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Stop cavitation from destroying your control valve trims

The devastating effects cavitation can have on control valve internals is widely appreciated. In this article, the authors therefore explain the origins of cavitation, provide an easy-to-use method to estimate cavitation sound levels and conclude with approaches to reduce or even avoid cavitation.

By Hans D. Baumann and
Jon Monsen

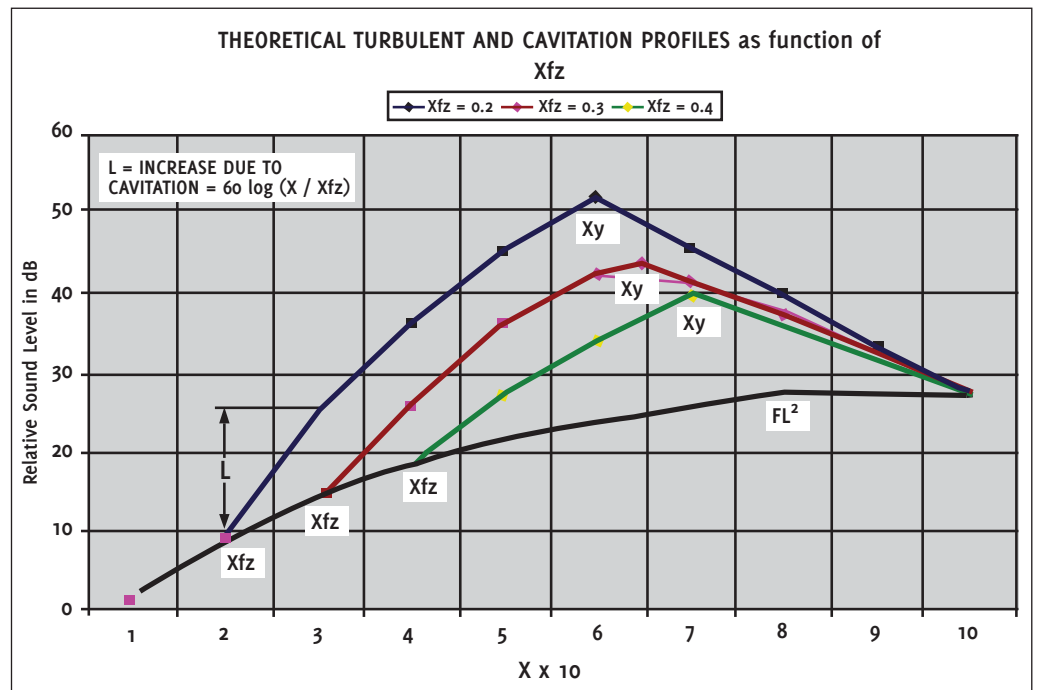


Figure 1. Relationship between hydro-noise levels and X_{fz} . (Black line shows turbulent sound.)

It has been observed that when the downstream pressure in a valve decreases at and beyond a corresponding pressure ratio, called X_{fz} (see Ref. 1), the static pressure in the valve's orifice reaches the vapor pressure of the liquid, this causes evaporation of some of the liquid with simultaneous formation of small vapor bubbles. At this point the liquid reaches its highest velocity. After passing the orifice, the liquid velocity starts to decrease while the static

pressure increases. When the liquid static pressure exceeds vapor pressure, the bubbles start to implode causing sonic shock waves. This causes acoustic sound waves following the laws of gas acoustics.

Following further reduction in outlet pressure causes more of the diameter of the jet to enlarge and its velocity to decrease in proportion to the increased static pressure following the laws of Bernoulli (assuming constant inlet pressure).

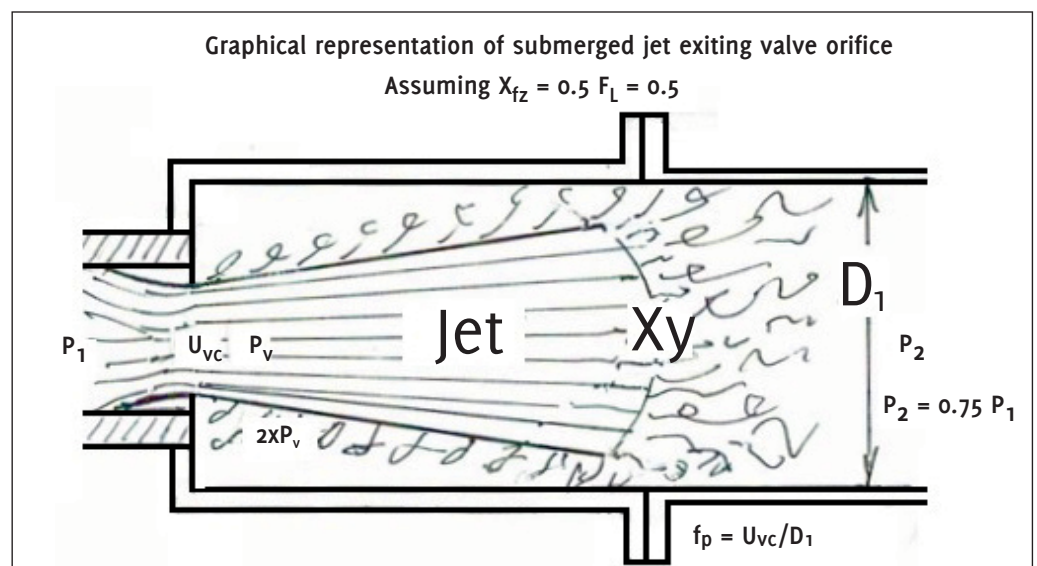


Figure 2: Schematic view of a submerged water jet. Here, f_p is the peak frequency of the turbulent jet sound (Hz.)

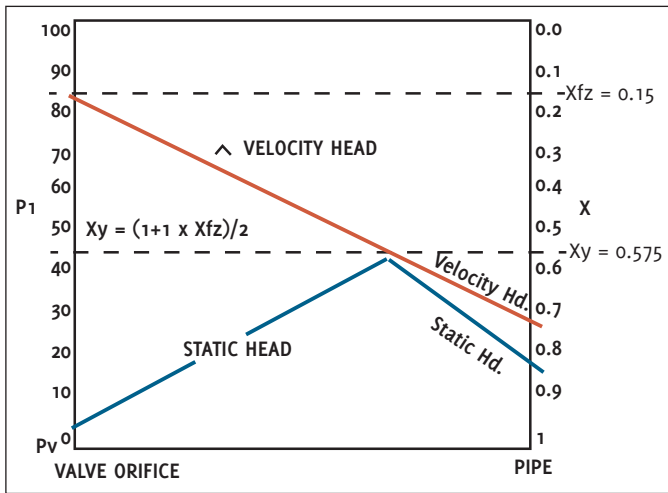


Figure 3: Schematic of pressure gradients of a submerged water jet. Inlet pressure = 100 psia, Xfz = 0.15, Xy = 0.575

The resultant sound pressure from cavitation increases corresponding to $60 \log(X/Xfz)$ following the acoustic gas laws (here X is a pressure ratio = $(P_1 - P_2)/(P_1 - P_v)$). Xfz is a coefficient of incipient cavitation. Figure 1 shows the relationship between Xfz, the coefficient of incipient cavitation and the resultant sound level. Manufacturers publish specific Xfz values for their valves. Xy signifies the maximum cavitation amplitude. It is defined as $Xy = (1+Xfz)/2$.

According to Bernoulli's law, in a horizontal pipe (or valve housing), any increase in the speed of the fluid occurs simultaneously with a decrease in static pressure without a change in the fluid's potential energy. Of course this also works in reverse, that is, static pressure will increase when the speed decreases. Here is the chain of events: $(P_1 - P_2) \rightarrow X \rightarrow \text{jet velocity} \leftarrow \text{static pressure}$. Finally, at a pressure ratio Xy, about half way between Xfz and X = 1, the cavitation process reverses. Here, the static pressure matches the dynamic pressure, or velocity head (see Figure 3). Of significance is that at this pressure ratio, the resultant downstream pressure is approximately one half of that of the downstream pressure at Xfz. Following Xy, further outlet pressure reduction results in the termination of bubble implosions followed by a rapid but somewhat disorganized decrease in sound pressure, until the cavitation sound pressure level reaches that of the turbulent water. This happens at, or close to, X = 1, followed by a process of flashing. At this stage all vapor bubbles which escaped implosions stay dispersed in the turbulent water.

Little kinetic energy (except some friction loss and acoustic power conversion) is converted during the jet's expansion. However following the Xy pressure ratio nearly all of the kinetic energy (equivalent to $P_1 - P_2$) is converted into heat due to turbulence.

Damage caused by cavitation

As Figure 4 shows, prolonged cavitation has eaten away one half of the original parabolic valve plug. Similar destruction has happened at the corresponding valve orifice. It should be noted that liquids other than water can be just as destructive. Consider oil in pipelines for example. There is a correlation between sound and erosion. Destructions like

these happen when the external sound level exceeds 80 dB(A); also Jon Monsen (Ref. 2) suggests a sliding scale such as 85 dB(A) for valve sizes 4" to 6", 90 dB(A) for sizes 8" to 14" and 95 dB(A) for even larger sizes. There is no known valve trim material which could withstand cavitation damage due to the extremely high pressure waves (exceeding 100,000 psi) of the imploding vapor bubbles,



Figure 4: Cavitation damaged valve plug and orifice. Photo Courtesy: Jon Monsen

even though, hardened trim such as ceramics or Stellite® have shown to prolong service life.

Estimating the sound level of cavitation

Here is a simplified, but accurate, method which can be performed using a simple device such as a pocket calculator.

Input needed:

Pipe diameter downstream of valve D in inch ($D = d/25.4$ if d in mm), P1 in psia (bara x14.5), P2 in psia (bara x14.5), flow coefficient Cv (at operating conditions), pressure recovery coefficient without expander FL, coefficient of incipient cavitation Xfz, liquid vapor pressure Pv, valve type modifier rw (3 for globe valves).

Initial calculations:

Pressure ratio $X = (P_1 - P_2)/(P_1 - P_v)$. $Xy = X$ at max. cavitation = $(1.00+Xfz)/2$.

Calculation steps:

$A = 8 \log(D) + 12 \log(Cv/D^2) + 4$ in dB
 $B = 35 \log(P_1) - 35$ for US, or $35 \log(14.5 p) - 35$ if p is in bara.
 $c_1 = 30 \log(10X) X_{\text{maximum}} = FL^2$.
 This equation gives turbulent sound level in dB(A) at 1m.
 c_2 if X is larger than Xfz but below Xy then $c_2 = 60 \log(X/Xfz)$. if X is above Xy then $c_2 = 60 \log(Xy/Xfz)$; otherwise use zero. This defines increasing cavitation sound level in dB
 c_3 if X is larger than Xy then $c_3 = 120 \log(X/Xy)$; otherwise use zero. This denotes decreasing cavitation sound level in dB.
 $C = c_1 + c_2 - c_3$ in dB(A).
 Final external sound level $LpAe1m = A + B + C - rw$ plus modifiers, in dB(A) at 1 m.

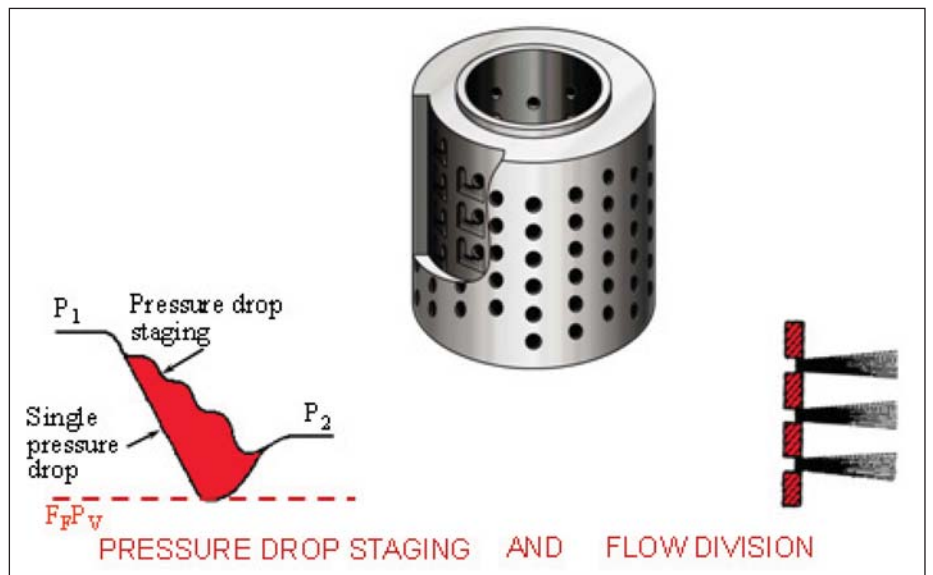


Figure 5: Typical three-stage anti-cavitation valve cage. Photo Courtesy: Hammel-Dahl, Inc.

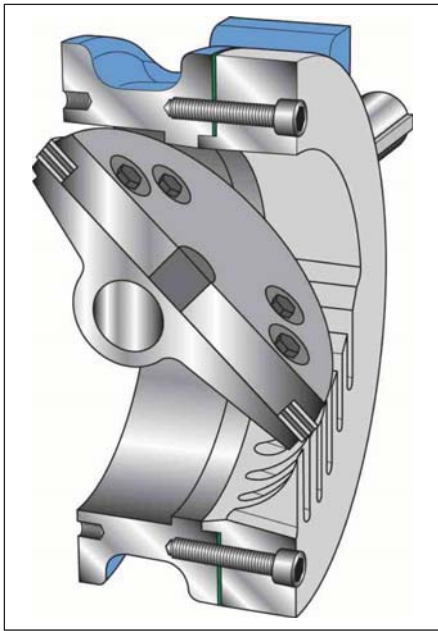


Figure 6: Anti-Cavitation butterfly valve, courtesy: Yeary & Associates, Inc.

Modifiers: For pipe schedule: Schedule 40 = 0, schedule 80 = -4 dB.

Worked example

A 3" butterfly valve installed in a 6" pipe, $P_1 = 100$ psia, $P_2 = 40$ psia, $P_v = 0.5$ psia, C_v under actual flow = 85. X_{fz} given as 0.28. $X = (P_1 - P_2)/P_1 - P_v = 0.6$ FL = 0.7. Note, X is limited to FL²

$$A = 8\log(6) + 12\log(85/36) + 4 = 14.7 \text{ dB(A)}$$

$$B = 35\log(100) - 35 = 35 \text{ dB(A)}$$

$$c_1 = 30\log(10 \times \text{FL}^2) = 20.7 \text{ dB(A)}$$

$$c_2 = 60\log(0.6/0.28) = 19.9$$

$$c_3 = 0$$

Total sound level = $A + B + c_1 + c_2 = 90.3 \text{ dB(A)}$. at 1 meter from a schedule 40 pipe.

How to avoid or reduce cavitation

The first line of the defense is to select valve types for your applications, which have a high X_{fz} factor. As you can see from Figure 1, the difference between an X_{fz} (incipient coefficient of cavitation) of 0.2 and one of 0.4 is an extra sound level of 14 dB. If this cannot be done, specify an "anti-cavitation" valve trim.

These devices consist typically of multi-ported cages or valve plugs, although special butterfly valves having serrated or slotted vanes also are effective. Similarly effective are ball valves having parts of a ball slotted or multi-ported.

Single stage multi-ported devices are only indirectly effective by increasing the X_{fz} factor of the valve. More effective are multi-stage devices such as the valve cage pictured in Figure 5, since they split up the overall pressure drop across the valve in addition to increasing the X_{fz} factor. Attention

must be paid to ensure that the X_{fz} factor of the last stage is high enough; otherwise the inner valve plug will be damaged.

For let-down valves, such as pressure reduction from a well, emptying a tank, or back-pressure from a cooling tower, the best solution is to use conventional valves and let a valve absorb the whole pressure drop in one stage. This makes X equal, or close to one. As you can see from the Figure 1, here the only sound level (and absence of cavitation) is from turbulence (the lower black line). However, this works only if there is no downstream pipe. This means that the valve has to be attached directly to a tank (Figure 7) or sump of a pump. Otherwise the fluid should be discharged to atmosphere, so for example to a settling pond. The reason is that downstream pipes build-up back-pressure due to the high volume generated by the high content of un-imploded vapor bubbles. Even though the vapor content might only be 2%, the ratio between vapor and water specific volume is above 1000:1. As a result, the vapor has to be compressed in order to pass through the pipe. The resultant back pressure, in turn, creates a secondary cavitation (now X is below 1) which typically is responsible for corroded pipes and elbows (erroneously attributed to "flashing"). Another tried approach is to install two identical valves in series. This is only marginal effective since the combined FL or X_{fz} is only increased by 40% instead of being doubled [$X_{fz} \text{ total} = (2x X_{fz})^{0.5}$]. Multi-ported resistance plates also are employed in order to split the pressure drop (keeping the valve's pressure drop below X_{fz} while the rest is absorbed by the plate). However this works only at fairly high flow rates since the pressure drop through the plate decreases rapidly by the square root of the flow rates.

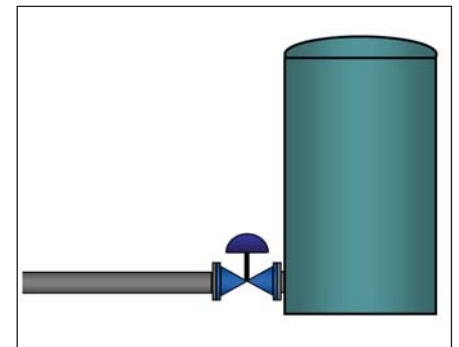


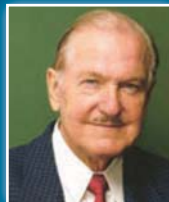
Figure 7: Example of control valve discharging directly into a flash tank. Courtesy John Molsen

Noise reduction for up to 10 dB has been achieved by injecting air into the cavitating liquid. Here the injected air bubbles absorb some of the impact of the cavitating vapor bubbles. Finally, pay attention to piping locations. Elevated inlet pipes increase inlet pressure and pressure drop, and lower pointing outlet piping increases suction and lowers outlet pressure. This could result in flashing if the outlet pressure is close to the vapor pressure.

References

- Ref 1: IEC Standard 60534-8-4, Prediction of Noise from Hydraulic Fluids, International Electrotechnical Commission, Geneva, Switzerland.
- Ref 2: Jon Molsen, Predicting Cavitation Damage in Control Valves, Flow Control Magazine, May, 2018, P. 14-17.
- Ref 3: Hans D. Baumann, How Loud is my Control Valve? VALVE-WORLD Magazine, May, 2018, P.1-2.
- Ref 4: Hans D. Baumann, Fluid Mechanics of Control Valves, ISA Publication, 2019.

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