due strictly to increased flow noise. Care must be taken in this region to insure that the control valves or other components in the system are not cavitating or causing excessive turbulence and noise. Region b-c covers incipient to light cavitation and c-d from light to the maximum intensity. Incipient and critical cavitation are determined by the intersection of the straight line portions of the data.

When evaluating or using σ_i and σ_c values, it is important to account for scale effects. An example of a scale effect is that the values of σ_i and σ_c for most valves (operating at a constant valve opening) will vary with the system pressure and valve size. This reflects the fact that sigma is not a true similarity parameter because it does not account for all of the factors that influence early stages of cavitation. Accurate analysis requires adjustments for scale effects. Scale effects exist for incipient critical and incipient damage. There are no scale effects at choked cavitation. The causes of scale effects and equations for making corrections are discussed in detail in a subsequent section.

There has been considerable research to investigate different techniques for processing and filtering the electronic signals. Past research has shown that cavitation noise can be detected in all frequencies, whereas hydraulic noise is typically low frequency (generally below 2000 Hz). Therefore, cavitation can be separated from the other disturbances by filtering out low frequency disturbances. In systems where there is significant background noise such as a pump close to the test valve or other hydraulic disturbances, it is typical to filter out everything below about 5000 Hz. For valves tested in relatively quiet systems it is helpful but not necessary to filter.

Another experimental consideration is how to average the accelerometer signal. The data plotted in Figure 4.1 are based on the average accelerometer output. Such

averaging can be done either electronically or (with a little experience) by visually averaging the accelerometer readout meter. The author has had similar success with both electronic and visual averaging.

Figure 4.1 Evaluation of Incipient and Critical Cavitation

The method just described has been used on orifices and numerous valves including butterfly, cone, ball and plug and gate valves. In general, the method should be

valid for any valve style that has a flow pattern similar to the ones just identified, i.e. valves that have only one or two ports and the jet discharges parallel to the pipe. The method does not work as well for globe style valves, especially those that have special multi-port cavitation trim. For these valves, the accelerometer data is still useful but incipient and critical cavitation cannot always be identified by the intersections of the straight lines shown in Figure 4.1.

Several methods have been used for globe and multi-port valves. These methods are based on the fact that incipient and critical cavitation have a very characteristic sound that can be identified quite accurately by a trained observer even without detailed accelerometer data. One procedure used by the author is to record the average accelerometer readings, plot them versus sigma and carefully listen to qualify the cavitation intensity audibly for each data point. If there is the characteristic break in the data when the intensity is at incipient and critical, the graphical method depicted in Figure 4.1 can be used. If not, the magnitude of the accelerometer at the sigma value where the audible intensity is at the critical cavitation level can be used as a reference value for tests at other valve openings or different flow conditions. After sufficient data have been collected, determine if there is a consistent value of the accelerometer corresponding to incipient and critical. Sometimes there will be a slight and consistent increase in the magnitude of the accelerometer level at incipient and critical cavitation with increased valve opening.

Even though this method is more subjective and requires experience, it usually provides results that are accurate enough for design and analysis. This is because there is a range of sigma values near incipient and critical where the cavitation does not vary significantly. It is not possible to determine these cavitation limits to the same degree of accuracy that one can determine the discharge and loss coefficients for the valve.

For multi-stage (sometimes called stack) valves, the flow passages are very small and it is difficult to quantify the cavitation intensity with an accelerometer. An observer can audibly detect when cavitation starts but its intensity tends to remain constant as the pressure drop is increased.

Another method used to detect early stages of cavitation is to look for spikes in the accelerometer data that exceed some reference level. Typically, cavitation events have several times the amplitude of normal turbulent flow noise and can be counted with a peak detection circuit.

4.2 INCIPIENT DAMAGE

As the cavitation level is increased beyond critical, the intensity of the noise and vibrations as well as the potential for erosion damage increases rapidly. The margin between the light cavitation level at the critical limit and the point of incipient damage varies with details of the valve design. Because of this, the intensity of the noise and vibrations at the point of incipient damage varies depending on the test device. For an orifice, with a small orifice to pipe diameter ratio, and for certain valves, fairly heavy cavitation is required before pitting occurs. Once pitting begins, however, only slight

increases in the system velocity, or a small drop in ambient pressure can rapidly accelerate the damage.

Erosion damage does not begin until the vapor cavities have sufficient energy and the collapse occurs on or very near a solid boundary. If the valve is designed like an orifice with a large expansion ratio, the collapse occurs away from the boundaries, and fairly heavy cavitation is possible without erosion. Normally, erosion damage begins well before the valve starts to choke and before it reaches its maximum noise and vibration intensity. Incipient damage is usually the next limiting operating condition after critical. It cannot be evaluated from accelerometer data.

A procedure has been developed for determining incipient damage, σ_{id} , but only limited data are available due to the experimental difficulties involved in measuring the onset of erosion damage. The limit corresponds to conditions where pitting is first detected on a soft aluminum test specimen placed at the boundary. Soft aluminum has been used as the test material to reduce testing time and to provide a conservative value for σ_{id} .

The initial step in the test procedure involves determining the location inside the valve body or in adjacent piping where cavitation damage first occurs. For some devices like an orifice and certain valve styles, it is quite obvious where damage will first appear. If it is not obvious, one method of locating the damage area is to coat the valve body and adjacent pipe with paint and operate the valve in moderate cavitation for a few hours and inspect the valve for paint removal. If none is observed, the cavitation level can be increased and the test repeated. Once the damage zone is identified, aluminum specimens are prepared, carefully polished to a mirror finish and inspected with a low power magnifying glass to be sure the surface does not have blemishes that look like cavitation pits. The specimens are then installed in the valve or pipe. They should be installed flush with the boundary so they do not create local disturbances that will effect the cavitation.

With the specimens installed, the valve is operated initially at a low level of cavitation for about 30 minutes. The specimens are removed, inspected and all pits counted, the zone of maximum pitting is identified and the corresponding pitting rate determined. This procedure is repeated at increasing levels of cavitation until data are collected at several pitting rates.

The recommended method of processing the data and evaluating incipient damage σ_{id} is to plot the experimentally determined maximum pitting rate versus pipe velocity. Figure 4.2 shows such data from tests on an orifice. Incipient damage is defined as the flow condition that causes a pitting rate of 1 pit/in²/min on the soft aluminum. All of the cavitation damage testing that the author has been involved in has shown that the log of the pitting rate varies linearly with the velocity so only a few data points are needed to identify incipient damage.

Figure 4.2 Determination of Incipient Damage Velocity

Since σ_{id} is evaluated using soft aluminum and valves are fabricated from stronger materials, is the σ_{id} limit too conservative? To answer the question consider the rate of increase in pitting rate with an increase in velocity. This is displayed graphically by the slope of the lines in Figure 4.2. The steepness of the lines shows that

the damage increases rapidly with velocity. Experiments have shown that the rate of weight loss is exponentially proportional to the velocity, ie.

$$\text{Weight Loss} \propto (V - V_{id})^n \quad (4.1)$$

in which V_{id} is the velocity at incipient damage. The exponent n has been evaluated experimentally to typically vary between about 4 and 8 (Refs. 20, 35, 36). Operating a valve beyond incipient damage, therefore, subjects it to the possibility of rapid erosion. Figure 4.2 shows that for a velocity increase of 13 percent, the pitting rate increases about 1000 times. It is, therefore, important to be conservative in estimating σ_{id} and not to operate at $\sigma_{sys} < \sigma_{id}$ because of the potential rapid increase in damage with increased velocity.

The increase in damage potential with velocity does not continue without limit. Once the valve starts to choke, the location of the damage moves and the intensity usually decreases due to the cushioning effect of free air that comes out of solution during the heavy cavitation. Some valves can actually operate damage free when fully choked. In such cases, the collapse and therefore the damage occurs in the pipe downstream from the valve rather than inside the valve.

For tests on some valves where the flow passages are small, such as for globe valves with cavitation trim, it is sometimes not possible to count individual pits and use the suggested test procedure for determining σ_{id} . For such valves, the author has determined when pitting first occurs by changing the flow conditions in small increments and observing the damage specimens for a roughening of the surface. Fine grained cavitation produces a surface that looks like it has been etched.

A method has been proposed for approximating cavitation damage intensity and estimating service life for a valve when the valve must operate at intensities beyond incipient damage (Ref. 47). This procedure can be applied to systems where $\sigma_{ch} < \sigma_{sys} < \sigma_{id}$. At sigma values greater than σ_{id} there will be no damage so the method does not apply. Above σ_{ch} , the cavitation collapse and the damage may move away from the valve and again, the method does not apply. This method is in the developmental stage and needs to be developed further.

The pressure downstream from the valve has a significant influence on the magnitude of the incipient damage cavitation limit. If it is close to or below atmospheric pressure, air can come out of solution and suppress the damage. This is good from the point of view of preventing cavitation damage but it can produce serious errors in scaling the experimental data to higher pressures. This is discussed again in the section on scale effects.

4.3 CHOKING CAVITATION

Once a valve is operating in the damaging cavitation range, the next limits of interest would be some limiting noise or vibration level and choking cavitation. Limited

data collected by the author on several different types of valves indicates that maximum noise intensity occurs after a valve starts to choke. Therefore, the next limiting operating condition in order of increasing cavitation intensity, is generally choking.

Choking cavitation is fairly simple in concept, but there are a number of factors that make it a relatively complex phenomenon. To understand the choking process, consider flow through an orifice. The orifice is used for discussion because it is easier to analyze than most valves due to its simple geometry and a well defined separation region.

Figure 4.3 demonstrates the effect of choking cavitation on the pressure recovery for an orifice. The pressure variation for flow through an orifice can be shown graphically by plotting the hydraulic grade line (HGL). The upper line (HGL #1) of Figure 4.3 represents the variation of pressure, along the centerline streamline, for non-choking flow through an orifice. Minimum pressure occurs just downstream from the orifice at the vena contracta. Pressure recovery occurs as the jet is dissipated and a normal hydraulic gradeline is re-established.

Figure 4.3 Hydraulic Grade Lines For Choking Cavitation

Figure 4.3 shows how the net loss is evaluated. Graphically, it is done by projecting the upstream and downstream hydraulic grade lines and measuring the vertical separation between them. This is done numerically by measuring the gross pressure drop ($P_1 - P_2$) and subtracting the calculated friction loss between the two

pressure taps (see Eqn. 3.16). Note that P_2 must be far enough from the orifice or valve that it is beyond the region where the hydraulic grade line is re-establishing. Normally six diameters is the recommended distance (Ref. 17).

Once a valve or orifice begins to choke, the magnitude of the loss and flow coefficients change. To understand this, refer again to Figure 4.3. The effect of choking can be explained by discussing the four hydraulic grade lines. HGL #1 represents typical flow conditions before choking occurs. It shows normal pressure reduction and recovery of the HGL. Note that in Figure 4.3 the absolute pressure line and the vapor pressure line are offset vertically so the pipe and orifice can be shown between them.

If the upstream pressure is maintained constant and the flow increased by lowering the pressure at P_2 , eventually the average pressure at the vena contracta P_j will drop to vapor pressure. HGL #2 in Figure 4.3 represents such a condition. Since the flow through a valve or orifice is determined by $P_1 - P_j$, the flow cannot be increased by further lowering the pressure at P_2 because the minimum average pressure at the valve, P_j , cannot drop below vapor pressure.

The orifice can be used as a flow meter because the flow is directly proportional to the square root of the pressure differential. When the orifice is to be used as a flow meter, it is preferable to use the maximum pressure drop ($P_1 - P_j$) for maximum meter sensitivity. P_j is the average pressure at the point of maximum jet velocity (vena contracta). Once the orifice coefficient is calculated, the flow can be determined by simply measuring the pressure differential.

Using ($P_1 - P_j$) in the equation for the orifice when used as a flow meter also has the advantage that the orifice coefficient is not effected by choking cavitation. Note that the flow orifice coefficient should not be confused with the flow and loss coefficients defined in Equations 3.1-3.4. In those equations, the net pressure drop must be used rather than ($P_1 - P_j$).

Q_{ch} is the maximum possible flow through the orifice at the given upstream pressure. Up to this point the discharge coefficient (based on Eqn. 3.1) remains constant and equal to its value for non-cavitating flow. Figure 4.4 discussed in the next section defines the flow Q_{ch} and pressure drop ΔP_{ch} at choking cavitation.

If the pressure in the downstream pipe at P_2 is further reduced, the flow and the pressure at P_j remain constant but the vapor cavity increases in length. HGL #3 in Figure 4.3 shows this condition.

As the downstream pressure is again reduced, the vapor cavity eventually extends beyond P_2 as shown by HGL #4, $P_2 = P_v$ and $\sigma = 1.0$. This condition is referred to as super cavitation, flashing or fully choked flow. If observed through a transparent pipe, one would see what appears to be a free discharge jet, but the jet is surrounded by a vapor cavity instead of air. The vapor pocket may extend for many diameters and the

cavity collapse can occur far from the valve. In one test, the author observed that the vapor pressure extended for about one hundred pipe diameters beyond the test valve.

At some downstream location, where the pressure and momentum forces balance, the cavity collapses, the pipe again flows full and a normal pressure gradient is established. An orifice, and some types of valves, can operate in super cavitation without any erosion damage to the orifice or to the valve. However, at the pipe location where the super cavity collapses, severe damage is likely to occur.

As the system conditions change from choked flow to super cavitation, the pressure drop increases but Q_{ch} remains constant. Therefore, the discharge and flow coefficients reduce and the loss coefficient increases. Applying the energy equation to determine the flow rate in a system where a valve or other component is choking, requires special calculations. The correct procedure will be demonstrated with examples in Section 6.8.

For valves, choking cavitation must be determined experimentally. For an orifice, however, the flow rate at a given upstream pressure corresponding to the onset of choking cavitation can be estimated by applying the energy equation between P_1 and P_j . This calculation requires: 1) knowing the contraction coefficient for the orifice for a free discharge condition, 2) setting $P_j = P_v$ and 3) either estimating the loss or assuming that the loss due to the contraction is negligible.

When a valve chokes in a pipe system, it requires a trial and error solution to determine the actual flow rate. The process involves:

1. Applying the energy equation to the system assuming that the valve is not choking.
2. Analyzing the valve to determine if it is choking. This involves determining the pressure just upstream from the valve, knowing its valve opening and therefore its C_d , calculating the sigma for the valve σ_{sys} and comparing it with experimental σ_{ch} data.
3. If σ_{sys} is less than or equal to σ_{ch} the valve is choking.
4. Calculating ΔP_{ch} by knowing σ_{ch} and the previously calculated P_u , and then calculate the choking flow by substituting ΔP_{ch} into the C_v equation. (This flow will be less than the original calculated flow using the energy equation.)
5. The P_u originally calculated will be too low because the flow was too high.
6. Repeating the calculations of P_u using the choking discharge and repeat the calculations.

The effect of a choking valve on the flow is greatest for systems where the valve produces a large percentage of the total system losses. This occurs when a valve is throttled and the friction and other minor losses are small.

4.3.1 Evaluating Incipient Choking The normal method of evaluating incipient choking does not require a measurement of the cavitation intensity. It is evaluated by finding the flow conditions, and the corresponding value of sigma, where the discharge coefficient is first reduced by cavitation. It is similar to the procedure used to evaluate the required net positive suction head for pumps.

The procedure is demonstrated using data for a 10-inch butterfly valve shown in Figure 4.4 (Ref. 42). In the absence of cavitation and up to heavy cavitation, the discharge (or velocity) is proportional to the square root of the net pressure drop. This is reflected in the figure by the straight-line portion of the data at a slope of 2:1. In this region, the C_d , C_v and K coefficients (Eqn. 3.1-3.3) are constant. As the valve begins to choke, the data points deviate from the straight line and the line eventually becomes horizontal. When the pressure at P_2 (Fig. 4.3) drops to vapor pressure super cavitation is reached.

The intensity of cavitation at selected points is noted in Figure 4.4. Note that in the region where choking begins and the data start deviating from a straight line, the intensity of the cavitation is noted as heavy or very heavy. Tests on numerous valves have documented that cavitation noise intensity, vibration level and erosion potential usually reach their maximum in the region of choking cavitation.

The point in Figure 4.4 where the data first deviate from a straight line has been identified in the literature as a cavitation index k_c which was defined in Equation 2.7 as:

$$k_c = \frac{\Delta P_{kc}}{P_1 - P_v} \quad (4.2)$$

in which ΔP_{kc} is the pressure drop at the flow where the device begins to choke. Unfortunately, this limit was originally defined as "incipient cavitation" (Ref. 33).

As shown by the data in Figure 4.4 the cavitation is very heavy at choking and is far beyond incipient. It may be that the reason behind the unfortunate choice of terminology for this limit is to be consistent with the definition of the required net positive section head used for pumps. It has been widely taught that a pump will operate "cavitation free" if the available net positive section head is greater than the required net positive section head. Such is not the case. The required net positive section head ($NPSH_r$) represents the suction pressure just above the pressure where the pump begins to lose efficiency, or begins to choke due to heavy cavitation. If a pump is set to operate at $NPSH_r$, cavitation and erosion damage are definite possibilities. Research has documented that to avoid cavitation, the net positive section head that must be provided for the pump, called the "available NPSH" ($NPSH_a$), must be significantly greater than $NPSH_r$ (Refs. 10, 13).

In this guide, the flow condition corresponding to the k_c limit is defined as incipient choking and is evaluated by the equation:

$$\sigma_{ich} = \frac{P_1 - P_v}{\Delta P_{kc}} \quad (4.3)$$

or

$$\sigma_{ich} = \frac{1}{k_c} \quad (4.4)$$

In Figure 4.4 data are shown for three different valve openings. Notice that for the two lower curves (largest valve openings) there is a transition region where the valve chokes gradually. At the smallest valve opening, choking occurs abruptly. This is explained further in the next section.

Incipient choking may be useful as an operating limit for a valve when cavitation intensity is of no concern but you do not want the flow capacity to be reduced. Applications would include: 1) Valves that operate only infrequently so that the erosion damage will not be serious, 2) Valves that are designed specifically to operate in choking cavitation, and 3) Valves that can easily and economically be replaced when the damage becomes excessive.

4.3.2 Evaluating Choking Cavitation Referring to the data in Figure 4.4 there is another design and operating limit that can be defined in the choking range. The extension of the straight lines identifies what is labeled ΔP_{ch} and Q_{ch} . This point corresponds to the traditional definition of choked cavitation. The choking discharge Q_{ch} and the choking pressure drop ΔP_{ch} can be determined graphically by intersecting the two straight lines as shown in the figure or by calculation. Both methods give the same result. The simplest method is by calculation using the experimentally measured Q_{ch} and the C_v value, based on non-cavitating flow, to evaluate ΔP_{ch} . The result is:

$$\Delta P_{ch} = SG \left(\frac{Q_{ch}}{C_v} \right)^2 \quad (4.5)$$

in which SG is the specific gravity of the liquid. The corresponding value of sigma at choking cavitation, σ_{ch} , can be calculated by substituting ΔP_{ch} into Equation 2.1. The result is:

$$\sigma_{ch} = \frac{P_1 - P_v}{\Delta P_{ch}} \quad (4.6)$$

The abruptness of the change in slope of the data as shown in Figure 4.4 and the extent of the transition region as choking occurs, depends on the device and the valve opening. For the butterfly operating at small valve openings (upper line in Fig. 4.4) and for orifices, and globe valves, experimental data show that σ_{ich} and σ_{ch} are almost

the same (ie. the change of slope is abrupt). For butterfly and segmented ball valves at larger openings, the data in Figure 4.4 (lowest set of data) show that the choking region extends from a differential pressure of about 45 psi to about 95 psi where the valve is fully choked. Note that although ΔP varies 100% over the choking range the flow only changes about 13%.

Figure 4.4 Determination of Choking Cavitation for a Butterfly Valve

For the butterfly valve, the reason why the choking range has a transition region at larger valve openings is due to the configuration of the flow passage. The butterfly valve has two irregular "lens shape" flow areas. The position of the disc at a partial opening concentrates more of the flow through one of the flow areas making one side more susceptible to cavitation than the other. The restricted flow at the narrow ends of each "lens shaped" area creates locally higher shear stresses causing cavitation (and eventually choking) to first occur in these regions. Eventually the entire circumference of both flow areas will cavitate. As a result of the non-uniform distribution of cavitation potential, choking occurs in different locations inside the valve at different flow rates. This causes the choking transition region. For an orifice and for valves (at small valve openings) that have a single well defined separation zone, the entire valve chokes at the same flow. For these valves σ_{ich} approaches σ_{ch} .

When the pressure at the downstream pressure tap (usually located six or more pipe diameters from the valve) drops to vapor pressure, the valve is fully choked. This condition is called super cavitation. The value of sigma for a valve operating in super cavitation is always equal to 1.0 because $\Delta P_{ch} = P_1 - P_v$, ie, $\sigma_{super} = 1.0$. Also, the flow is the same as for σ_{ch} . As a result, it is really not a condition that can be easily evaluated experimentally nor can it be predicted from experimental data. Once σ_{ch} is reached and the flow is constant (for a given upstream pressure), it is difficult to calculate the extent of the region subject to vapor pressure. It is possible to determine when a valve will choke, but difficult to predict the cavity length when a valve is super cavitating. However, if $\sigma_{sys} < \sigma_{ch}$, it can be reasonably assumed that the valve is super cavitating.

The intensity of cavitation and the corresponding noise, vibration and erosion damage at the valve generally are at their maximum in the region where the valve chokes. However, if the valve moves into super cavitation the situation is quite different. For valves like butterfly, ball and cone valves and orifices, where the jet exits the valve parallel to the pipe, the cavities can collapse a considerable distance from the valve and little damage is likely to occur at the valve. However, in the downstream pipe where collapse occurs, serious problems may be encountered. Elbows and other piping components located a considerable distance downstream from a cavitating valve can sustain severe cavitation damage if the valve is operating in a super cavitating condition.

Tests have performed using an orifice installed in an elevated pipe with an elbow located about 50 pipe diameters downstream. The average pressure in the discharge pipe could be maintained at vapor pressure due to the elevation of the pipe relative to the downstream reservoir. One of the purposes of the tests was to see if damage could be eliminated in a super cavitating system by gradually bringing the pressure up from vapor pressure by the elevation change as the water flowed down the vertical pipe after it was turned downward by the elbow located about 50 diameters from the cavitating orifice.

When the system was operated in super cavitation, the only location where damage was observed was at the elbow. The impingement of the jet on the outer surface of the elbow caused local cavity collapse and erosion damage.

For valves like the standard globe valve designs (with flow from under the seat and no special cavitation trim installed) the jet impinges internally against the valve body and some of the cavitation will collapse inside and cause erosion damage even when the valve is super cavitating.

Valves with multiple losses in series and systems of multiple orifices operate most efficiently when they are designed so that each stage is operating at the same level of cavitation. Such valves do not manifest the typical choking behavior. They actually choke but there is little change in the overall coefficients or on the system capacity. Consider a system with several orifices in series designed so none of them cavitate under normal conditions. With such a design, the upstream orifices are sized to produce most of the pressure drop. The last orifice typically provides a relatively small drop. For example, one system that the author designed had four orifices in a series. Each orifice was designed to operate at critical cavitation. The pressure drops for each orifice starting with the one upstream was 240, 120, 50 and 20 psi respectively. If the downstream pressure in that system were dropped to vapor pressure, the last orifice would choke but the others would not. In this example, the discharge coefficient C_d and the flow coefficient C_v for only the last orifice will be reduced because of the choking. The net effect on the overall coefficient of the system of four orifices will be negligible because the other orifices do not choke.

One must be careful when considering selecting σ_{ich} or σ_{ch} for an operational limit. It may be the worst condition for operating some valves, since it is near the point where the rate of erosion damage, noise and vibrations are maximum.

4.3.3 Evaluating Super Cavitation Super cavitation occurs when the average pressure downstream from a valve or orifice is at vapor pressure for a distance of at least six pipe diameters. This means that the absolute pressure at the location of the downstream pressure tap is at vapor pressure and the value of sigma is identically 1.0, Substituting $\Delta P_{net} = P_1 - P_v$ into the denominator of the sigma equation gives:

$$\sigma_{super} = \frac{(P_1 - P_v)}{(P_1 - P_v)} = 1.0 \quad (4.7)$$

As was discussed in relationship to Figure 4.3, once the valve chokes, further lowering of the downstream pressure does not increase the flow but it does increase the length of the vapor cavity. Since the flow is identical to the flow determined at choked cavitation, Q_{ch} , it is almost impossible to predict the conditions in the downstream pipe that will cause super cavitation and the length of the cavity. Only slight variations in the downstream pressure can result in large changes in the size of the super cavity.

The best guidance that can be offered about how to predict when super cavitation will occur is to first determine ΔP_{ch} (Eqn. 4.5), and then calculate $P_d = P_u - \Delta P_{ch}$. If the system P_d is much less than the calculated P_d , the valve will be operating in super cavitation. The lower the downstream pressure, the longer the length of the super cavity will be. The maximum length is normally limited by obstructions in the pipe. The cavity generally cannot extend beyond the first obstruction and cause damage to other downstream elbows, valves or other obstructions. With super cavitation there is no need to account for friction losses and determining a net pressure drop. There is essentially no friction loss in the pipe where the super cavity exists.

Super cavitation is sometimes referred to as flashing. In the vaporization phase of choking cavitation, flashing and super cavitation are essentially identical. In this guide the distinction between these two occurrences relate to the mechanism causing the phase change from the vapor phase back to the liquid phase. The author prefers to use the term flashing to refer to systems where hot water under high pressure is flashed into steam by a sudden pressure reduction. Condensation back into the liquid stage occurs primarily by cooling and there is no violent collapse of the cavities and generally no classical cavitation erosion damage. With super cavitation, the collapse of the vapor phase occurs violently due to pressure recovery. Severe cavitation damage generally occurs in the collapse region.

This limit is not appropriate for design because the length of the cavity, the location of the damage and the effect on the downstream piping is so hard to predict.

4.4 PRESSURE RECOVERY FACTOR F_L

Another parameter that can be used to quantify flow conditions at choked cavitation is the liquid pressure recovery factor F_L defined as the ratio of the choked C_v to the non-choked C_v :

$$F_L = \frac{C_v \text{ (choked)}}{C_v \text{ (non-choked)}} \quad (4.8)$$

This relationship equates to:

$$F_L = \frac{Q}{C_v \sqrt{\frac{(P_1 - F_f P_v)}{sg}}} \quad (4.9)$$

in which C_v is the non-choking valve flow coefficient, Q is the choked flow and F_L is the liquid pressure ratio factor (Ref. 16). For most practical applications where the water temperature is below 200 degrees Fahrenheit, F_f can be ignored because its effect on F_L will be less than 1% which is about the accuracy of experimentally evaluating F_L . The

terms in Equation 4.9 must all be in absolute units. Setting $F_f = 1$, F_L and σ_{ch} can be related by:

$$F_L = \frac{1}{\sigma_{ch}^2} \quad (4.10)$$

When a valve is operating beyond choked cavitation as defined by Equation 4.6, a special calculation procedure must be followed to calculate the flow. The problem is that when the valve is operating at $\Delta P_{sys} > \Delta P_{ch}$, the flow does not increase with increased pressure drop. It will be equal to Q_{ch} . The procedure is as follows: 1) determine if the valve is choked by comparing σ_{sys} with σ_{ch} . If $\sigma_{sys} < \sigma_{ch}$, the valve is choked. 2) Find ΔP_{ch} from Equation 4.6 and 3) substitute ΔP_{ch} into the C_v equation to get the flow. This process is demonstrated with examples in Section 6.6.

4.5 MAXIMUM NOISE AND VIBRATION LEVEL

The flow condition where the noise and vibration level is maximum is identified as σ_{max} . This limit is not suggested as a design limit but is included to indicate where the maximum cavitation intensity is in relation to the other limits. It is considered useful in selecting the most appropriate limit for each valve application.

5.0 SCALING AND SCALE EFFECTS

Virtually all cavitation data are obtained experimentally and often with reduced size models operated at reduced pressure and velocity. If complete similitude is not achieved geometrically, dynamically and with the fluid properties, the experimental data may not accurately represent the performance of the prototype. Such discrepancies between model and prototype are referred to as scale effects. This must be distinguished from "scaling" data. "Scaling" refers to extrapolating data from one condition to another, assuming complete similitude. This means that the value of the cavitation index at the selected operating limit does not vary with size or pressure.

An example of scaling data would be as follows. Assume tests on a model valve operating at a valve opening where $K = 2.5$ indicated that its critical cavitation index is $\sigma_c = 2.1$ at the following test conditions: $P_1 = 56$ psia, $P_v = 0.2$ psia. Using Equation 2.1 for σ , the net pressure drop is calculated as: $\Delta P_{net} = (56 - 0.2)/2.1 = 26.57$ psi. The corresponding velocity of the fluid is calculated using the equation for the loss coefficient K (Eqn. 3.2), being careful to use consistent units.

$$V = \sqrt{\frac{2 \Delta P_{net}}{K \rho}}$$

Ignoring scale effects, calculate the pressure differential for the prototype valve operating at $P_1 = 190$ psia and $P_v = 2.41$ psia. With no scale effects the prototype valve would have the same σ_c at any operating pressure and be independent of the valve size. One could therefore "scale" the experimental data and predict the ΔP across the prototype valve at critical cavitation for any P_1 using the sigma in Equation 2.1, as follows:

$P_1 = 190$ psia, $P_v = 2.41$ psia, and using Equation 2.1 for σ ,
 $\Delta P_{net} = (190 - 2.41)/2.1 = 89.3$ psi, and the velocity for the prototype
would be calculated by substituting the new ΔP_{net} into the equation:

$$V = \sqrt{\frac{2 \Delta P_{net}}{K \rho}}$$

This procedure is similar to scaling cavitation data from a model pump to a prototype pump which simply involves assuming that the $NPSH_r$ is the same for both pumps.

5.1 CAUSES OF SCALE EFFECTS

Unfortunately, cavitation is a classical example for which the techniques of dimensional analysis have not produced a sigma that is free of scale effects for all

conditions. The difficulty is caused by the numerous variables that effect the cavitation process. The primary variables that influence inception and subsequent levels of cavitation and are included in the derivation of sigma are: 1) The critical pressure required to form a vapor cavity (usually assumed to be the liquid vapor pressure), 2) absolute pressure, 3) velocity or pressure drop and 4) boundary geometry.

Some of the variables that are not included in the formulation of the cavitation parameter are factors influencing vaporization of the liquid such as viscosity, surface tension, air content and water quality. There are also factors that influence the growth and collapse of the cavities. These include turbulence intensity and the time that the bubble is in the low pressure zone, which in turn are effected by velocity and size of the system. Temperature is important because of its effect on vapor pressure.

If there are significant differences in the fluid properties between the water used in the laboratory and the water in the system where the valve is installed, some errors can result. This is particularly a problem with cavitation tests in recirculating water tunnels. Continual recirculation of the water can alter the air content and nuclei spectrum of the water. Even when artificial means are used to control the nuclei, significant effects on the onset of cavitation have been observed (Refs. 14, 15).

Scale effects information has been accumulated by the author and his students (Refs. 6, 7, 9, 11, 23, 24, 25, 32, 35, 36, 37, 38, 39, 42), by the Metropolitan Water District of Southern California (Ref. 46) and others (Refs. 3, 4, 5, 28, 29, 30, 31). The impact of this research has been the evaluation of sigma corresponding to different levels of cavitation for several types and sizes of valves plus the quantitative evaluation of scale effects associated with changes in the size and variations of operating pressure.

The two most significant cavitation scale effects that have been identified are associated with changes in valve size and operating pressure (Ref. 11). Experimental data for valves show that there are significant scale effects in the early stages of cavitation due to increased velocity or pressure and size (Refs. 11, 37). The scale effects due to pressure or velocity may be related to turbulence scaling. For certain geometries, the magnitude of the fluctuating pressures responsible for cavitation may not scale proportional to V^2 . Data show that valves will cavitate worse when operating at a higher pressure. The exception to this is for choking cavitation. There are no scale effects at choking.

Some effort has been devoted to quantify the cavitation characteristics and evaluate scale effects for valves, orifices, elbows by establishing values of sigma corresponding the various levels of cavitation. The purpose of this section is to explain scale effects, identify which levels of cavitation have scale effects, quantify them and demonstrate the proper application of experimental cavitation data by identifying how to adjust for scale effects associated with variations in size and operating pressure. Such adjustments are necessary because failure to correct for these scale effects when using tests on relatively small valves obtained at low pressures can result in significant errors in predicting cavitation for larger valves at high pressures. It is important to

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note that when making scale effect adjustments it is necessary to adjust the experimental sigma values before comparing it to the system sigma value.

In fluid systems, many scale effects are associated with the Reynolds Number, which is a dimensionless parameter frequently used to account for the influence of viscous forces. It is defined as:

$$Re = \frac{Vd}{\nu} \quad (5.1)$$

in which ν is the kinematic viscosity, V the average velocity in the pipe and d the pipe or valve diameter. Note that the Reynolds Number can change with variations in velocity, size or viscosity.

In the past, some researchers have correlated cavitation scale effects with the Reynolds Number. However, cavitation data for orifices and valves show that Re is not a parameter that produces similitude for all cavitation conditions. Research shows that the scale effect is not related to the Reynolds Number but is independently related to variations in pressure (or velocity) and to size. (Ref. 43).

Figure 5.1 shows the variation of σ_c with Re for orifices installed in pipes of different sizes with a constant ratio of orifice diameter to pipe diameter. It is observed from the figure that for a constant pipe size, σ_c does not vary over a range of Re . In this case Re changes by increasing the pipe velocity while varying the upstream pressure and the flow such that σ and the cavitation intensity are constant. The constancy of sigma with changing velocity indicates that there is no pressure scale effect at critical cavitation for orifices.

If, however, the velocity and cavitation intensity are held constant and Re is increased by increasing the pipe diameter, the value of σ_c increases. For certain values of Re there are two values of σ_c . It is, therefore, apparent that there is an effect on σ_c associated with changes in the size of the pipe which are not properly accounted for with Re . Therefore, this scale effect is treated as a size scale effect rather than a Reynolds Number scale effect.

The size scale effect may be caused by the increased time of the nuclei in low pressure region as the system size increases. For a larger system, more of the nuclei can grow to a critical diameter because they are subjected to low pressures for a longer time due to the larger size of the separation region and the lower frequencies of the pressure fluctuations. As size increases, the nuclei may also be influenced more by the higher frequency fluctuations because they are subjected to more cycles.

Figure 5.1 Size Scale Effects for Orifices. (Ref. 43)

5.2 CAVITATION LIMITS EFFECTED BY SCALE EFFECTS

Scale effects are most prevalent in the early stages of cavitation. Consequently, incipient and critical cavitation limits require the largest scale effects adjustments. In contrast, choked cavitation does not require any scale effects adjustments. This is because the choking condition is controlled by the local average velocity and average pressures in the system. Turbulence, eddy formation and dissipation, water quality and various other factors that cause scale effects associated with the onset of cavitation do not influence choking cavitation. For example, if the flow area and the coefficient of contraction for a constriction (such as an orifice) is known, the energy equation can be used to predict the velocity and pressure reduction at the vena contracta. For a given upstream pressure, the flow required to produce vapor pressure at the vena contracta can be calculated and σ_{ch} determined.

For valves, it is difficult to measure the cross sectional areas of the flow passage inside valves and analytical means are not available to predict the coefficients of contraction for complicated flow passages. Consequently, it is necessary to depend on tests results. It is generally easier to measure choked cavitation directly than measure the internal areas of the valve and the contraction coefficients and then calculate σ_{ch} .

Not all situations require a scale effect adjustment. The need to make such a correction to experimental data varies with both the level of cavitation and with the type of device. Table 5.1 summarizes those situations where scale effect adjustments have been identified.

Table 5.1 Situations Requiring Scale Effects Adjustments

<u>Device</u>	<u>Incipient</u>	<u>Critical</u>	<u>Inc. damage</u>	<u>Choking</u>
Orifices	SSE	SSE	PSE	None
Valves	SSE & PSE	SSE & PSE	PSE	None

Note: SSE refers to size scale effects adjustments
PSE refers to pressure scale effects adjustments

The rest of this section presents equations and procedures for making size and pressure scale effects adjustments for valves and orifices.

5.3 PRESSURE SCALE EFFECTS

When experimental data for sigma at incipient, critical or incipient damage are plotted on a log-log scale versus P_1 , P_v , the result is a series of straight lines of approximately constant slope. There is one line for each valve opening. The data produce simple scale effects equations which are valid for incipient, critical and incipient damage. The development of these equations is given in more in detail in other references. (Ref. 43) The resulting equations are:

$$\sigma_i = \text{PSE} (\sigma_{i \text{ ref}} - 1) + 1 \quad (5.2)$$

$$\sigma_c = \text{PSE} (\sigma_{c \text{ ref}} - 1) + 1 \quad (5.3)$$

$$\sigma_{id} = \text{PSE} (\sigma_{id \text{ ref}} - 1) + 1 \quad (5.4)$$

where

$$\text{PSE} = \left(\frac{P_1 - P_v}{P_{1 \text{ ref}} - P_{v \text{ ref}}} \right)^x \quad (5.5)$$

in which $\sigma_{i \text{ ref}}$, $\sigma_{c \text{ ref}}$ and $\sigma_{id \text{ ref}}$ are experimentally determined reference cavitation index values evaluated at an absolute reference test pressure $P_{1 \text{ ref}}$ and reference absolute vapor pressure $P_{v \text{ ref}}$. The exponent X has been evaluated from extensive experimental data. P_1 and P_v are the absolute pressures at which the adjusted sigma is desired. When there are no pressure scale effects $X = 0$. The reason for the -1 and +1 in the equations is that the original equations were developed using σ_1 (Eqn. 2.2) and $\sigma = \sigma_1 + 1$.

Values of X for scaling incipient and critical cavitation for several butterfly valves are listed in Table 5.2. The seven butterfly valves listed in Table 5.2 represent extremes in body and disc design. The average value of X for these valves is 0.28. The maximum deviation of X from the average value is about 15 percent. This amount of deviation generally causes an insignificant variation in the values predicted by Equations 5.2 - 5.3, unless the prototype pressure is much larger than the experimental value. In the absence of experimental pressure scale effects data on a specific butterfly valve, scale effects caused by difference in pressure between model and prototype can be approximated using the average value of X obtained from Table 5.2. To be more conservative, one could use the maximum value of the exponent in Table 5.2.

The X values for the globe and needle valves listed in Table 5.2 are somewhat lower than for the butterfly or ball valves. This indicates less pressure scale effect since when $X = 0$, pressure scale effects are absent. The reason for this decreased scale effect is associated with details of the valve geometry near the seat. The jets from needle and globe valves are similar to that for an orifice. For all three devices, the jets discharge into a sudden enlargement created by sharp discontinuities of the boundaries and are completely surrounded by liquid. This provides optimum conditions for suppressing cavitation and minimizing pressure scale effects. In contrast, the flow around the disc of a butterfly valve does not have a single well defined separation zone. This more complicated flow pattern seems to increase the pressure scale effects.

Table 5.2 Pressure Scale Effects Exponents for σ_i and σ_c for Valves (Ref. 43)

Valve Type	Exponent X	Pressure Range of Tests, psi
4-inch Butterfly	0.28	17 to 96
6-inch Butterfly	0.28	20 to 190
12-inch Butterfly	0.28	17 to 174
12-inch Butterfly	0.24	14 to 135
12-inch Butterfly	0.28	12 to 135
20-inch Butterfly	0.30	10 to 80
24-inch Butterfly	0.24	22 to 107
Average=	0.28	
2-inch Ball	0.30	17 to 246
8-inch Ball	0.28	7 to 149
12-inch Ball	0.24	14 to 90
Average=	0.27	
24-inch Skirted Cone	0.22	66 to 220
16-inch Globe	0.14	66 to 194
8-inch Pelton Needle	0.14	66 to 150

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Research has shown that there are no pressure scale effects at incipient and critical cavitation for orifices (Ref. 45) but that there is a pressure scale effect at

incipient damage (Ref. 36). This can cause a problem when scaling the data to high pressures because the incipient damage sigma eventually will become larger than the critical sigma. It is the opinion of the author and his associates that pressure scale effects reduce and probably disappear at higher pressures. The test data used to evaluate the pressure scale effects have been limited to about 250 psi due to laboratory limitations. Additional research is needed to resolve this conflict. It is suggested that when scaling to pressure above about 300 psi the results may be overly conservative. Until this problem is cleared-up it is recommended that the PSE adjustments still be made up to the actual system pressure and realize that the results are likely quite conservative for pressures above 300 psia.

Table 5.3 lists X values for incipient damage for a butterfly valve, a globe valve and an orifice. The value of the exponent X varies with the type of device.

Table 5.3 Pressure Scale Effects Exponents for σ_{id} for Valves and Orifices (Ref. 43)

Device	Exponent X	Pressure Range of Tests Psi
Orifice	0.19	49 to 200
Butterfly valve	0.18	39 to 145
Globe valve	0.11	30 to 100 psi

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A value of $X = 0.30$ for a globe valve was reported based on an early set of test data (Ref. 43). The reason for this discrepancy was investigated. This set of globe valve data was taken at low upstream pressures. The value of $\sigma_{id \text{ ref}}$ at the lowest upstream pressure was obtained with pressures on the discharge side being slightly negative. Observations through a view window documented that air came out of solution when there was a low downstream pressure. The free air cushioned the collapse of the cavities reducing the chance of damage. This caused the σ_{id} for the lowest pressure test to be too low compared to the σ_{id} values at higher pressures and resulted in an artificially high pressure scale effect exponent.

The primary reason why this is mentioned is to stress the influence of natural de-aeration on cavitation damage at low pressures and the possible errors that can be introduced in using such data. One must be careful using experimental data taken at low pressures when trying to determine the pressure scale effects exponent to predict damage at high pressures. A value of $\sigma_{id \text{ ref}}$ obtained from tests at low pressures will be smaller than expected. Using it to predict a σ_{id} at a higher pressure with the X values in Table 5.3 will result in an adjusted value that is too low and the valve will experience excessive cavitation damage. Or if the data are used to find the exponent, the exponent X will be too large.

5.4 SIZE SCALE EFFECTS FOR INCIPIENT AND CRITICAL

The original research work to evaluate size scale effects was also done using orifices (Refs. 35, 36). The results of this work were later checked against size scale effects data obtained on numerous types and sizes of valves up to 36-inch diameter, and just recently checked against data for a 96-inch butterfly valve. Research to date has found that size scale effects influence only incipient and critical levels of cavitation. The resulting equations recommended for making size scale effects adjustments (SSE) on σ_i and σ_c are:

$$\sigma_c = \text{SSE} (\sigma_{c \text{ ref}} - 1) + 1 \quad \text{or} \quad \sigma_i = \text{SSE} (\sigma_{i \text{ ref}} - 1) + 1 \quad (5.6)$$

$$\text{SSE} = \left(\frac{D}{d}\right)^Y \quad (D \text{ is limited to } \leq 36\text{-inches}) \quad (5.7)$$

$$Y = 0.3 K^{-0.25} \quad (5.8)$$

in which K is the valve loss coefficient (Eqn. 3.2), $\sigma_{c \text{ ref}}$ and $\sigma_{i \text{ ref}}$ are reference cavitation data for a valve of size d and σ_c and σ_i are the cavitation index values scaled to a valve of size D; which can be larger or smaller than the reference valve of size d. These equations are valid for orifices and valves.

Equations 5.6-5.8 were developed for valves up to 36" diameter. It has been the authors opinion that size scale effects on valves larger than 3-foot diameter would be over-predicted by these equations. This is based on the fact that, for many hydraulic modeling situations involving size or Reynolds number scale effects, there is a size or upper limit of Reynolds number beyond which the size scale effects become negligible. For example, recall the variation of the friction factor f with Reynolds number. At large Re, f becomes independent of Re.

The author was recently able to compare field test data at incipient cavitation on a 96-inch butterfly valve to laboratory tests on an 8-inch geometrically similar model butterfly valve. At full open the valve had a $C_d = 0.900$ and tests on the full size valve showed that it was experiencing light cavitation at a sigma of 39.0. Tests on the 8-inch model identified $\sigma_i = 25$ and $\sigma_c = 19$. The adjusted sigma values corrected for size scale effects from 8-inch to 96-inch with Equations 5.7 and 5.8 are: $\sigma_i = 74$ and $\sigma_c = 56$. These values are obviously much higher than the field test data. When the size scale effects correction was based on D = 36-inch diameter, as the upper size limit, the adjusted sigma values were $\sigma_i = 48$ and $\sigma_c = 37$. This agrees well with the observed $\sigma = 39$ at light cavitation in the 96-inch valve. It is, therefore, recommended that SSE adjustments be limited to a maximum valve size of 36-inches. This upper limit on the PSE adjustment is similar to the 300 psi upper limit suggested for the PSE adjustment.

Studies have been made by the authors former graduate students using orifice plates to identify if there are size scale effects at incipient damage (Refs. 35, 36, 25). Tests were made on orifices installed in three different pipe sizes. The analysis was

based on determining incipient damage based on a pitting rate of 1 pit/sq in/min. The study could not identify any size scale effects at incipient damage based on pitting rate. However, the research (Ref. 35) clearly showed that the size of the pits significantly increased with pipe diameter. Therefore, once material begins to be removed by cavitation, the larger systems will produce proportionally more weight loss than would be predicted from model tests. Since incipient damage is defined as the condition causing a pitting rate of 1 pit/min/sq in, which should correspond to a condition where there is essentially negligible damage no SSE correction is made to account for the increased size of the pits. Tests have also verified that there are no size or pressure scale effects at choking cavitation.

5.5 COMMENTS ON SCALE EFFECTS

In applying any of the scale effect equations to incipient or critical cavitation for elbows, orifices, or valves it is important to realize that the equations are inherently conservative. This is because of the way incipient and critical cavitation are determined. They are determined based on an absolute number of cavitation events rather than on events per unit area or per unit volume of flow.

Dynamic similarity would imply that the same percentage of nuclei would be cavitating in systems of different size. As a result, if one compared the cavitation conditions for a 3-inch valve and a 12-inch valve with the same percentage of nuclei per unit volume of water growing into vapor cavities and the velocity of flow were the same in both pipes, the larger system would be expected to have 16 times as many cavitation events per unit of time as the small one. Incipient or critical as defined in this guide requires both systems to operate at approximately the same number of events per unit of time. This forces the larger system to operate at a larger value of the cavitation index. This means that the incipient and critical cavitation conditions for different sized pipes will sound about the same intensity but for a larger system the potential for vibration and erosion damage is less because there were fewer events per unit volume of flow.

6.0 CAVITATION DATA

Application of the previously discussed method of cavitation analysis requires experimental data. Complete data are available only for a few valves. For most valves, data are available for only one or two of the six cavitation limits. This section provides representative data that can be used with the design/analysis procedure to evaluate the cavitation potential of valves in a given system.

6.1 BUTTERFLY VALVE DATA

Butterfly Valve

Figure 6.1 shows experimental data identifying the flow coefficients and the six cavitation limits for a 6-inch butterfly valve. Note that the cavitation data are correlated using C_d rather than valve opening. Using C_d is the recommended basis of comparison for different valves because it bases the comparison on the energy dissipating capability of the valve, not just on percent valve opening. The upper two plots allow the user to relate the C_d values in the sigma plot to valve opening and to C_v . They also allow the reader to compare the data with similar data for other butterfly valves to see how similar they are. If the valve under consideration has similar C_v and/or C_d curves, the cavitation data in Figure 6.1 should apply to the other valve.

The author has tested numerous styles of butterfly valves and found that the values of sigma at incipient, critical and choking cavitation are similar. Consequently, in the absence of cavitation data on a specific style of butterfly valve, the data in Figure 6.1 (except for the incipient damage limit) can be used as a source of reference data for use with the scale effects equations to estimate the performance of conventional butterfly valves. Data for incipient damage have only been collected for one butterfly valve.

Butterfly valves that have special disc designs for modifying their cavitation behavior will probably have significantly different flow coefficient curves and will not meet the criteria for using the data in Figure 6.1.

Figure 6.1 Reference Cavitation Data for Butterfly Valve

In contrast to the similarity of butterfly valves, there are significant differences in the cavitation behavior of other types of valves. The differences in their cavitation behavior will be documented by data presented later in this section.

Incipient damage data are presently available for only one butterfly valve so it is not possible to say if most butterfly valves will have similar sigma values at incipient damage. It is the authors opinion that there may be measurable differences in the damage characteristics of different styles of butterfly valves. This is especially true of damage that occurs near the valve seat. As a result, the incipient damage data in Figure 6.1 can only be used to approximate incipient damage for other butterfly valves.

To help select the appropriate limit for design and operation, it is informative to compare the flow rates (or velocities) corresponding to the various cavitation limits. This tells how much the velocity must be increased to obtain the next higher level of cavitation. Table 6.1 shows cavitation limits and the corresponding velocities in order of increasing cavitation intensity for the butterfly valve data in Figure 6.1. The data were calculated at a C_d value of 0.5. The velocity calculations are for an upstream pressure of $P_1 = 82.5$ psia, and $P_v = 0.5$ psia.

The numbers in the last column show the percent change in velocity required to increase the cavitation from the preceding limit. For example, a 20.9% flow increase is required to increase the cavitation from incipient to critical. This shows that for a butterfly valve there is a fairly wide range of flows where the cavitation varies from light and intermittent to light and steady. In contrast, the data also show that once the critical limit is exceeded, the intensity increased rapidly with increased flow. To change the cavitation from critical (where the intensity is only light and steady) to incipient damage where the noise and vibrations are significantly higher, requires that the flow be increased by 23.5%.

The data in Table 6.1 also show that cavitation damage begins before the butterfly valve starts to choke. Specifically, choking starts when the flow is only 12.7% beyond incipient damage. However, since the damage varies exponentially with velocity, the damage at incipient choking can be 230% greater than it is at incipient damage (based on an exponent of $n = 7$ in Eqn. 4.1, ie. damage = $1.127^7 = 2.31$, or 231% increase). This is pointed out to emphasize that the incipient choking (or the k_c point as defined in other literature) may not be a safe design point for some valves since it can result in a significant erosion damage potential.

The choking data for the butterfly valve show that the valve first starts to choke (σ_{ich}) at 37.2 fps (based on $C_d = 0.5$, $P_1 = 82.5$ psia, and $P_v = 0.5$ psia) and was fully choked at 40.8 fps. Note that the velocity does not increase once fully choked flow has been reached. At super cavitation, the flow is the same as at choking. Super cavitation is when the vapor cavity downstream from the valve extends beyond the downstream pressure tap so $P_d = P_v$ and sigma = 1.0.

Table 6.1 Cavitation Limits for a 6-inch Butterfly Valve

<u>Cavitation Limit</u>	<u>Sigma value</u>	<u>Velocity.fps</u>	<u>% Increase in Flow</u>
Incipient	8.32	22.1	
critical	5.70	26.7	20.9
incipient damage	3.74	33.0	23.5
incipient choking(k_c)	2.93	37.2	12.7
choked flow	2.44	40.8	9.6
maximum vibration	1.93	40.8	0.0
super cavitation	1.00	40.8	0.0

Note: Data scaled from Figure 6.1 at $C_d = 0.5$, $P_1 = 82.5$ psia, $P_v = 0.5$ psia)

For a butterfly valve data in Table 6.1, the maximum cavitation intensity occurs between fully choked flow and super cavitation. The maximum cavitation intensity point and the zone of super cavitation are not generally recommended as a design or operational limit. The primary reason is that the flow is very unstable. To explain this statement, first recall that super cavitation is the condition where the average pressure downstream from the valve is at vapor pressure for several pipe diameters, and the jet discharging from the valve is surrounded by a large vapor pocket. It is difficult to control the length of the super cavity, even in the laboratory. Laboratory observations of choking and super cavitation have documented that the flow is very unsteady. Only a small decrease in the downstream pressure can cause a large increase in the length of the super cavity. Conversely, a small increase in the pressure can cause the super cavity to disappear.

This phenomenon is similar to "column separation" which occurs during some transients and can cause high pressures. The other reason why the maximum intensity is not recommended as a design operating condition is that it may be the point of maximum erosion damage.

6.2 CONE VALVE DATA

Cone and Ball Valve

Figure 6.2 shows flow coefficient and cavitation data for incipient, critical, incipient damage and choking cavitation limits for a 6-inch full ported, skirted cone

Figure 6.2 Flow Coefficients and Cavitation Data for 6-inch Skirted Cone Valve

valve. The designation of the valve as a skirted cone valve means that the flow control element, which is a conical plug, is either solid or has a cylindrical skirt wrapped around the plug to give it the appearance of being solid. In small sizes the plug is frequently solid. For large valves making the conical plug solid is not practical.

Table 6.2 shows the percent increase in flow required to obtain the selected levels of cavitation for the same system conditions used for Table 6.1.

Table 6.2 Cavitation Limits for a 6-inch Cone Valve

<u>Cavitation Limit</u>	<u>Sigma value</u>	<u>Velocity.fps</u>	<u>% Increase in Flow</u>
Incipient	5.82	26.4	
critical	4.22	31.0	17.5
incipient damage	3.40	34.6	11.6
choked flow	1.87	46.7	35.0

Note: Data scaled from Figure 6.2 at $C_d = 0.5$, $P_1 = 82.5$ psia and $P_v = 0.5$ psia.

The cone valve is designed with two throttling ports in series, one at the inlet to the cone and one at the outlet. Generally, the two ports are the same size. With the pressure drop being divided between the two throttling ports (in series), the overall cavitation performance should be better than valves with a single stage of throttling, like a butterfly valve which has two throttling ports in parallel. The cavitation performance of the cone valve can be compared with the butterfly valve using the data in Table 6.3. The data for both valves are for the same valve size and the data are calculated at the same upstream pressure and at the same value of the discharge coefficient.

Table 6.3 Comparison of the Cavitation Performance of Butterfly and Cone Valves

<u>Limit</u>	<u>Butterfly Valve</u>		<u>Cone Valve</u>		<u>% Diff in Vel</u>
	<u>Sigma</u>	<u>Vel.fps</u>	<u>Sigma</u>	<u>Vel.fps</u>	
Incipient	8.32	22.1	5.82	26.4	19.5
critical	5.70	26.7	4.22	31.0	16.1
inc.damage	3.74	33.0	3.40	34.6	4.90
choked flow	2.44	40.8	1.87	46.7	14.5

Note: Data taken from Tables 6.1 and 6.2 at $C_d = 0.5$, $P_1 = 82.5$ psia and $P_v = 0.5$ psia

In making the comparison between different types of valves, it is more informative to make the comparison based on flows rather than sigma values. For example, the difference between σ_c for the two valves is 26%. This suggests that the cone valve may have significantly better cavitation performance than the butterfly. However, if the comparison is based on flow, the difference between the corresponding flow rates at critical cavitation is only 16.1%.

If the comparison between the two valves is based on incipient damage, the cone valve can only provide 4.9% more flow compared to the butterfly valve. Using cavitation as the basis for valve selection, this small gain would not offset the cost difference between the valves. There are, of course, other considerations besides cavitation to consider when selecting a valve type.

Data are presented in Figure 6.3 which compare the critical cavitation limits for a 24-inch skirted cone valve with that for a 24-inch skeleton cone valve. Note that the skeleton design has considerably larger values of σ_c (meaning that it has poorer cavitation performance). The skeleton design allows considerable flow to go around the plug rather than passing through the main ports. This is also reflected in the larger value of the flow coefficients for the skeleton style valve. The flow around the plug reduces the effectiveness of the cone valve to that of a single stage of throttling, more like the butterfly valve. This is one example of how differences in valve design have significant effects on its cavitation performance.

Size and pressure scale effects on critical cavitation can be demonstrated by comparing the 6-inch and 24-inch data for the skirted cone valves. This comparison is made in Figure 6.4 and shows that the larger valve has larger σ_c values, meaning that it cavitates easier. The data for the 6-inch valve was taken at a lower pressure ($P_1 = 52$ psia) than the 24-inch valve ($P_1 = 165$ psia). The figure also shows the effect of adjusting the data for the 6-inch valve for size and pressure scale effects. The data identified as "6-inch scaled" has been adjusted for SSE and PSE using Equations. 5.2-5.8. The adjusted data does not exactly fit the 24-inch data but the valves were not identical and the testing was done by two organizations using slightly different testing procedures. In light of these limitations the agreement between the 6-inch scaled data and that of the 24-inch valve is acceptable.

Figure 6.3 Flow Coefficients and Cavitation Data for 24-inch Cone Valves

Figure 6.4 Reference Cavitation Data for 6 and 24-inch Cone Valves

6.3 GLOBE VALVE DATA

Globe Valve

Of all valves, the globe style probably has the largest variations in the design of both the body and the configuration of the control element. Consequently the capacity and cavitation performance will be vastly different. It is not possible to provide a typical set of data describing a globe valve as was done for butterfly and cone valves. Data are presented to show the range of the various cavitation limits. Even though there is a significant amount of data available for globe style valves, no valve has been tested to evaluate all six cavitation limits.

Figure 6.5 shows discharge coefficient and critical cavitation data for 13 different globe style valves. All data were corrected for size and pressure effects. The data were scaled to 6-inch diameter and a common upstream pressure of $P_1 = 87$ psia.

There are significant variations in the designs of the valves but all have a single stage of throttling. The differences in the hydraulic performance of the valves is demonstrated by the large variations in the C_d versus valve opening curves. For the cone and butterfly valves the cavitation limits correlated quite well with C_d . For the globe valves there does not seem to be any consistent correlation. For example the valve with the highest C_d curve also has the smallest values of σ_c (at large C_d values). In contrast, the 16-inch valve (shown with the diamond symbols) has a large C_d full open but has the highest σ_c values meaning that it has the worst cavitation performance. Several items are evident from the flow and critical cavitation data presented in Figure 6.5:

1. To accurately assess the cavitation potential of a globe valve, experimental data are needed on each valve being analyzed.
2. If specific cavitation data are not available for the critical cavitation level, one conservative approach would be to select data from a curve representing the upper envelope of the data in Figure 6.5.
3. Correlating critical cavitation for globe valves using C_d does not produce a unique relationship that can be used to predict the performance of other valves.
4. The flow characteristics of a globe valve can be custom designed by altering the shape of the control element. For example, the 6-inch 1500# valve is a quick opening design and the valve labeled 8-inch parab provides a relative linear opening characteristic. The actual flow characteristic of the valve is dependent on the system in which it is installed.

The cavitation performance of globe valves can be significantly improved by installing special cavitation trim. The principles used to design the special cavitation trim are: 1) divide the flow into numerous small jets, 2) design the orifice holes so no cavitation occurs inside the hole, 3) have the jets directed so they do not impact against the valve body, 4) enlarge the valve body to remove the cavitation collapse from the boundary and 5) for applications requiring high pressure drops, design the trim with multiple losses in series. These principles are discussed in more detail and demonstrated with examples in Section 7.

Critical Sigma Values Scaled to 6-inch Valve Diameter

Figure 6.5 Flow Coefficients and Critical Cavitation Data for globe Valves

Figure 6.6 demonstrates the difference that cavitation trim can make on the critical cavitation limits. The data were obtained using a 4-inch globe valve body and installing four different sets of internals. Trim #4 consisted of a relatively standard plug design with a single stage of throttling. The other trims had multiple stages of

dissipation. The most dramatic improvement is with Trim #3. This trim had several stages of throttling with numerous flow passages in parallel. Each flow passage had essentially the same resistance. The C_d is increased by exposing more passages as the plug is opened. No cavitation could be detected at any valve opening pressure differentials up to almost 200 psid with about 1 psi downstream.

The lowest possible value of the cavitation index is 1.0 corresponding to super cavitation where the downstream pressure = vapor pressure. The lowest test point for the test on trim #3 was $\sigma = 1.07$ and no cavitation noise could be detected. The multiple losses in series make it possible for the valve to take large pressure drops without cavitating.

Because each flow passage in the Trim #3 valve had the same resistance, the valve had the same cavitation performance at all valve opening. This is verified in Figure 6.6 which shows that the critical sigma value is constant at 1.07. It actually may be a little misleading to show a critical cavitation index for Trim #3 since there was essentially no cavitation. It is shown to indicate that the valve does not have any significant cavitation until it is almost choking.

One of the compromises to this type of trim is the low C_d values. Note that the C_d full open was 0.025 ($C_v = 12$). This is solved by increasing the stroke of the valve which increased the number of parallel flow passages or by having large single stage holes near the full open position. Another option is to reduce the number of throttling stages for the flow passages at larger valve openings. This is appropriate when the required pressure drop across the valve reduces due to increased system losses at higher flow rates.

There are many variations in the type of trim that can be installed in a globe valve body. This allows valves to be custom designed to satisfy almost any cavitation need.

Only limited data are available identifying incipient damage for globe valves. Figure 6.7 presents data for three valves with flow from under the seat (normal flow direction). The difference in the two 6-inch valves is that the body diameter of the

valve designated 6-inch was larger than the body of the valve designated 6-inch new. The 3-inch x 2-inch valve had a body designed for a normal 3-inch valve but it had a seat and plug designed for a 2-inch valve.

Figure 6.6 Reference Cavitation Data for Globe Valves

For the 3-inch x 2-inch valve $\sigma_{id} \sim 1.0$ at the two smallest valve openings. This reflects the fact that no damage could be detected until the sigma was about 1.0 (which corresponds to fully choked flow). The tests were done at pressure differentials up to 200 psi and zero pressure downstream. The reason for this is that with the valve body being oversized and the valve opening small, the jet is dissipated before it can penetrate to the boundary and cause damage. The valve cannot choke because there is insufficient momentum in the jet at small openings. The benefit of over sizing the valve body is improvement in the cavitation. The disadvantage is the loss in flow capacity as shown by the smaller C_d values in Figure 6.7 compared to the two 6-inch valves.

For the two 6-inch valves, at small valve openings (small C_d values) there is little effect of body size on the cavitation limits. Both globe valves bodies are large enough to produce relatively good cavitation performance when the jet discharging from the seat is small compared to the valve body. At large openings (large C_d values), the larger body style has significantly better cavitation performance. This is another example of how changes in the valve design causes significant differences in its cavitation performance.

Data for choking cavitation for five different globe valves are shown in Figure 6.8 with their corresponding flow coefficients. These data can be used to approximate choked cavitation for single stage globe valves. Critical cavitation data for these same valves is presented in Figure 6.5. One interesting point is that for the 3-inch x 2-inch valve in Figures 6.7 and 6.8, σ_{id} is lower than σ_{ch} at $C_d = 0.1$. This reflects the fact that this valve chokes before it causes damage, at that valve opening.

Multi-stage globe valves do not exhibit typical choked behavior. The reason for this was previously discussed in Section 4.3.2. In brief, only the last stage of the valve can choke and since the last stage is designed to produce only a small fraction of the total required pressure drop, choking has little influence on the overall valve performance.

Figure 6.9 shows the most complete data for a globe valve. It identifies four of the six cavitation limits. This valve is the 3-inch x 2-inch valve discussed previously.

Figure 6.7 Flow Coefficient and Incipient Damage Data for Globe Valves

Figure 6.8 Flow Coefficient and Choking Data for Globe Valves

Figure 6.9 Flow Coefficient and Cavitation Data for 3-inch Globe Valve

6.4 ORIFICE CAVITATION DATA

Cavitation and C_d data for circular, concentric, sharp edge, and thin orifices plates are listed in Table 6.4 and shown graphically in Figures 6.10 and 6.11. The requirement of the orifice being thin means that the thickness of the plate is small compared to the diameter of the hole so the jet springs free and does not re-attach to the wall of the hole. Usually the back side of the plate is relieved at a 45 degree angle to further reduce the thickness of the plate at the lip of the hole. If the plate is about as thick as the hole diameter and is not chamfered on the back side, the jet can reattach, cause loud whistling, change the flow characteristics and cause cavitation damage inside the hole.

The same equations that are used for valves to make scale effects corrections can be used to make pressure scale effects corrections for orifices. Equations 5.2-5.5 are for pressure scale effects and Equations 5.6-5.8 are for size scale effects. Note that for orifices there are only size effects at incipient and critical cavitation limits and pressure scale effects only for incipient damage.

Table 6.4 Cavitation Limits for Orifices (Ref. 43)

d_o/d	A/A_o	A/A_j	C_d	σ_i	σ_c	σ_{id}	σ_{ch}
1	2	3	4	5	6	7	8
0.389	6.60	10.5	0.100	2.10	1.96	1.45	1.27
0.444	5.06	8.00	0.133	2.30	2.00	1.67	1.32
0.500	4.00	6.22	0.179	2.62	2.20	1.83	1.39
0.667	2.25	3.31	0.385	4.38	3.16	2.73	1.74
0.800	1.56	2.17	0.648	7.62	4.89	4.19	2.78

- Notes:
1. Pipe diameter, $d = 3$ -inch, $P_{1\text{ ref}} = 102$ psia, $P_{v\text{ ref}} = 0.17$ psia
 2. A is the area of the pipe and A_o is the area of the orifice hole and A_j is the area of the contracted jet.
 3. d is the pipe diameter and d_o is the diameter of the orifice hole
 4. The C_d is based on the net pressure drop and should not be confused with the orifice coefficient published by ASME (Ref. 1) for orifices used as flow meters

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To simplify the use of the data, like that in Figure 6.10 and especially to use the data in spreadsheet applications or other computer programs to solve cavitation problems, it is convenient to fit exponential equations to the data. The following equations can be used for this purpose.

$$\sigma_i = 1.55 + 4.88 C_d + 5.66 C_d^2 + 1.95 C_d^3 \quad (6.1)$$

$$\sigma_c = 1.38 + 4.97 C_d - 3.24 C_d^2 + 6.06 C_d^3 \quad (6.2)$$

$$\sigma_{id} = 1.25 + 2.70 C_d - 3.15 C_d^2 + 0.493 C_d^3 \quad (6.3)$$

$$\sigma_{ch} = 1.08 + 2.28 C_d - 4.38 C_d^2 + 7.57 C_d^3 \quad (6.4)$$

$$C_d = 0.019 + 0.083 \beta - 0.203 \beta^2 + 1.35 \beta^3 \quad (6.5)$$

$$\beta = d_o/d$$

$$\beta = 0.193 + 2.34 C_d - 3.94 C_d^2 + 2.73 C_d^3 \quad (6.6)$$

Figure 6.10 Cavitation Limits for 3-inch Orifices.

Figure 6.11 Discharge Coefficient versus Diameter Ratio for Orifices.

6.4.1 Zones of Cavitation Damage In a situation where cavitation damage will occur, it is important to know the extent of the damage region. If the orifice is, by definition, a "thin plate orifice" no damage will occur on the orifice plate. The flow separates from the lip of the orifice and all the cavitation occurs downstream. If the orifice is a "long orifice" where the plate is thick compared to the hole diameter, the jet can re-attach to the wall of the hole and cavitation damage can occur inside the orifice hole. All data in this section are restricted to "thin plate" orifices.

Experimental results are available that identify the region downstream from an orifice where the damage will occur. The data show that damage occurs farther from the orifice as the value of d_o/d ratio decreases and as the sigma reduces. This is true for the upstream and downstream limits of pitting as well as the zone of maximum damage. Figure 6.12 shows data for a 2.4-inch orifice installed in a 3-inch diameter pipe. This is the orifice listed in Table 6.4 at $d_o/d = 0.80$. Note that damage begins at $\sigma_{id} = 4.19$ and the orifice chokes at $\sigma_{ch} = 2.78$.

When pitting density was plotted versus distance, as shown in Figure 6.12, it is seen that the upstream location of cavitation damage for a particular orifice remains almost stationary for all levels of cavitation. For the orifice used to obtain the test data in Figure 6.12, the upstream location of the damage is about 0.4 pipe diameters from the orifice. The location of maximum pitting and the downstream location moved downstream as the damage rate increased. Up to the point where choking begins, the damage is

as the damage rate increased. Up to the point where choking begins, the damage is restricted to the first three pipe diameters. The data also show that there is a dramatic increase in the pitting rate as the sigma is reduced and that the zone of damage lengthens with reduced sigma. When the orifice is fully choked (super cavitation), the damage will occur far downstream from the orifice.

Figure 6.12 Fitting Distribution Downstream from Orifice (Ref. 43)

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6.5 NOZZLES AND SUDDEN ENLARGEMENTS

The performance of nozzles and sudden enlargements can be predicted from orifice data. The incipient, moderate, and severe cavitation levels for orifices and nozzles have been compared (Ref. 3). Contraction coefficients for free flow were applied to orifice data and a cavitation index number plotted for each of the three cavitation levels versus the jet-enlargement area ratios. By assuming that the contraction coefficients for submerged orifice flow were essentially the same as for free discharge flow, good agreement was found between orifice and nozzle cavitation data.

Contraction coefficients C_c for circular, concentric sharp edged orifices have been calculated (Ref. 27 p. 57). C_c is the ratio of the contracted area of the jet to the area of the orifice. The values are for two dimensional jets but experimental data have verified their accuracy for three dimensional flow. The data were fit to a polynomial equation valid for $0 < d_o/d < 0.8$. The resulting equation is:

$$C_c = 0.611 + 0.024\left(\frac{d_o}{d}\right) - 0.013\left(\frac{d_o}{d}\right)^2 + 0.196\left(\frac{d_o}{d}\right)^3 \quad (6.7)$$

in which C_c is the ratio of the contracted jet area to the pipe area.

The cavitation characteristics of orifices, nozzles and sudden enlargements were compared based on the effective jet diameter at the vena contracta (Ref. 36). It was concluded that for all practical purposes, the cavitation data obtained from tests on orifices could be applied to sudden enlargement energy dissipators formed by nozzles or smaller pipes discharging into larger pipes. One must be careful to make the comparison based on the jet area and not the orifice area.

The C_d value for a nozzle or sudden expansion can be estimated using data for single hole orifices or nozzles provided the area ratio is based on the minimum area of the contracted jet.

Incipient cavitation for nozzles and pipe expansions computed using orifice data may deviate slightly from orifice data (Refs. 28, 29). This study showed that cavitation is not only effected by the geometry of the jet, but also by the entrance conditions of the expansion. The development of the boundary layer, intensity of the eddies, and therefore, the cavitating conditions are dependent upon the velocity gradient around the jet as it enters the expansion. This gradient is steep for orifices, but becomes more gradual for nozzles or pipes discharging into sudden enlargements because a boundary layer develops. This results in a conservative prediction of cavitation levels for nozzles from orifice data.

6.6 MULTI-HOLE ORIFICES

The concept of using a multiple-hole orifice plate device has been used in several applications for creating pressure reductions and reducing noise. It is used in many of the newer type high pressure drop devices (Ref. 44) and has been used in pressure reducing devices for circulating water systems of nuclear power plants (Ref. 41). There are numerous other examples of where the multi-hole orifice plates or their equivalent have been used in hydraulic systems (Ref. 44). A three hole orifice was used in a large outlet tunnel from a dam in Canada (Ref. 30).

The multi-hole plate offers several advantages and some disadvantages. The most obvious disadvantage is potential plugging. For it to be used in a system, the fluid must be free of any sizeable suspended material. Some of the advantages are that they produce less noise and vibrations, they can be spaced close together in the event that orifices in series are required, and cavitation damage, when it occurs, is restricted to a shorter distance downstream from the orifice plate.

To use a multi-hole orifice it is necessary to be able to calculate its C_d and obtain cavitation data. Limited experimental data show that the multi-hole orifice discharge coefficient appears not to be a function of the number, spacing, or diameter of the holes but only of the ratio of the total areas of all holes to the pipe area. This makes it possible to use the C_d versus d/d_o data for single hole orifices in Table 6.4 to design multi-hole orifice plates.

One significant difference between the single hole and multi-hole orifice is choking cavitation. The multi-hole orifice begins to choke near the center of the cluster

of holes. As the downstream pressure drops, more holes choke until eventually all holes choke (except when the discharge coefficient is very small). This means that there is a measurable difference between the sigma value at incipient choking and choked flow for the multi-hole orifice.

6.7 VALVES AND ORIFICES IN SERIES

One of the most effective means of producing large pressure drops without excessive cavitation is to use valves and/or orifices in series. Valves in series gives optimum performance (compared to orifices) because each valve can be adjusted to operate at a selected level of cavitation. The advantage of orifice plates in series is that they are much less expensive. The disadvantage is that they are a fixed resistance and all control of flow must be provided by the valve. If a wide range of flow control is required, a single valve in series with orifice plates will result in the valve cavitation at smaller flow rates. When this combination is used, the valve should be placed upstream from the orifices so it has the maximum downstream pressure.

The overall loss and discharge coefficient for orifices (or valves) placed in series can be calculated by the equation:

$$K = K_1 + K_2 + K_3 + \dots \quad (6.8)$$

$$C_d = \sqrt{1 + \frac{1}{C_{d1}^2} + \frac{1}{C_{d2}^2} + \dots + \frac{1}{C_{dn}^2}} \cdot N \quad (6.9)$$

in which K_1 and C_{d1} is the loss and discharge coefficient for the first orifice plate, K_2 and C_{d2} for the second orifice plate, etc. and N is the total number of orifices.

One of the important factors in the design of series valves and orifices is the proper spacing. There are at least three criteria that governs the spacing. The first is to space the orifices far enough apart so they produce essentially their full pressure drop. When single hole orifices are placed too close together, the jet from the upstream orifice will go through the second orifice before the jet is fully dissipated and before the pressure is recovered. This makes the actual pressure drop significantly less than the calculated pressure drop. If this is allowed, the orifices will not perform as desired and Equation 6.8 does not apply.

Past research has shown that the pressure downstream from a multi-hole orifice fully recovers within 3 to 4 pipe diameters. This means that the orifice plates can be spaced approximately 3 pipe diameters apart and still produce essentially 100 percent of their normal head loss. It is possible to space them closer but an estimate of the reduction in the pressure drop would be necessary. Single hole orifices must be spaced at least six pipe diameters apart. The only way to space them closer is to offset the holes so that the jet is not directed at the next hole. This only works for orifices with relatively small holes.

In an experimental program, three different combinations of three plates placed in series were calibrated to determine if the combined C_d of each set of plates was effected by hydraulic interference caused by the close spacing. The orifice plates were installed 2.5 pipe diameters apart. The three sets of data showed that for a spacing of 2.5 diameters the combined discharge coefficient was approximately 5 percent larger than that calculated by Equation 6.8. This is because the pressure was not completely recovered between the orifices. If desired, the proper C_d could be obtained by reducing the number of holes.

The second constraint is their cavitation performance. If the plates are spaced too close together, the cavitation generated at the upstream orifice can cause cavitation damage to the downstream valve or orifice. Another related problem of close spacing is that the turbulence generated at the upstream orifice causes the downstream orifice to cavitate easier.

Results from two experimental studies are available identifying the cavitation characteristics of multi-hole orifice plates. One study (Ref. 41) was carried out with multi-hole orifice plates with test conditions where the pressure downstream was negative. These tests were for only one plate. No tests were done with plates in series. For those tests, large amounts of air came out of solution because of the low downstream pressure and high turbulence generated by the jets. This air suppressed the cavitation and prevented cavitation damage for all tests, even fully choked flow. It is anticipated that the same multi-hole orifice plate operating at positive pressures would suffer cavitation damage and have cavitation characteristics slightly worse than that of a single hole orifice plate.

Another study was made of cavitation damage caused by a multi-hole orifice (Ref. 23). The orifice had 16, 1/2 inch diameter sharp edged holes with an equivalent orifice area of 3.14 in². The orifice was mounted in a 6-inch diameter pipe and had a measured C_d of 0.075. Tests were conducted at $P_1 = 102$ psia and incipient damage occurred at $\sigma_{1d} = 0.41$. The test results were compared to a 6 x 2 inch orifice, $C_d = 0.071$ (same area ratio as the orifice with 16 holes 1/2-inch diameter) where incipient damage occurred at $\sigma_{1d} = 0.30$. It was concluded that damage occurred at a larger sigma value for the multi-hole orifice because the outside row of holes is closer to the pipe wall than the single jet. It was also observed that the zone of cavitation damage on the pipe wall was restricted to a short distance from the orifice rather than extending several pipe diameters downstream like the single hole orifice.

A third concern with close spacing is the possibility of the upstream valve causing the downstream valve disc to flutter. The author recalls one installation where two butterfly valves were placed about one diameter apart. The upstream valve was used for control and the other valve served as a block or isolation valve. The operator on the isolation valve continually vibrated loose because of flutter of the disc caused by the high velocity jet from the upstream valve. For single hole orifices, butterfly, cone, ball and gate valves, it is necessary to have between 5 and 8 pipe diameters between them to allow full pressure recovery. In certain situations, globe, cone and other types of valves

can flange to flange but the overall discharge coefficient for the series valves cannot be calculated by Equation 6.8 since no pressure recovery occurs between the valves. When the valves are close together, the level of cavitation should be limited to something considerably less than incipient damage.

Single hole orifices should be spaced about 6 to 8 diameters apart to obtain full energy dissipation and pressure recovery, and have them operate essentially independent of the orifice located just upstream.

If space is a problem, multi-hole orifices can be used. Figure 6.13 shows how the pressure recovers downstream from multi-hole orifice plates. The data show that the pressure is fully recovered in about 3 pipe diameters. This means that for situations where it is necessary to install multiple plates in series, they can be spaced 3 diameters apart and obtain full pressure recovery. This means that each orifice plate will operate at its normal discharge coefficient and Equation 6.8 can be used to calculate the overall coefficient. If the plates are spaced closer than 3 pipe diameters, they will not produce their full pressure drop and the number of holes will need to be reduced. If the plates are spaced less than 1 pipe diameter apart, it is recommended that they not be designed for incipient damage. Cavitation from the upstream plate can damage the downstream plate.

Figure 6.13 Pressure Recovery for Multi-hole Orifice in 20-inch Pipe

Application of the data presented in this section and the scale effects corrections discussed in Section 5 will be demonstrated with several examples.

Example 6.1 A 6-inch butterfly valve is operating under the following conditions: $P_U = 80.8$ psig, $P_D = 37.6$ psig, $P_V = 1.16$ psia, $Q = 1.29$ cfs, $P_b = 12.2$ psia, $\gamma = 62.4$ lb/cu-ft. Determine if the valve is cavitating.

The procedure will involve comparing the value of σ_{sys} with the value of σ at incipient, critical, incipient damage etc. from Figure 6.1. To get the σ values to compare with σ_{sys} requires knowing C_d , scaling the σ values scaled from the figure and adjusting the sigmas for pressure scale effects.

Calculate C_d

$$P_U - P_D = 80.8 - 37.6 = 43.2 \text{ psi}$$

or

$$43.2 \cdot \frac{144}{62.4} = 99.7 \text{ ft of head}$$

$$V = \frac{Q}{A} = \frac{1.29}{0.196} = 6.58 \text{ fps}$$

$$C_d = \frac{6.58}{\sqrt{(2 \cdot 32.2 \cdot 99.7 + 6.58^2)}} = 0.082$$

Calculate σ_{sys}

$$\sigma_{sys} = \frac{(80.8 + 12.2 - 1.16)}{43.2} = 2.13$$

From Figure 6.1 at $C_d = 0.082$, $\sigma_{c \text{ ref}} = 2.45$ and $\sigma_{id \text{ ref}} = 1.85$, for $P_{1 \text{ ref}} = 82$ psi, and $P_{V \text{ ref}} = 0.2$ psia.

Since σ_{sys} is between $\sigma_{c \text{ ref}}$ and $\sigma_{id \text{ ref}}$ the valve is cavitating. The next step is to make pressure scale effects adjustments to $\sigma_{id \text{ ref}}$. There is really no need to make scale effects adjustments to critical cavitation since it is clear that the cavitation level is beyond critical and making pressure scale adjustments will only verify that fact.

Adjust $\sigma_{id \text{ ref}}$ for pressure scale effects for a butterfly valve with $X = 0.18$ (Table 5.3)

$$PSE = \left(\frac{(80.8 + 12.2 - 1.16)}{82.0 - 0.2} \right)^{0.18} = 1.02$$

$$\sigma = 1.02 \cdot 1.85 = 1.89$$

Since σ_{sys} is slightly greater than σ_{ld} , the cavitation intensity is just under incipient damage.

Example 6.2 An orifice with $d_o/d = 0.47$ is to be installed in a 3.0-inch pipe where $P_u = 98.6$ psi, $P_b = 12.36$ psi, $P_v = 0.18$ psi. Calculate the allowable pressure drop and flow if the orifice is to operate at critical cavitation.

Using Figure 6.11 or Equation 6.5, find $C_d = 0.154$ and from Figure 6.10 or Equation 6.2 get $\sigma_c = 2.10$ (remember that there are no pressure scale effects for orifices at critical cavitation). Since the pipe is the same diameter as the reference cavitation data, there are no size scale effects.

Calculate ΔP allowable from

$$\Delta P = \frac{(P_u + P_b - P_v)}{\sigma_c}$$

$$\Delta P = \frac{(98.6 + 12.36 - 0.18)}{2.10} = 52.75 \text{ psi}$$

Calculate the flow by first finding the velocity from Equation 3.2, where $K = 1/C_d^2 - 1 = 41.17$ using Equation 3.2

$$V = \sqrt{\frac{(2 \Delta P)}{(\rho K)}}$$

$$V = \sqrt{\frac{(2 * 52.75 * 144)}{41.17 * 62.4 / 32.2}} = 13.8 \text{ fps}$$

$$Q = A V = 0.0491 \text{ sq-ft} * 13.8 \text{ fps} = 0.677 \text{ cfs}$$

Example 6.3 An orifice with a hole diameter of 9.38-inches is installed in a 15.25-inch (inside diameter) pipe. It will operate at a constant upstream pressure of 175.4 psig, $P_v = 1.74$ psia, and $P_b = 13.9$ psia. Calculate the allowable pressure drop and flow if the orifice is to operate at critical cavitation.

Note that the procedure is similar to Example 6.2 with the addition of the need to make size scale effects corrections. First find the critical cavitation

$$\frac{d_o}{d} = \frac{9.38}{15.25} = 0.615 \quad C_d = 0.307 \quad \text{Eqn. 6.5 or Fig. 6.11)}$$

$\sigma_{c \text{ ref}} = 2.74$ (for $d = 3$ -inch, Eqn. 6.2 or Fig. 6.10)

at reference pressures of: $P_{1 \text{ ref}} = 102$ psia, $P_{v \text{ ref}} = 0.17$ psia

Use Equations 5.6-5.8 for size scale effects with $D = 15.25$, $d = 3$

$$Y = 0.3 \left(\frac{1}{0.307^2} - 1 \right)^{-0.25} = 0.17$$

$$SSE = \left(\frac{15.25}{3} \right)^{0.17} = 1.32$$

Calculate the allowable pressure drop and flow if the orifice is to operate at critical cavitation.

Calculate ΔP allowable from

$$\Delta P = \frac{(P_u + P_b - P_v)}{\sigma_c}$$

$$\sigma_c = SSE (\sigma_{c \text{ ref}} - 1) + 1 = 1.32 (1.74) + 1 = 3.30$$

$$\Delta P = \frac{(175.4 + 13.9 - 1.74)}{3.30} = 56.84 \text{ psi}$$

Calculate the flow by first finding the velocity from Equation 3.2, where $K = 1/C_d^2 - 1 = 9.61$. Using Equation 3.2

$$V = \sqrt{\frac{2 \Delta P}{\rho K}}$$

$$V = \sqrt{\frac{(2 * 56.84 * 144)}{9.61 * 62.4/32.2}} = 29.6 \text{ fps}$$

$$Q = A V = 1.38 \text{ sq-ft} * 29.6 \text{ fps} = 40.9 \text{ cfs}$$

Example 6.4 A 6-inch butterfly valve is operating at 67.5% open with $P_u = 75$ psig, $P_d = 25$ psig, $P_v = 0.43$, $SG = 1.0$ and $P_b = 14$ psia. Determine if the valve is cavitating and calculate the flow. The process involves: finding the C_d at 50% open from Figure 6.1, calculating σ_{sys} , finding the cavitation limits from Figure 6.1 and comparing σ_{sys} to the different limits to decide if the valve is cavitating. Depending on the results, it may be necessary to make scale effects adjustments.

From Figure 6.1, $C_d = 0.60$ $P_1 = 75 + 14 = 89$ psia

$$\sigma_{sys} = (89 - 0.43)/(75 - 25) = 1.77$$

From Figure 6.1 the cavitation limits are:

$$\sigma_l = 10.2$$

$$\sigma_c = 6.6$$

$$\sigma_{ld} = 4.3$$

$$\sigma_{lch} = 3.5$$

$$\sigma_{ch} = 2.9$$

Since σ_{sys} is less than σ_{ch} the valve is choking. No scale effects corrections are needed because there are none at choking. To calculate the flow, it is necessary to first determine ΔP_{ch} and then use either the C_d or C_v equation. ΔP_{ch} is calculated from σ_{ch} , $\Delta P_{ch} = (89 - 0.43)/2.9 = 30.54$ psi.

From Equation 3.10 or Figure 6.1 $C_v = 805$, so $Q = 805 / (30.54)^{0.5} = 145$ gpm.

Note that because the valve is choking, the actual pressure differential of 50 psi cannot be used in the C_v equation to calculate the flow. Doing so would result in a flow that is higher than the actual.

7.0 CONTROLLING CAVITATION

The previous sections provided information on the cavitation performance of various type of valves. Examples were given to demonstrate how to apply the data to determine the intensity of cavitation that a valve will be subjected to when operating at a given set of system conditions. If such an analysis indicates that the valve, orifice, or other device will be operating at a cavitation level greater than can be tolerated, the design or system must be altered. This section provides information on various techniques which can be used to design a new system or modify an existing one so it can operate at the desired level of cavitation. The various techniques will be described and examples provided.

7.1 VALVE STYLE

By examining the cavitation data provided in Section 6, it can be seen that there are significant differences in the cavitation performance of different types of valves. There can also be significant variation in the performance of valves of the same type due to differences in design of the flow passages. This was demonstrated in Figure 6.3 by comparing the data for 24-inch skirted and skeleton cone valves. The skirted cone is designed so that essentially all of the flow goes through the main ports. For the skeleton design, the outside of the cone is not solid and a significant amount of flow goes between the cone and the valve body. Figure 6.3 shows that the σ_c for the skeleton cone valve is almost double that of the solid cone valve, meaning that it cavitates much easier.

A similar comparison was made between the globe valves in Figure 6.5 which shows large differences in the cavitation performance of the different valves.

The point that is being made with these two examples is that there can be significant differences in the cavitation performance of some styles of valves and one should request cavitation data from the various manufacturers on each specific valve. Do not assume that the cavitation performance of all valves of the same style is the same. One exception to this is the butterfly valve. Tests on dozens of conventional butterfly valves by the author and his colleagues have shown that they have similar values of sigma at incipient, critical and choking. Not enough data are available to compare the valves at incipient damage and maximum noise level but it is likely that these limits will also be similar.

If using conventional valves such as butterfly, cone, ball, or globe valves still produces excessive cavitation, it may be advisable to look at new valve styles developed in recent years. There are a number of valves on the market which can produce high pressure drops under controlled cavitation conditions.

The new cavitation control valves are generally a variation of the globe style valve with special cavitation trim. These valves operate generally on one of two principles. One style of valve uses the principle of flow through an orifice discharging into a sudden enlargement. This is one of the best arrangements for suppressing

cavitation for a single stage of dissipation. There are two reasons for this. One is that a well defined and large separation region minimizes the shear stresses, and therefore, reduces the chance of the cavitation. The second reason is that with a large separation zone, the cavitation occurs in the fluid, away from solid boundaries. This allows significant cavitation to exist before damage occurs at the boundary. These valves typically divide the flow into many small jets in parallel formed by trim containing a large number of orifices.

There are several valves on the market which use this principle (Ref. 44). The trim is usually cylindrical in shape with multiple holes. Some are designed so the jets discharge internally and impinge on each other near the center of the valve body. Other styles have the jets discharging outward and toward the outside valve boundary. Dividing the flow into many small jets reduces the noise and vibrations compared to a valve with a single throttling port.

The other style of cavitation control valve contains a large number of flow paths in parallel, each containing multiple constrictions in series. As the valve plug is opened, more and more flow passages are exposed. These valves can be designed so that each flow passage has the same loss characteristics or designed so the resistance of the flow paths decrease as the valve opens. If the valve is designed with each flow passage having the same loss characteristics, the valve will have the same cavitation performance at all valve openings. This is different from conventional valves (like butterfly, cone, globe, etc. as shown in Figures 6.1-6.5) where the cavitation index values increase significantly with valve opening.

The required pressure drop across most control valves usually decreases as the flow increases due to increased system losses. Therefore, it is common to reduce the loss across the flow passages at larger valve openings. Some styles use large, single stage orifice holes in place of the more restrictive multiple losses in series for the last part of the valve stroke. This greatly increases the flow capacity of the valve when it is full open.

The operation of a multi-stage valve can be best visualized by comparing it to a number of orifice plates in series. By properly dividing the pressure drop between the orifice plates, one can increase the allowable pressure drop almost indefinitely and still control the cavitation. The design procedure for this type of valve is outlined in the next section. These multi-stage valves can be designed to operate at sigma values at or near to choking with little noise and vibrations.

7.2 ORIFICES AND VALVES IN SERIES

One of the most effective techniques for suppressing cavitation is to dissipate energy by placing multiple losses in series. This can be done with a single valve using special cavitation trim like the type described in the preceding section or by installing orifices in series downstream from the main control valve. There are two reasons why this technique is so effective in controlling cavitation. First, the pressure drop created by the downstream orifice increases the pressure at the upstream orifice allowing it to

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produce a larger pressure drop without cavitating. Secondly, since a larger pressure drop is possible, the upstream orifice will have a smaller C_d , which means that the cavitation index is smaller and the allowable pressure drop is further increased. Figure 6.7 and Table 6.4 show how the sigma values for an orifice reduces as the C_d reduces.

The use of orifices in series in a pipe is effective only when the range of flows is less than about 1:3. The orifices are designed to have optimum performance at maximum flow. The pressure drop created by the orifices is proportional to the square of the flow rate. If the flow is reduced to 1/3 the pressure drop will reduce to 1/9.

The following examples demonstrate the procedure for designing multiple orifices plus the influence of the cavitation limit and the influence of the pipe diameter on the number of orifices required.

Example 7.1 Design a series of orifices to be installed in a 12-inch pipe to reduce the pressure from 538 psi to 24 psi at a flow rate of 20 cfs. $P_b = 13.5$ psia, $P_v = 0.2$ psia. Design the system so that each orifice operates at critical cavitation. The calculations are shown in Table 7.1. Reference data for the system and orifices are listed in the upper two blocks of the Table.

The solutions and are most easily done with a spreadsheet because the calculations require a trial and error solution. Table 7.1 shows the calculations for this example. The first set of calculations in the spreadsheet are outlined below. Start the cavitation analysis by assuming a pressure for the first orifice. Reference data for the system and orifices are listed in the upper two blocks of data.

Orifice #1

$$P_u = 538 \text{ psi (given)}$$

$$\Delta P = 255 \text{ psi (assumed, this is where the trial and error is done)}$$

$$P_{v \text{ ref}} = 0.2 \text{ psia}$$

$$P_{1 \text{ ref}} = 102.2 \text{ psia}$$

$$V = \frac{Q}{A} = 25.46 \text{ fps}$$

$$C_d = \frac{25.46}{(64.4 \cdot 2.308 \cdot 255 + 25.46^2)} = 0.130$$

$$\sigma_{\text{sys}} = \frac{(538 + 13.5 - 0.2)}{255} = 2.16$$

Table 7.1 Calculations for Series Orifices at Critical Cavitation

Reference System Data

Flow=	20.00	cfs
Pipe Diameter=	12.00	inch
Pipe Area=	0.785	sq-ft
Velocity=	25.46	fps
Barometric Pressure=	13.5	psia
Vapor Pressure=	0.20	psia
Upstream Pressure=	538.00	psig
Downstream Pressure=	24.00	psig

Reference Cavitation Data

P1 ref (Pu+Pb)=	102.20	psia
Ref Baro. Press. Pb ref=	12.20	psia
Pv ref=	0.2	psia
Reference Pipe Dia.=	3.00	inch
Design Level: Critical Cav.		
PSE Exponent, X=	0.00	
Reference Cav. Data: Eqn. 6.2		

Orifice #	Pu psi	Pd psi	DP psi	System Sigma	Cd orifice	Sigma c ref Eqn. 6.2	PSE critical	SSE critical	Sigma crit. Adjust	do/d eqn.	do in.
1	2	3	4	5	6	7	8	9	10	11	12
1	538.0	308.0	230.0	2.40	0.136	2.01	1.00	1.17	2.35	0.446	5.35
						2.04					
2	308.0	188.0	120.0	2.68	0.187	2.24	1.00	1.20	2.68	0.511	6.13
3	188.0	123.5	64.5	3.12	0.252	2.52	1.00	1.24	3.12	0.576	6.91
4	123.5	86.0	37.5	3.65	0.323	2.85	1.00	1.27	3.63	0.630	7.56
5	86.0	62.5	23.5	4.23	0.396	3.21	1.00	1.31	4.22	0.671	8.05
6	62.5	47.1	15.4	4.92	0.470	3.63	1.00	1.35	4.91	0.706	8.47
7	47.1	36.6	10.5	5.75	0.542	4.09	1.00	1.40	5.70	0.738	8.86
8	36.6	28.9	7.7	6.48	0.601	4.52	1.00	1.43	6.48	0.769	9.23
9	28.9	24.0	4.9	8.61	0.686	5.22	1.00	1.50	7.82	0.826	9.91

Column Source of data

- 1 Orifice Number, starting from far upstream
- 2 Pressure upstream from orifice, given for first orifice, for others Pu = Pd of US orifice
- 3 Pressure downstream from orifice
- 4 Pressure drop determined by trial and error until Col. 5 is slightly greater than or equal to Col. 10
- 5 Calculated system sigma
- 6 Discharge coefficient
- 7 Reference cavitation data for orifices (Eqn. 6.2 or Fig. 6.7)
- 8 Pressure scale effects equation (Eq. 5.5)
- 9 Size scale effects equation (Eqs. 5.7 & 5.8)
- 10 Adjusted critical sigma value = Reference Sigma * PSE * SSE
- 11 Calculated orifice to pipe diameter ratio, using Equation 6.6
- 12 Orifice hole diameter

Make the PSE and SSE adjustment using Equations 5.5 and 5.7-5.8 (Remember that there are no pressure scale effects at critical cavitation for orifices). Since PSE corrections are needed in subsequent examples the procedure will be shown here using X = 0.

$X = 0$ ie. no PSE

$$PSE = \left[\frac{(538 + 13.5 - 0.2)}{(102 - 0.17)} \right]^0 = 1.00$$

$$Y = 0.3 K^{-0.25} \quad (\text{Eqn. 5.8})$$

$$K = \frac{1}{0.136^2} - 1.0 = 53.1$$

$$Y = 0.3 (53.1)^{-0.25} = 0.111$$

$$SSE = \left(\frac{D}{d} \right)^Y = \left(\frac{12}{3} \right)^{0.111} = 1.17 \quad (\text{Eqn. 5.7})$$

$\sigma_{c \text{ ref}}$ can be calculated from Equation 6.2, as 1.98
 $\sigma_c = 1.17 \cdot 1.00 \cdot 2.01 = 2.35$

Since $\sigma_{\text{sys}} = 2.16$ is slightly greater than $\sigma_c = 2.14$, the assumed pressure drop is close to the maximum allowable. The rest of the calculations are listed in Table 7.1. The pressure drop in Column 4 is changed for each orifice until the σ_{sys} is equal to or slightly larger than the adjusted σ_c in column 10. The pressure downstream from the first orifice is the upstream pressure for the next orifice.

For this system, eight orifices in series are required to dissipate the energy and have the orifices operate at critical cavitation. Note how the allowable pressure drop varies with each orifice. The upstream orifice has an allowable pressure drop of 255 psi and the downstream orifice can only operate at a 5.0 psi pressure drop.

Example 7.2 Rework Example 7.1 and design a series of orifices to be installed in the 12-inch pipe so that each orifice operates at incipient damage. The calculations are shown in Table 7.2. Note that at incipient damage there are pressure scale effects but no size scale effects corrections are needed. Reference data for the system and orifices are again listed in the upper two blocks of the table.

By allowing the orifices to operate at incipient damage rather than critical cavitation, the number of orifices can be reduced from eight to five. The next example shows that increasing the pipe diameter to reduce the velocity also reduces the required number of orifices.

Table 7.2 Calculations for Series Orifices at Incipient Damage

Reference System Data

Flow=	20.00	cfs
Pipe Diameter=	12.00	inch
Pipe Area=	0.785	sq-ft
Velocity=	25.46	fps
Barometric Pressure=	13.5	psia
Vapor Pressure=	0.20	psia
Upstream Pressure=	538.00	psig
Downstream Pressure=	24.00	psig

Reference Cavitation Data

P1 ref (Pu+Pb)=	102.20	ps a
Ref Baro. Press. Pb ref=	12.20	psia
Pv ref=	0.2	psia
Reference Pipe Dia.=	3.00	inch
Design Level:	Incip Damage	
PSE Exponent, X=	0.19	
Reference Cav. Data:	Eqn. 6.3	

Orifice #	Pu psi	Pd psi	DP psi	System Sigma	Cd orifice	Sigma id ref Eq. 6.3	PSE inc. dam.	SSE inc. dam.	Sigma id Adjust	do/d eqn.	do in.
1	2	3	4	5	6	7	8	9	10	11	12
1	538.0	295.0	243.0	2.27	0.133	1.63	1.38	1.00	2.25	0.441	5.29
2	295.0	160.0	135.0	2.28	0.177	1.84	1.23	1.00	2.27	0.499	5.99
3	160.0	86.0	74.0	2.34	0.236	2.10	1.11	1.00	2.32	0.562	6.74
4	86.0	45.0	41.0	2.42	0.310	2.41	0.99	1.00	2.40	0.621	7.45
5	45.0	24.0	21.0	2.78	0.415	2.86	0.90	1.00	2.57	0.681	8.17

Column Source of data

- 1 Orifice Number, starting from far upstream
- 2 Pressure upstream from orifice, given for first orifice, for others $P_u = P_d$ of US orifice
- 3 Pressure downstream from orifice
- 4 Pressure drop determined by trial and error until Col. 5 is slightly greater than or equal to Col. 10
- 5 Calculated system sigma
- 6 Discharge coefficient
- 7 Reference cavitation data for orifices (Eqn. 6.3 or Fig. 6.7)
- 8 Pressure scale effects equation (Eq. 5.5)
- 9 Size scale effects equation (Eqs. 5.7 & 5.8)
- 10 Adjusted incipient damage sigma value = Reference Sigma * PSE * SSE
- 11 Calculated orifice to pipe diameter ratio, using Equation 6.6
- 12 Orifice hole diameter

Example 7.3 Redesign the system in Examples 7.2 using a larger pipe and still having the orifices operate at incipient damage. In Examples 7.1 and 7.2 the velocity in the pipe was relatively high (25.5 fps). If the pipe diameter is expanded at the orifice location so the orifices are in a 16-inch diameter pipe, Table 7.3 shows that only three orifices would be needed. The primary reason for the reduction in the required number of orifices is that with a smaller velocity, each orifice operates at a smaller C_d where they have better cavitation performance.

In the three preceding examples, the design considered only a single discharge. If a variable flow is required, the system must include a control a valve and the design must include an analysis to determine the safe throttling range of the valve. The next example shows a design where two orifices are located downstream from a globe valve.

Table 7.3 Calculations for Series Orifices At Incipient Damage

Reference System Data

Flow=	20.00	cfs
Pipe Diameter=	15.25	inch
Pipe Area=	1.268	sq-ft
Velocity=	15.77	fps
Barometric Pressure=	13.5	psia
Vapor Pressure=	0.20	psia
Upstream Pressure=	538.00	psig
Downstream Pressure=	24.00	psig

Reference Cavitation Data

P1 ref (Pu+Pb)=	102.20	psia
Ref Baro. Press. Pb ref=	12.20	psia
Pv ref=	0.2	psia
Reference Pipe Dia.=	3.00	inch
Design Level: Incip Damage		
PSE Exponent, X=	0.19	
Reference Cav. Data: Eqn. 6.3		

Orifice #	Pu psi	Pd psi	DP psi	System Sigma	Cd orifice	Sigma id ref Eq. 6.3	PSE inc. dam.	SSE inc. dam.	Sigma id Adjust	do/d Eq. 6.5	do in.
1	2	3	4	5	6	7	8	9	10	11	12
1	538.0	178.0	360.0	1.53	0.068	1.30	1.38	1.00	1.41	0.335	5.11
2	178.0	63.0	115.0	1.66	0.120	1.57	1.13	1.00	1.64	0.421	6.43
3	63.0	24.0	39.0	1.96	0.203	1.96	0.95	1.00	1.91	0.528	8.06

Column Source of data

- 1 Orifice Number, starting from far upstream
- 2 Pressure upstream from orifice, given for first orifice, for others $P_u = P_d$ of US orifice
- 3 Pressure downstream from orifice
- 4 Pressure drop determined by trial and error until Col. 5 is slightly greater than or equal to Col. 10
- 5 Calculated system sigma
- 6 Discharge coefficient
- 7 Reference cavitation data for orifices (Eqn. 6.3 or Fig. 6.7)
- 8 Pressure scale effects equation (Eq. 5.5)
- 9 Size scale effects equation (Eqs. 5.7 & 5.8)
- 10 Adjusted incipient damage sigma value = (Reference Sigma-1) * PSE * SSE + 1
- 11 Calculated orifice to pipe diameter ratio, using Equation 6.6
Note that because the Cd value is below the range of experimental data for the first orifice there is uncertainty about the orifice diameter calculation using Eqn. 6.5
- 12 Orifice hole diameter

Example 7.4 Design a pressure reducing station for a 12-inch diameter pipeline using one globe valve and one or more orifice plates in series with all devices operating at or below incipient damage. Assume that the upstream pressure is constant at 71 psig, $P_b = 13.2$ psia, $P_v = 0.3$ psia and the pipe discharges into the atmosphere so $P_d = 0$. The design discharge is 11.7 cfs. Also find the minimum flow where the globe valve can control the flow and not be damaged by cavitation.

Design Principles

1. To provide the maximum controllability, the system should be designed so that the valve is full open at maximum discharge. Therefore, start the design by selecting one or more orifice plates that provide essentially all the pressure drop, the proper discharge and operate at or below incipient damage.

2. The design procedure is trial and error because it is necessary to first assume a pressure drop and calculate the C_d and σ_{sys} . Then find reference cavitation data at $\sigma_{id \text{ ref}}$ for the orifices (and later the globe valve), correct it for scale effects and compare the adjusted σ_{id} to the σ_{sys} .

3. The data are listed in Table 7.4. The first set of calculations are listed below.

Globe Valve The globe valve is set full open at maximum flow so the orifices can take all the pressure drop possible. The full open C_d for the globe valve is 0.40. The calculated pressure drop (7.8 psi) and sigma value (10.68) are listed in the table. No cavitation analysis is needed for the full open globe valve since the system sigma is so high.

Orifice #1 (upstream orifice) For a pipe diameter of 12-inch, the problem is to determine the hole diameter for the first orifice so it operates at incipient damage. The solution is trial and error, until $\sigma_{sys} = \sigma_{id}$.

Try a pressure drop of 38 psi

$$V = Q/A = 14.9 \text{ fps}$$

$$P_u = 63.2 \text{ psi}$$

$$P_d = 63.2 - 38 = 25.2 \text{ psi}$$

$$C_d = \frac{14.9}{\sqrt{(64.4 \cdot 2.308 \cdot 38 + 14.9^2)}} = 0.194$$

To check for cavitation, find reference cavitation data in Table 6.4 or use the accompanying equations.

$$P_{v \text{ ref}} = 0.2 \text{ psia}$$

$$P_{1 \text{ ref}} = 102.2 \text{ psia}$$

$$X = 0.19 \text{ (Table 5.3)}$$

$$\sigma_{id \text{ ref}} = 1.92 \text{ (Eqn. 6.3)}$$

$$\sigma_{sys} = \frac{(63.2 + 13.0 - 0.30)}{38} = 2.00$$

Adjust for pressure scale effects.

$$\sigma_{id} = (1.92 - 1.0) \left(\frac{(63.2 + 13.2 - 0.30)}{82 - 0.2} \right) 0.19 + 1.0 = 1.91$$

Since σ_{sys} is just slightly smaller than σ_{id} , the solution is correct.

Selecting the orifice hole diameter is done using Equation 6.6 which gives $d_o/d = 0.519$ and $d_o = 6.23$ -inches. If it is desired to have multiple holes in the orifice, select the hole diameter and use the total orifice area to calculate the number of holes required.

The rest of the calculations are listed in Table 7.4. For each orifice, and for the valve at minimum flow, the ΔP in Columns 4 & 16 must be adjusted until the σ_{sys} in Columns 5 or 17 is equal to or slightly less than the adjusted σ_{id} in Columns 10 or 22.

For the second and third orifices, $P_u = P_d$ for the upstream orifice. The calculations are continued until enough orifices are selected to drop the available pressure of 71 psi to the required downstream pressure (zero in this example). Note that at maximum flow, the 12-inch globe valve will cause a pressure drop of 7.8 psi when it is wide open. This leaves 63.2 psi for the three orifices. For this example it takes 3 orifices to limit the cavitation level to incipient damage.

The analysis should be repeated at smaller flow rates to find when the globe valve begins to operate at incipient damage. This is done by keeping the orifice sizes and C_d values constant and calculating the pressure drop across each at reduced flow. The flow is reduced until $\sigma_{sys} = \sigma_{id}$ for the valve. The spreadsheet must be modified slightly. The bottom set of data show the calculations. The values of C_d for the orifices are fixed at the values in the upper set of data. The P_u values are calculated by:

$$P_u = P_d + \frac{K \cdot V^2}{2.308 \cdot 2g}$$

where

$$K = \frac{1}{C_v^2} - 1$$

The minimum flow where the globe valve is operating at incipient damage is determined by trial and error. At 5.00 cfs, the three orifices produce 5.7 psi pressure drop, the valve drops 65.3 psi and operates at a σ_{sys} which is just larger than σ_{id} . Therefore, the valve can control the flow between 5.00 and 11.7 cfs without damaging cavitation. Below 5.00 cfs the valve will be damaged.

Table 7.4 Calculations for Series Orifices and Valve at Incipient Damage

Reference System Data

Maximum Flow=	11.7	cfs
Pipe Diameter=	12.0	inch
Pipe Area=	0.785	sq-ft
Velocity=	14.90	fps
Barometric Pressure=	13.0	psia
Vapor Pressure=	0.30	psia
Downstream Pressure=	0.00	psig
Upstream Pressure=	71.00	psig

Reference Cavitation Data

Orifice P1 ref (Pu+Pb)=	102.20	psia
Globe valve ref P1=	82.00	psia
Ref Baro. Press. Pb ref=	12.20	psia
Pv ref=	0.2	psia
Reference Orifice Pipe Dia.=	3.00	inch
Design Level: Incip Damage		
PSE Exponent, X=	0.11	valve
PSE Exponent, X=	0.19	orifice
Reference Orifice Cav. Data: Eqn. 6.3		

Orifice #	Pu psi	Pd psi	DP psi	System Sigma	Cd orifice	Sigma id ref Eq. 6.3	PSE inc. dam.	SSE inc. dam.	Sigma id Adjusted	do/d eqn.	do in.
1	2	3	4	5	6	7	8	9	10	11	12
Design for max. flow with valve full open, Cd = 0.40											
12" globe	71.0	63.2	7.8	10.68	0.400	Valve full open, no cavitation calculations needed					
#1 orifice	63.2	25.2	38.0	2.00	0.194	1.92	0.99	1.00	1.89	0.519	6.23
#2 orifice	25.2	8.2	17.0	2.23	0.284	2.30	0.90	1.00	2.07	0.602	7.23
#3 orifice	8.2	0.0	8.2	2.56	0.393	2.77	0.84	1.00	2.32	0.670	8.04

Orifice #	Pu psi	Pd psi	DP psi	System Sigma	Cd orifice	Sigma id ref Eq. 6.3	PSE inc. dam.	SSE inc. dam.	Sigma id Adjusted	do/d eqn.	do in.
13	14	15	16	17	18	19	20	21	22	23	24
Find the minimum flow for the globe valve to operate at incipient damage											
Min. Flow=	4.10	cfs	(find Min. Flow by trial and error until system sigma equals adjusted sigma id)								
12" globe	71.0	7.8	63.2	1.32	0.054	1.29	0.96	1.05	1.31		
#1 orifice	7.8	3.1	4.7	4.38	0.194	No need to check for cav. at min. flow				0.519	6.23
#2 orifice	3.1	1.0	2.1	7.56	0.284	No need to check for cav. at min. flow				0.602	7.23
#3 orifice	1.0	0.0	1.0	13.67	0.393	2.77	0.80	1.00	2.22	0.670	8.04

Column Source of data

1 thru 12 See Tables 7.1-7.3

14 Calculated pressure upstream from orifice or valve

15 Pressure downstream from orifice

16 Calculated pressure drop based on Cd and flow

17 Calculated system sigma

18 Discharge coefficients for orifices is the same as in Col 6

19 Reference cavitation data

20 Pressure scale effects equation (Eq. 5.5)

21 Size scale effects equation (Eqs. 5.7 & 5.8)

22 Adjusted incipient damage sigma value = Reference Sigma * PSE * SSE

23 Orifice to pipe diameter ratio, same as in Column 11

24 Orifice hole diameter, same as in Column 12

7.3 VALVES IN PARALLEL OR LARGER VALVES

If a conventional valve cannot produce the required pressure drop without excessive cavitation, one possible solution is to use a larger valve or place two valves in parallel. This concept has limited application and works only for systems where the selected valve is only operating slightly beyond the desired level of cavitation. It is not a practical solution when significant reduction in the cavitation intensity is required. This is because over sizing control valves causes them to operate at very small openings which may result in seat damage, poor control and increase the chance of hydraulic transients. Even though this idea has limited application, it is discussed because it demonstrates an important principle of cavitation and selection of control valves. The principle was introduced in Example 7.3.

To explain how over sizing a valve reduces cavitation, first consider the cavitation characteristics of most valves. Referring to Figures 6.4-6.7, it is observed that for all cavitation limits, the magnitude of σ increases significantly as the C_d or valve opening increases. Since the magnitude of σ is inversely related to the allowable pressure drop across the valve, a valve operating at a smaller opening can produce a larger pressure drop at the same level of cavitation. Consequently, if a larger valve is used, the flow velocity is reduced but the required pressure drop is the same. This forces the valve to operate at a smaller opening and smaller C_d where it has better cavitation performance.

Similarly, if two valves are placed in parallel, the velocity through each can be less than that of a single valve and each valve must operate at a smaller valve opening and a smaller C_d to produce the same pressure drop.

This alternative should not be used if it requires selecting a valve that is more than one nominal size larger than what would be selected as the optimum size based on good flow control.

7.4 USE OF SUDDEN ENLARGEMENTS

Another method for suppressing cavitation and erosion damage is removing the boundary from the cavitation zone by use of sudden enlargements. This is one of the most effective methods of dissipating energy while suppressing cavitation. In the limit, if the valve were designed to discharge into a large tank, where there is essentially no boundary, some valves could operate cavitation damage free in that environment at essentially any pressure differential. The criteria is that there be no cavitation inside the valve at the seats, etc. One of the disadvantages of using this approach is that the loss coefficient for the valve increases as the diameter ratio of the sudden enlargement increases. This reduces the valves capacity. If large pressure drops are desirable, the increased loss is not a problem. However, if it is necessary to operate a system with low head loss, it may be necessary to select the next larger valve size.

The sudden enlargement solution works best for valves where the jet from the control element can be directed immediately into the enlarged pipe section. Figure 7.1

shows this principal applied to a butterfly valve. If the valve is designed so that the seat is at the outlet flange and the shaft offset from the disc (as shown in the figure), the jets separate from the seats into the enlarged pipe. The cavitation occurs near the center of the pipe, protecting the boundary from damage. This concept also has application for valves where the jets discharge internally, like globe valves. Tests show that if the body of the globe valve is enlarged, it can operate at larger pressure differentials without cavitation.

Figure 7.1 Use of Sudden Enlargement With a Butterfly Valve

7.5 CAVITATION RESISTANT MATERIALS

Another method of suppressing cavitation damage is to cover parts of the valve and downstream piping with cavitation resistant materials. With this solution, one tolerates the cavitation intensity but tries to prevent erosion by increasing the erosion resistance of the boundary.

Table 7.4 (Ref. 19) lists relative resistance of numerous types of materials to cavitation erosion damage. These data were obtained from tests using a magnetostriction

device where cavitation is created by a very high frequency vibration of a small button of material. The weight loss in milligrams over a two hour period is listed for each material. It is seen that there is a wide variation in the resistance of the various material. One caution which should be mentioned regarding application of the data in Table 7.4 is that different types of cavitation may result in different erosion rates. With the magnetostriction device, the cavitation events are very small and occur at an extremely high rate since the device is vibrating at approximately 20,000 cycles per second. If the same material is subjected to cavitation which occurs in a flowing system such as valves or pumps, the cavitation events are much larger and occur at a lower frequency and consequently the different materials may behave different.

Table 7.5 Cavitation Erosion Resistance of Metals (Ref. 18 and 43)

Alloy	Magnetostriction weight loss after 2 hours (mg)
Rolled stellite	0.6
Welded aluminum bronze	3.2
Cast aluminum bronze	5.8
Welded stainless steel (2 layers, 17 Cr-7 Ni)	6.0
Hor rolled stainless steel (26 Cr-13 Ni)	8.0
Tempered rolled stainless steel (12 Cr)	9.0
Cast stainless steel (18 Cr-8 Ni)	13.0
Cast stainless steel (12 Cr)	20.0
Cast manganese bronze	80.0
Welded mild steel	97.0
Plate steel	98.0
Cast steel	105.0
Aluminum	124.0
Brass	156.0
Cast iron	224.0
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Laboratory testing has also been conducted to evaluate the relative resistance of aluminum, brass, 304 stainless and carbon steel subjected to cavitation caused by flow through an orifice (Ref. 25). Cavitation was detected by counting the pitting rate at different levels of cavitation to determine incipient damage. The experimental data are listed in Table 7.5. The data show that carbon and stainless steel begin to pit at about the same sigma value with aluminum and brass damaging at a much lower incipient sigma.

The data show that basing the incipient damage condition on tests with soft aluminum is somewhat conservative since σ_{id} for the aluminum is about half as much as it is for the other three materials. It is this authors opinion that basing σ_{id} on tests with aluminum is a reasonable design limit and it is not overly conservative. This conclusion is based primarily on the fact that once damage begins it increases exponentially with

increased velocity. Operating beyond incipient damage determined using aluminum greatly increases the chance of damage.

Table 7.6 Incipient Cavitation Damage Data For Different Metals (Ref. 25 & 43)

Material	P1 (psi)	Incipient Damage Sigma	Incipient Damage Velocity
Aluminum	92.2	0.804	15.8
Brass	92.2	0.512	16.9
Carbon Steel	93.2	0.397	17.8
304 Stainless Steel	92.2	0.404	17.6

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Recent research has been reported of flow tests on cavitation erosion resistance of different materials (Ref. 12) in a flowing system. Figure 7.2 summarizes these data. Pre-weighed test bars of different material having the same dimensions were placed in a cavitating jet. Each sample was subjected to 193 fps water jet for 28 days. The weight loss of each alloy was converted into wear or volume loss using the density of the test bar to determine the actual amount of metal removed. The data shows significant differences in the wear resistance of the alloy metals.

Figure 7.2 Cavitation Erosion Damage for different metals (Ref. 12)

7.6 AERATION

Injecting air to suppress cavitation is a technique which has been used for many years to suppress cavitation. If an adequate amount of air is injected into the proper region, the noise, vibrations, and erosion damage can be significantly reduced. If the air is introduced so that it gets dispersed throughout the region where cavitation originates, it suppresses the formation of vaporous cavitation and cushions the collapse. When cavitation nuclei are made up of a large percentage of gas, vaporization of the fluid is greatly reduced due to expansion of the gas. This type of cavitation is called gaseous cavitation and it usually does not cause serious damage.

In the zone where cavitation collapses, a small amount of air provides a cushioning effect reducing the noise, vibration and the possibility of erosion damage. This is primarily due to the increased compressibility provided by the air and a corresponding reduction in the acoustic wave speed of the liquid. As little as 1% free air can reduce the wave speed from its maximum value in water of about 4200 fps to a few hundred fps. The magnitude of the shock wave created when the cavity collapses is reduced linearly with the reduction of the wave speed. This significantly reduces or eliminates the damage.

The concept of using air to suppress cavitation has been successfully used for years to eliminate cavitation damage on dam spillways. There are a few fundamental differences between injecting air into a pipe to suppress cavitation at a valve, orifice, pump or turbine compared to using an air slot to suppress cavitation on a spillway. One of these differences is that in a pipe the cavitation is usually local; being generated at a valve, pump, turbine or other minor loss. As a result, it is necessary only to protect a small portion of the system from cavitation damage. With an open channel application such as a spillway, it is often necessary to protect large areas of the surface from cavitation erosion damage.

For air slots on spillways or at gate slots, it is usually desirable to maximize the amount of air drawn in by the air slot. This maintains a high air concentration at the boundary for a greater distance. Since buoyant forces and pressure gradients cause the air to rise to the surface, an air slot can only protect the boundary for a finite length. In some cases multiple air slots in series have been used.

For pipe flow, one must be concerned about the effect of the free air on the accuracy of flow meters, reduction of flow due to air accumulation at high points, transients caused by large slugs of air moving through the pipe or through a control valve and various other similar problems. It is, therefore, necessary for closed conduit flow to minimize the amount of air required to suppress cavitation.

When using air to suppress cavitation at a valve, pump, turbine, etc., it is difficult to identify all of the zones of cavitation and to supply air to those areas. Often cavitation in a closed conduit system will occur when the average pressure in the cavitation region is above atmospheric. This necessitates the use of compressed air. Such a solution is possible, but it becomes more complex, costly and less reliable. The

amount of air required to suppress cavitation is generally between 1-5% by volume of the water flow rate. This figure is based on a study with orifice plates (Ref. 8), a cone valve (Ref. 22) and a butterfly valve (Ref. 9). The percentage for other types of valves may be different.

One of the mistakes with air injection is placing the air inlet port in the wrong location so the air does not get into the cavitation zone. The author is aware of an installation where air was supplied to a 36 inch butterfly valve to suppress cavitation. The valve was installed with the shaft vertical and two 3 inch air ports were located on the sides of the pipe so they were in a plane perpendicular to the shaft. The air ports sucked in large quantities of air but did not reduce the cavitation. The problem was that the air drawn in by the high velocity jets was washed downstream. It was not able to penetrate through the jets and get into the separation zone where the cavitation was occurring. When the air ports were placed on the top and bottom of the pipe close to the shaft, the cavitation was suppressed. Similar problems have been observed in the laboratory.

Injecting air usually does not entirely eliminate the cavitation damage. Table 7.7 lists cavitation aeration data for a 6-inch skirted cone (solid conical plug) valve and a 6-inch butterfly valve. The data show that the percent reduction in σ_{id} when air was introduced varied between about 10% and 60%. Even though the cavitation damage was not completely eliminated, the reduction was significant enough to consider injecting air.

Table 7.7 Influence of Injecting Air on σ_{id} . (Ref. 43)

V.O. deg	C_d	σ_{id}	σ_{id}	%Change in σ_{id}	%Air (air)
<u>6" butterfly valve (Ref. 9)</u>					
20	0.0322	0.39	0.22	- 43	3.3
40	0.244	0.77	0.57	- 26	4.3
50	0.343	1.12	0.816	- 27	3.4
60	0.481	2.73	0.664	- 62	5.0
<u>6" skirted cone (Ref. 22)</u>					
30	0.090	0.85	0.35	- 59	1
40	0.155	0.97	0.40	- 59	1
50	0.26	1.32	0.75	- 43	1
60	0.40	1.75	1.55	- 11	1
70	0.61	3.35	2.85	- 15	1

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