Avoid control valve application problems with physics-based models

Kinetic energy criteria have many limitations

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Control valves play a central role in the control and optimal performance of a myriad of industrial flow processes. In "severe service" applications, control valves are equally crucial for safely dissipating high process fluid energy levels to avoid valve and piping damage from acoustic noise, vibration, cavitation and erosion. To varying degrees, all of these potentially damaging phenomena scale with flow velocities in the valve and valve trim, leading some valve manufacturers to recommend specific limits to fluid kinetic energy (*KE*) in the valve trim.

This article explores the rationale for *KE* limitations and demonstrates that *KE* criteria often provide very rough approximations of the actual physical phenomena that cause valve problems. In some instances, *KE* criteria mandate overly conservative and unnecessarily expensive control valve product solutions. On the other hand, *KE* criteria can significantly underestimate damage potential due to their weak correlation with a physical process, e.g., erosion caused by droplet or particle impaction surface forces. We advocate simple, physics-based methods that realistically model potentially problematic fluid phenomena, ensure safe valve operation and admit more cost-effective control valve solutions.

Simple, physics-based methods realistically model potentially problematic fluid phenomena.

Overview of control valve application problems. Control valves can experience a wide range of problems – from poor shutoff to excessive noise and, in extreme cases, component failure. All of these problems are created by adverse interaction of the flow with the valve components.

For instance, plug and/or seat surface erosion compromises the integrity of the interface, leading to degraded valve shutoff performance. Several flow-related phenomena can cause plug and/or seat erosion to occur, including impaction of particulates (i.e., slag, sand, catalyst fines, etc.) or liquid droplets, and cavitating flow at the seat. On the other hand, aerodynamic noise problems depend on the flow velocity and mass flowrate through the trim and valve outlet, as well the ability of the valve noise to excite (and propagate through) downstream piping. In some cases, different control valve problems arise from the same basic flow-valve interactions. For instance, large turbulent pressure fluctuations that interact with the trim can create both trim vibration and excessive aerodynamic noise. Component failure typically arises from extreme cases of poorly managed valve-related flow phenomena, e.g., large-scale valve cavitation or grossly excessive valve or expander noise.

High trim velocity and flow energy, per se, are not application problems in control valves. Rather, excessive noise, damaging trim or piping vibration, and trim or valve erosion are potential valve problems caused by flow physical phenomena. In turn, each flow phenomena depends on – but is not uniquely created by – flow velocity and energy. As a result, other authors have suggested general, velocity-based methodologies to assess the potential for a broad range of control valve application problems.

Miller[1] proposed applying valve trim maximum *KE* criteria to avoid a host of valve problems, including component vibration, breaking parts, excessive aerodynamic noise, trim and/or valve pitting and erosion caused by liquid cavitation or flashing, and surface erosion by solid particulate. Based on operational experiences, Miller and Stratton[2] present an allowable trim *KE* limit for a given control valve application that depends on the application service conditions. For example, they advocate a 70-psi (485-kPa) limit for a clean-flowing process fluid, but suggest a 40-psi (275-kPa) limit for cavitating and multiphase trim flows.

In practice, the trim *KE* is:

$$KE = \frac{1}{2}\rho V^2$$

where ρ equals the fluid density and *V* the trim velocity, *KE* equals the kinetic energy per unit volume and takes the units of pressure; it is also known as the dynamic or velocity pressure.[3]

The main strengths of the *KE* criteria are their ease of calculation, roots in field experience and universality. Unfortunately, these strengths also mirror the major weakness of the *KE* criteria: Such a broad metric selected simply cannot adequately model the wide range of physical phenomena responsible for flow-related control valve application problems. It is precisely because different flow phenomena depend to varying extents on flow velocity that a blunt quantity, such as trim *KE*, fails as a valid predictor for all control valve application flow problems – one size cannot fit all!

To cite but one example, the *KE* criteria do not take into account the material properties of the valve components subject to cavitation, flashing or impacting particulate attack. Thus, the *KE* model would recommend the same kinetic energy limits for trim components made of Stellite 6, carbon steel or even aluminum. In reality, Stellite 6 shows far superior resistance to cavitation damage than carbon steel or aluminum.[4]

This article focuses on the physical phenomena at the root of control valve problems and how to model the phenomena to help application engineers make reliable and safe control valve selections for challenging valve applications. In the following section, we review the key flow phenomena responsible for control valve problems and critically compare existing physics-based models of the fluid phenomena to the *KE* criteria to assess the viability and limits of *KE* limit-based control valve sizing. Ultimately, we recommend models to predict the onset of potentially troublesome valve flow phenomena and enable proper valve selection and problem-free valve operation.

Models for control valve application problems.

Excessive acoustic noise and component vibration. Excessive acoustic noise can give rise to a host of problems. In more moderate guises, acoustic noise can annoy personnel in the vicinity of the valve, sometimes violating OSHA or local noise ordinances, leading to fines and, in more extreme cases, mandated facility shutdown. As valve noise levels rise, vibration of valve components and the downstream piping increases, shortening component life and compromising instrumentation reliability (e.g., ultrasonic flow meters).

Eventually, noise levels reach the point where they excite such intense vibration that the valve components fail rapidly and/or the downstream piping breaks via fatigue failure.[5] Fortunately, allowable noise levels (80 – 85 dBA) lie well below the threshold of component and piping failure (typically 120 dBA measured at a distance of 1 meter from the downstream piping wall),[6] so that the vibration problem can be adequately managed by addressing the noise problem.

Control valve companies have invested significant effort in developing methods to accurately predict control valve noise from a wide range of valve and trim styles, culminating in the ISA[7] and IEC[8] control valve aerodynamic noise prediction methods. In particular, the IEC method is an internationally-accepted procedure verified by a large test database. A discussion of how control valve noise is generated, propagates downstream of the valve, excites downstream piping and radiates from the piping provides a basis for assessing the ability of *KE* to quantify noise and vibration potential.

The throttling action of control valves creates a high-speed jet in the valve trim, which in turn generates intense turbulent velocity and pressure fluctuations (a.k.a. sound). Magnitude of the sound power scales with the trim flow Mach number, M_j, raised to the 6.6th-power:[8]

$$SPL \propto mU_j^{\,2} M_j^{\,3.6} \propto M_j^{\,6.6}$$

where *m* denotes the mass flowrate and U_j the trim velocity. In addition, the sound also has a frequency component. The jetting action creates sound at a variety of frequencies, with peak energy production at the peak frequency, f_p, which scales linearly with the trim velocity and inversely with the trim effective jet diameter, d_j:

$$f_p \propto \frac{U_j}{d_j}$$

Both components play a crucial role in the degree of component or pipe excitation, since the power quantifies the quantity of energy available to excite the component, while the relative coincidence of the sound and component natural frequencies determines the degree of excitation (vibration) that actually occurs. For example, a small amount of acoustical energy at a frequency coincident with the downstream piping can generate very high levels of piping vibration (and hence, sound radiation). Whereas sound occurring at a frequency an order of magnitude higher than the natural frequency of the downstream piping will require approximately 10 times more sound power to create the same excitation.

Sound waves produced by flow in the valve trim propagate into the downstream piping. High flow Mach numbers downstream of the valve can generate additional noise, particularly if the valve of diameter, D_V, expands into a larger diameter downstream pipe. A recent draft of the IEC control valve noise standard (IEC, 1999) builds on the initial document[8] to include a method to calculate expander noise. Expander noise occurs when high-velocity fluid (M_V>~0.3) exits from the valve into larger diameter downstream piping, *Dp* creating a second high-speed turbulent jet with turbulent velocity and pressure fluctuations. The modified jet model results in a dipole-dominated expander noise sound power level, SPL_E, proportional to the 6th-power of the valve exit Mach number:

$$SPL_E \propto M_v^6$$

at a peak frequency, f_V:

$$f_v \propto \frac{U_v}{D_v}$$

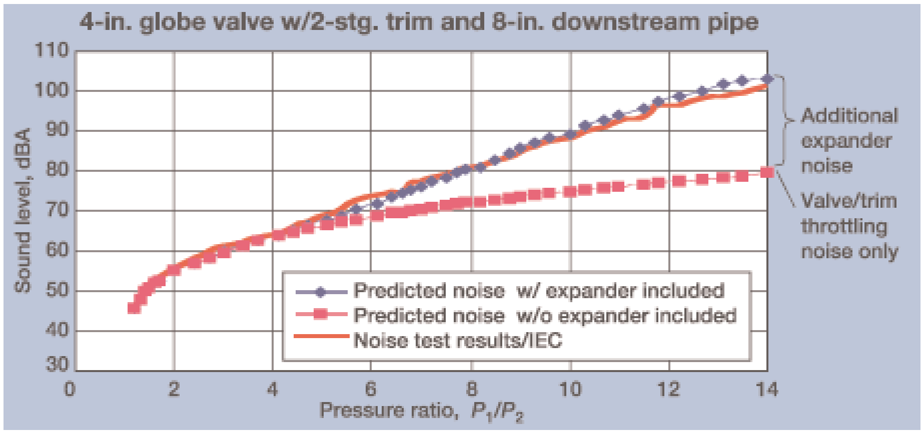

Fig. 1. Calculating expander noise is important, particularly at higher upstream-to-downstream pressure ratios.

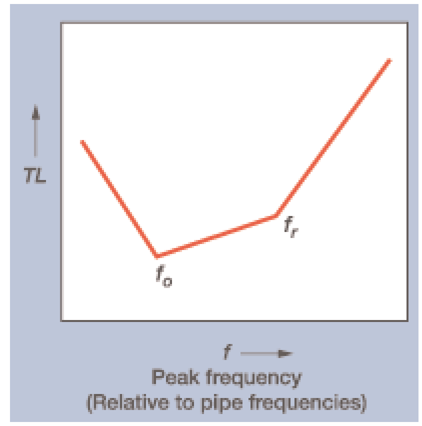

Fig. 2. The scaling relationships for transmission loss reflect the distinct excitation regimes.

In practice, expander noise can overwhelm valve trim noise[9,10] and negate any noise reductions afforded by a low-noise valve. Fig. 1[10] clearly demonstrates the importance of calculating expander noise, particularly at higher upstream-to-downstream pressure ratios. In this instance, neglecting expander noise at a pressure ratio of 12:1 leads to an underprediction of valve noise by almost 20 dBA, causing the valve to not fulfill the typical 85-dBA requirement and potentially leading to a host of more serious valve and piping vibration problems.

Sound inside the pipe excites (i.e., causes to vibrate) the pipe wall to varying extents, depending on the relationship between the sound peak frequencies and the coincidence frequency of the piping, f_o, and the pipe ring frequency, f_r:[8]

$$f_r = \frac{\pi D_p}{c_p}$$

$$f_o = \frac{f_r}{4}\left(\frac{c_2}{c_{air}}\right)$$

where c_p represents the sound speed of the downstream piping material (e.g., steels typically ≈ 5,000 m/s) and c_2 the sound speed of the fluid in the downstream pipe. The sound realizes maximal pipe excitation when the peak frequency matches the first pipe coincidence frequency, $f_p = f_o$. Or, as expressed in Fig. 2, the minimum transmission loss, T_L, for sound propagation from the gas inside the pipe to the gas outside of the pipe wall occurs at $f_p = f_o$. The scaling relationships for transmission loss reflect the distinct excitation regimes identified in Fig. 2.

$$f_p < f_o \quad \Delta TL \propto 20\log_{10}\left(\frac{f_o}{f_p}\right) + 13\log_{10}\left(\frac{f_o}{f_r}\right) \quad \text{(Typically Expanders, Standard Valves)}$$

$$f_o < f_p < f_r \quad \Delta TL \propto 13\log_{10}\left(\frac{f_p}{f_r}\right) \quad \text{(Typically Standard Valves)}$$

$$f_r < f_p \quad \Delta TL \propto 20\log_{10}\left(\frac{f_p}{f_r}\right) \quad \text{(Typically Low-noise Trims)}$$

Besides being considered generally poor piping design,[11] high Mach number flows in the downstream piping have been demonstrated to effectively decrease transmission loss by a factor, L_g:[12,13]

$$L_g = 16\,\mathrm{Log}_{10}\left(\frac{1}{1 - M_2}\right)$$

The *SPL* is modified to reflect the A-weighted *SPL* by adding 5 dBA.* Ultimately, the vibrating downstream piping radiates aerodynamic noise to the surrounding air in a manner consistent with cylindrical spreading models, with the resulting *SPL* at a distance of 1 meter from the pipe wall, $Lp_{Ae,1m}$:

$$\mathrm{Lp}_{Ae,1m} = \mathrm{Lp}_{Ae} - 10\,\mathrm{Log}_{10}\left(\frac{D_p + 2t_p + 2}{D_p}\right)$$

The IEC (and ISA) aerodynamic noise prediction method rescued valve end-users from a prior noise prediction quagmire, where different valve manufacturers' noise prediction methods varied by up to 30 dBA from each other for similar valves.[14] The method has stood up to years of testing and has been verified for a wide range of valve sizes and styles, ranging from butterfly to drilled-hole and tortuous path valves. If used by valve manufacturers, this physics-based methodology ensures accurate noise predictions.

In contrast to the physics-based IEC aerodynamic noise prediction method, a *KE* criterion cannot hope to provide an accurate prediction of control valve noise. *KE* does not include a physical model for the noise source (turbulent pressure fluctuations), nor for transmission of the sound through the pipe walls. Moreover, it does not address valve expander noise. In sum, the IEC method is a far more accurate and versatile tool for assessing valve aerodynamic noise than trim *KE,* and enables selection of the most economical valve to fulfill the application noise requirements.

Excessive trim and piping vibration. High levels of valve plug, cage and downstream piping vibration can have adverse consequences for control valves and process equipment. Moderately high levels of valve plug and cage vibration can damage or loosen the components, decreasing the valve's ability to effectively control an industrial process. Significantly stronger trim vibration can break trim components, generating metal fragments that flow into the downstream piping and process equipment; these metal

* The 5 dB reflects an average correction to account for all sound frequency peaks (IEC, 1999).

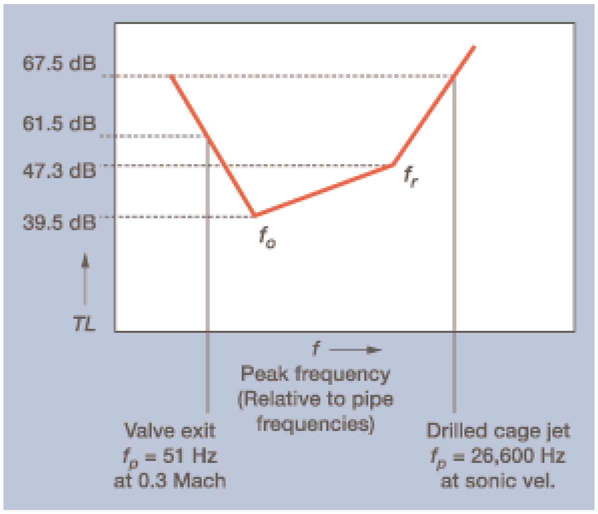

Fig. 3. Due to the relatively large valve exit diameters, expander noise occurs at relatively low frequency.

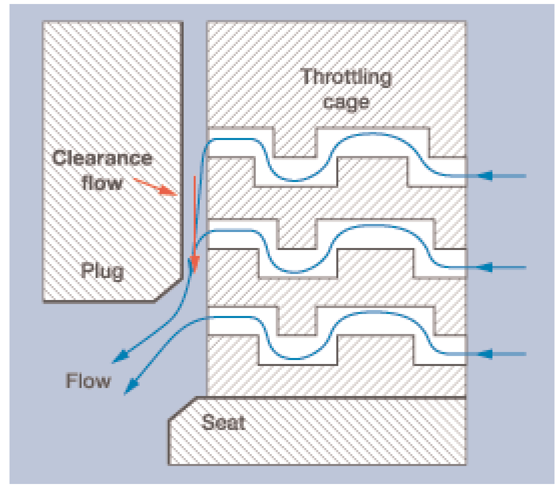

Fig. 4. If clearance between the plug and guiding surface is sufficiently large, the clearance flow path will eventually pose less resistance to flow than the exposed flow paths.

shards would likely destroy a turbine, compressor, pump or heat exchanger downstream of the valve.

In extreme cases, downstream pipe vibration can even lead to downstream piping failure via acoustic pipe fatigue.[5] Smaller, welded line connections to the piping are particularly vulnerable to high-amplitude vibrations in the downstream piping.[15]

As noted in the aerodynamic noise discussion, piping vibration results from high-energy pressure fluctuations interacting with downstream piping and exciting the piping. The same basic principle applies to valve component vibration: ability of a pressure fluctuation to excite a structure depends on fluctuation amplitude and the relationship between the fluctuation frequency and the structure's natural frequencies. If fluctuation frequency coincides with a natural frequency of the structure, the fluctuation readily causes the structure to vibrate. On the other hand, a structure is far less susceptible to excitation by fluctuations removed from the structure's natural frequency.

Fundamentally, the quantity of *acoustic energy* generated by the flow provides realistic estimates of the energy available to excite *component vibration,* because both aerodynamic noise and component vibration energy are generated by the same physical mechanism: turbulent pressure fluctuations in the fluid field. Thus, the IEC aerodynamic noise prediction method enables one to evaluate potential for trim and valve vibration more directly and more accurately than a rough *KE* criterion. A similar IEC standard[16] yields estimates for hydrodynamic noise, with analogous relevance to vibration potential in liquid valve applications. Our experiences confirm this result: We have found that operating a control valve below 95 dBA will prevent onset of vibration problems. Since most valve applications require sound pressure levels below 85 dBA, valves satisfying this aerodynamic noise requirement will avoid vibration problems.

In contrast, *KE* provides a very rough estimate of the energy available to excite valve components. *KE* scales with the velocity squared, whereas actual pressure fluctuation magnitude scales with trim velocity raised to the 6.6th-power, or valve exit velocity to the 6th-power (for expander noise). Very importantly, *KE* criteria fail to even address the degree to which the flow energy excites any sort of valve component or downstream piping.

Moreover, Miller[2] applies the *KE* criteria only to the valve throttling area, but not the valve outlet. Thus, the *KE* criteria ignore a most dangerous form of valve vibration: vibration created by expander noise. As discussed earlier, pressure fluctuations at frequencies close to the valve component natural (or resonant) frequency readily induce significant vibration levels in a component; component response to pressure fluctuations decreases rapidly at excitation frequencies further from the component resonant frequency.

Fig. 3 displays a plot of sound transmission losses as a function of frequency, showing the f_p values for air flow through a drilled-hole cage (hole diameter of 3.2 mm) and out of a 400-mm valve (16-in.), expanding into a 600-mm (24-in.) downstream pipe with characteristic frequencies f_o and f_r. Due to the relatively large valve exit diameters, expander noise occurs at a relatively low frequency that lies much closer to f_o, the frequency at which maximum piping vibration occurs. In other words, expander noise will more readily excite vibration in downstream piping than the drilled-hole cage trim noise.

Again, the same basic physical processes apply to all valves. As a result, with proper valve body and piping sizing (i.e., managing velocities in these areas), identical trim flow paths will provide the same basic noise performance (under the same flow conditions) independent of valve size. For example, each hole in a 6-in. valve with 0.25-in. drilled-hole trim will, under the same upstream and downstream conditions, produce the same noise as each hole in a 16-in. valve with 0.25-in. drilled-hole trim (assuming that valve body and downstream Mach numbers are reasonable and hole spacing is consistent).

In practice, valves with small-hole trim operate at sonic velocities in a range of applications and still result in acceptable noise and vibration levels, providing many years of trouble-free operation. It is not unusual to operate cage-style globe valves outfitted with a trim with multiple, small drilled-holes spaced properly, at pressure ratios greater than 3-to-1, resulting in sonic trim exit velocities and downstream noise levels of less than 70 dBA. Typical examples include gas recirculation (compressor bypass), steam dump and vent-to-atmosphere.

Liquid flow-induced vibration. High-pressure liquid flows can also cause plug vibration problems, usually resulting from a combination of throttling at very low lift points (very close to the seat ring) and excessive clearance between the plug and the cage guiding surface. If clearance between the plug and guiding surface (typically a cage) is sufficiently large, the clearance flow path will eventually pose less resistance to flow than the exposed flow paths, effectively short-circuiting the intended flow through the torturous path (Fig. 4). As a result, the bulk of the application pressure drop then occurs across the plug-cage clearance, generating unintended and excessive liquid flow velocities that result in local cavitation and plug vibration.

Once it begins, the problem only gets worse, since the vibration and cavitation erode the plug-guide interface, opening up even greater clearances and exacerbating the existing cavitation and vibration, eventually leading to plug rotational instability and valve stem failure. In general, the relatively low capacity of tortuous-path radial-flow trim designs renders them more prone to such problems, particularly if the application requires extensive staging and calls for very high rangeability. Proper valve trim design and manufacturing, application of hard trim guide surfaces as well as limitations on minimum allowable throttling flow levels can all help to mitigate or resolve this problem. As with vibration, the *KE* criteria fail to address liquid flow-induced vibration because they again do not model the source and magnitude of the problem and nature of the valve plug-guide surface clearance flow.

Valve erosion caused by droplet and particle impaction and cavitation. The term "erosion" comes from the Greek word *erodere* – literally, to gnaw away. At lower levels, erosion can damage seating surfaces, compromising valve shutoff and control performance. Severe erosion can cause component failure or even breaching of the valve or pipe pressure boundary in exceptional instances. Many flow phenomena can cause erosion, including flow cavitation, flow unsteadiness and particle / droplet surface impaction. They all share the basic damage mechanism: a very brief and sharp overpressure that plastically deforms the surface material, steadily weakens the affected surface over time and leads to surface material removal. First, we will explore the basic mechanism in more detail and then discuss each particular phenomenon responsible for erosion.

Damage mechanism. Flow parallel to the surface (shear) and steady-state fluid impingement produce very low surface pressures and stresses[17] that lie well below valve material yield strengths. For example, a steady 152 m/s (500 ft/s) water jet produces stagnation pressures of ≈ 11.5 MPa (1.67 kpsi). This is well below the yield strength of 316 stainless steel (≈ 205 MPa, or ≈ 30 kpsi).[18] On the other hand, *unsteady* impinging flow produce much greater surface forces. When a particle or droplet impacts a surface, or a vapor bubble collapses on a surface, the pressures rise precipitously and create very high, localized material stress; an unsteady liquid jet can generate similar transient pressures and stresses.

Several investigators have successfully applied a "water hammer" model to quantify the surface forces generated by liquid droplet impaction and transient jet impingement. Fundamentally, P_{wh} reflects the pressure of the pressure wave created when the unsteady fluid volume impacts the surface. The "water hammer" model[19] posits that surface pressure, P_{wh}, varies as a function of the fluid density, ρ, fluid sound speed, *c*, and fluid velocity normal to the surface, U_N,

$$P_{wh} = \rho c U_N \left(1 + 2\frac{U_N}{c}\right)$$

In practice, fluid velocity for liquid rarely approaches the sound speed of water (≈1.497 m/s, or 4,915 ft/s at 25°C)[20]

$$P_{wh} \cong \rho c U_N$$

and the water hammer equation simplifies to:

The previous expression also yields surface pressure estimates for particle or droplet impaction.[21] In that scenario, *c* represents the particle or droplet material sound speed and U_N the particle / droplet velocity at impact. We now examine the different erosion mechanisms in the context of the water hammer model.

Erosion by clean valve flows. As cited earlier, shear flows cannot generate surface forces sufficient to yield typical valve material surfaces, but flow unsteadiness can generate much higher surface forces via the water hammer effect. In a global sense, valve flows can be considered steady, but the turbulent, separating flow field inside of valves produces an unsteady flow situation at most locations within a valve. The flow velocity fluctuations typically scale with the local velocity, suggesting that, for example, flow unsteadiness in the valve trim should scale with the trim velocity. Because magnitude of actual velocity fluctuation is usually significantly less than magnitude of the mean velocity, the water hammer model – using the local velocity – should produce conservative estimates for clean valve flow erosion potential.

Thiruvengadam[22] conducted experiments to study the correlation between water hammer surface pressures, surface material yield strength and onset of surface erosion. He found that surface damage began when the water hammer pressure equals one-third to one-half the material fatigue strength for a given number of cycles. Applying this information, a conservative valve design would ensure that flow at key points in the valve, such as the trim, seat ring and valve exit, do not approach the velocities needed to generate surface pressure water hammer pressures equal to one-third of the material endurance limit. Logic suggests that the same methodology and result should hold for gaseous flows as well.

Erosion by particulate-laden valve flows. Humphrey[23] authored a review surveying the role of fluid flow in particle impact-driven erosion. Particles with very little inertia (i.e., very little mass and/or velocity) relative to the fluid flow tend to closely follow the fluid motions and have minimal erosive potential, e.g., 10-μm catalyst fines in refining fluids. In contrast, highly inertial, larger particles cannot readily track the fluid motions and are prone to ballistic trajectories and surface impaction. Examples of the second, erosive scenario include 200-μm sand particles in a gas let-down valve and 500-μm oil droplets in a wellhead application.

The same formulation of the water hammer model also applies to particulate and droplet impacts because the mechanics are the same: colliding objects produce a pressure wave in both the particle (or droplet) and surface, generating intense, transient pressures. In this instance, the water hammer model would employ the impacting particle material sound speed for *c* and the impact velocity for U_N. Assuming that the particle / droplet follows a fully ballistic trajectory to the surface, i.e. that it moves at the maximum local velocity and does not slow down before striking the surface, ensures a conservative estimate of impact pressures for assessing damage potential.

In fact, results of Engel (from Springer[21]) show that surface pressures created by an impacting droplet equal roughly one-half those predicted by water hammer formulation, lending credence to this approach. In an application scenario, the engineer would ensure that the maximum flow velocity in the valve, typically at the throttling cross-section, lies below the threshold "water hammer" velocity calculated per the method outlined previously.

Erosion by droplets in flashing flows. Flashing denotes the permanent transition of a portion of the liquid flow at the valve inlet to a vapor flow at some point in the valve, owing to a downstream process condition (temperature and pressure) that resides in the gaseous or two-phase thermodynamic regimes; of themselves, "high fluid velocities" cannot cause flashing. Damage occurs when liquid droplets that fail to complete the transition from liquid to the gas phase impact and erode a valve surface.

Ideally, all of the liquid would transition from the liquid to the gaseous phase at the same point in the valve. In reality, inhomogeneities in the flow field ensure that this does not happen. Furthermore, phase transitions do not happen instantaneously but at some finite rate. The net result is that a flashing valve flow can contain a substantial mass-fraction of liquid droplets that impact on the downstream portion of the valve. For instance, deaerator level control valve applications can suffer extensive erosion damage from high-velocity droplets remaining in the flashing flow.

To combat potential damage from these flows, valve manufacturers have developed valve designs such as angle valves with expanded outlets, reducing potential for droplet impaction, as well as lining the valve outlet with sacrificial or more rugged material. To resist particle impact erosion, designs such as the sweep angle valve (Fig. 5), with streamlined flow paths, have also been used with considerable success.

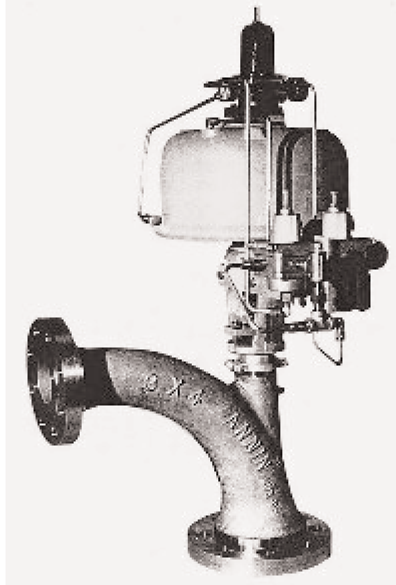

Fig. 5. Designs such as the sweep angle valve resist particle impact erosion.

Because of the unsteady nature of the droplet formation process and the associated uncertainties in the quantity, size, velocity and droplet location, quantifying erosion by droplets formed in flashing flows is more complex than the particulate case. A reasonable and conservative approach to protect the valve from droplets remaining in the flashing flow is to make the same assumption introduced with the particle / droplet impaction scenario – that is, that the droplets are fully ballistic – and to calculate water hammer pressures under these conditions. As noted earlier, this should yield a conservative result, because the flashing particles remain from a point in the valve where a greater portion of the fluid was a liquid, and the primary phase flow velocity was substantially less than the downstream velocities.

This implies that all droplets should impact at lower velocities than the local velocity. Ballistic droplets, by definition, are not affected appreciably by the flow and, hence, remain at the lower, upstream velocity, while smaller droplets, which track the flow more closely, can decelerate to some extent before impact. To prevent erosion in a valve with flashing flow, the application engineer needs to ensure that the velocity in the relevant part of the valve, e.g., exiting from the trim and the valve, lies below the threshold "water hammer" velocity calculated per the method outlined previously.

Comparison of water hammer erosion model and KE criteria. Table 1 presents the water hammer model results when applied to clean air and water flows – i.e., without droplets and particles – operating at the upper bound of a proposed *KE* criterion for a valve with hardened CA6NM trim material. The table directly compares the predicted surface overpressures at the maximum flow velocities allowed by the *KE* criteria[2] with the conservative one-third endurance limit result found by Thiruvengadam.[22]

The air-flow case demonstrates that the *KE* criterion results in surface overpressures less than 5% of the damage threshold, suggesting that the *KE* criterion has little, if any, relevance at all to the erosive potential of a clean gas flow. It also reinforces the impression that the *KE* limitations may have evolved from experiences with liquid flows.

At the posited *KE* velocity limit, the water flow case generates transient surface overpressures less than one-half the estimated pressure level required to begin damaging the trim. In this instance, the *KE* criterion is shown to be quite conservative, and could lead to selection of a more costly valve design than dictated by the flow physics.

Although the *KE* criteria appear to recommend overly conservative valve velocities for clean liquid flows, they tend to admit overly aggressive solutions for flows with entrained droplets or particles. Table 2 displays the water hammer surface pressures calculated using the *KE* limitations advocated earlier.[2]

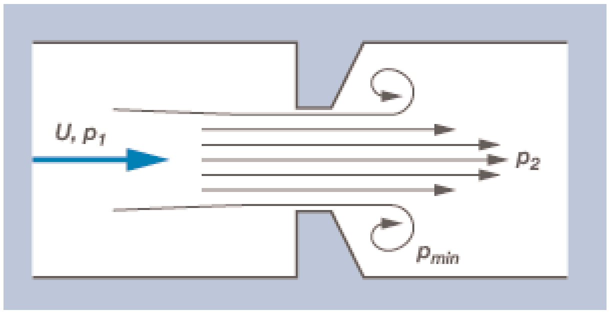

Fig. 6. In control valves, cavitation typically occurs near or at the exit of the valve trim.

Table 1. Comparison of water hammer erosion model and KE velocity criteria: clean air and liquid flows

Physical quantity	Air flow	Water flow
(All values calculated at 25°C unless noted)		
Valve outlet pressure, MPa	2.0	2.0
Flow *KE* criterion, kPa	483	483
Impacting material density, kg/m^3	24.1	998
KE criterion maximum flow velocity, m/s	200	31
Fluid sound speed, m/s	343	1,460
Water hammer pressure, MPa at max. allowable *KE* velocity criterion	3.6	47
CA6NM, 1/3 endurance limit, MPa, allowed[22]	108	108
KE criterion/$\sigma_{1/3}$, %	3.3	43.5

In striking contrast to the clean-flow results, the *KE* criteria-defined operational limits generate surface pressures *2 to 15 times greater* than the damage threshold level described by Thiruvengadam.[22] Moreover, the *KE* criteria allow much higher particulate-laden air flows than water flows. In reality, particles in air flows have significantly more damage potential than water flows at the same velocity because the particles have a greater tendency to follow the denser, more viscous water flows than the lighter, less viscous air flows. Hence, particles in air will attain more ballistic trajectories, impact surfaces at higher velocities, create higher surface pressures and damage the affected valve surface to a greater degree than the same particles in a water flow (at the same velocity as the air flow).

The particulate and droplet cases clearly point out that the *KE* criteria do not adequately model the physics of a potentially erosive valve flow, or, as shown previously, simply get the physics wrong. One could argue that the *KE* criteria would yield a credible result for a different valve trim material, e.g., Stellite 6 hard-facing. However, this points out another very basic flaw with the *KE* criteria: They do not take into account the properties of the valve material! To the *KE* models, aluminum provides the same erosion resistance as Stellite 6. A useful model for erosion damage must take into account the surface forces created at particle / droplet-surface impact, as well as the ability of the surface to resist the impact forces. The *KE* criteria do neither.

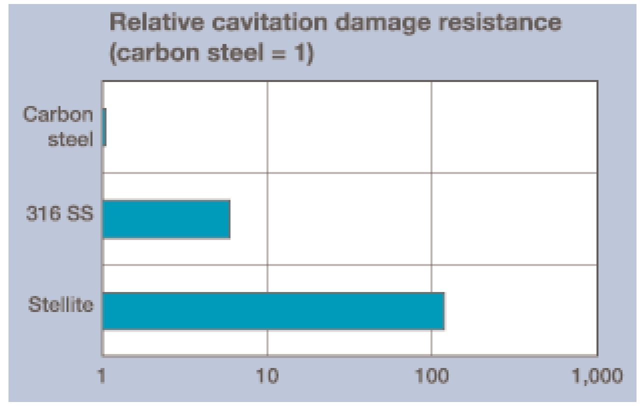

Fig. 7. Stellite is about 150 times more resistant to cavitation damage than carbon steel.

Thus, they provide minimal guidance to the control valve application engineer as to appropriate valve velocities or valve materials in erosive applications.

Corrosion of valve and/or valve trim material can appreciably weaken the relevant surfaces and will tend to, if present, exacerbate control valve erosion problems. Even worse, corrosion may create an erosion problem where one did not exist before, for example, in stainless steels. Although the surface stresses generated by the process liquid and/or particles lie below the "water-hammer" threshold and should not lead to surface damage, rapid removal of protective oxide layers from stainless steels by the flow or particles accelerates corrosion / erosion material removal. To avoid erosion / corrosion problems in corrosive applications, we advocate selecting a material that will resist the corrosion threat, then apply the water hammer model to ensure that the valve design satisfies the material erosion limits.

Cavitation erosion. Cavitation onset occurs when the pressure at some point in a liquid drops below its vapor pressure (at the process condition), causing a portion of the liquid to change phase into vapor. If the pressure recovers above the process fluid's vapor pressure, the vapor pocket (bubble) implosively collapses. This can lead to surface damage depending on location and strength of the collapse. In control valves, this process typically occurs near or at the exit of the valve trim, where flow velocities are high and low pressures arise in low-pressure separated flow regions (Fig. 6).

Although both droplet / particle impaction and cavitation damage surfaces via overpressures, the mechanism of cavitation damage / erosion fundamentally differs from the impaction mechanism. An impacting particle or droplet generates high surface stresses through a pressure wave that occurs at impaction. In cavitating flows, shock waves or "reentrant" liquid microjets created during very rapid vapor bubble collapse generates very high, localized pressures.[4] If the bubble collapse occurs very near or on a surface, the very high pressures generate stress and plastically deform the surface. Eventually, enough cavitation events will cause the surface to fail, leading to a "pitted" surface profile indicative of cavitation damage. The more frequent the cavitation events, the more quickly surface damage develops.

Table 2. Comparison of water-hammer erosion model and *KE* velocity criteria: article/droplet-laden flows

Physical quantity (All values calculated at 25°C unless noted)	Water droplet (in air)	Sand particle (in air)	Sand particle (in H_2O)	Flashing water vapor *
Downstream valve pressure, MPa (absolute)	2.0	2.0	2.0	1.91
Flow *KE* criterion, kPa	483	276	276	276
Impacting material density, kg/m^3	998	2,600	2,600	853
KE criterion maximum flow velocity, m/s	151	151	23.5	240
Impacting material sound speed, m/s	1,497	3,810	3,810	1,053*
Water-hammer pressure, MPa, at max allowable *KE* velocity	272	1,618	236	314
CA6NM, 1/3 endurance limit, MPa, allowed[22]	108	108	108	108
KE criterion/$\sigma_{1/3}$, %	251	1,498	218	291

* Saturated water vapor at 210°C (410°F), 1.91 MPa (276 psi).[24]

Similarly, more vigorous cavitation accelerates surface damage and increases the extent of the damaged surface area.

The propensity, intensity and damage potential for cavitation under a given process condition depend on a wide variety of flow factors, including flow field topology (valve geometry), local flow velocity, turbulence levels, flow unsteadiness and the application pressures. Beyond the flow field, the surface material plays a critical role in assessing rate of cavitation damage. Generally, once cavitation begins near a surface, it will eventually damage the surface. Tougher metals can dramatically slow the damage rate, but they, too, will eventually yield.

As displayed in Fig. 7, 316 stainless steel is about six times more resistant to cavtiation damage than carbon steel, while Stellite 6 possesses approximately *150 times* the cavitation damage resistance of carbon steel.[4] In addition, the nature of the process fluid plays a measurable, yet secondary, role in damage potential, since liquids with higher surface tensions, such as water, tend to generate more vigorous bubble collapse than fluids with lower surface tensions (hydrocarbons, multi-component fluids).[4] Fluid temperature also affects cavitation damage potential, primarily because material properties vary as a function of temperature. Finally, cavitation can also accelerate existing corrosion mechanisms as outlined in the prior erosion / corrosion discussion, reinforcing need for corrosion-resistant materials in corrosive applications with cavitation potential.

Clearly, because of the wide range and sheer quantities of variables influencing cavitation damage potential, there is no simple, universal calculation method for theoretically predicting cavitation damage levels. Consequently, until 1995, cavitation damage thresholds were not clearly defined and valve manufacturers applied their own criteria based on their past testing and field experience. To improve the situation, the ISA set out to develop a reliable method for assessing control valve cavitation.[25] The "sigma" model for valve cavitation inception and damage begins with a simple parameter, σ, which reflects the ratio of cavitation resistance to cavitation potential, where p_1 and p_2 denote the valve upstream and downstream pressures, and p_v the process fluid vapor pressure at the process temperature:

$$\sigma = \frac{p_1 - p_v}{p_1 - p_2}$$

To clarify, a sigma value is calculated for the *application conditions*. As expected, the smaller the σ value, the greater the cavitation potential of a flow condition in a valve application.

The ISA Recommended Practice identifies three cavitation regimes (Fig. 8), that are determined experimentally for each distinct valve geometry. The incipient cavitation index, σ_i, specifies the application condition at which cavitation can first be detected via an accelerometer or hydrophone with high-frequency sensitivity (5 – 50 kHz). Incipient cavitation represents very infrequent bubble formation and collapse events, likely caused by turbulent and unsteady pressure fluctuations. As cavitation frequency and intensity increases, lower-frequency cavitation events become more common and the flow enters the constant cavitation regime, σ_c. Process conditions with lower sigma values continue to increase the cavitation and vibration measurement intensity, until they reach σ_{mv}, the maximum vibration index.

At yet lower sigma values, vibration levels actually decrease, as the low-pressure regions and vapor bubbles grow in size, decreasing bubble collapse rate. The preponderance of bubbles cushions the collapse of other bubbles, further mitigating cavitation intensity. Crucially, the sigma practice calls for the manufacturer to carry out damage testing to determine σ_{mr}, the cavitation index below which the valve should not be applied.

As shown in Fig. 8, σ_{mr} typically lies between σ_c and σ_{mv}; the precise value depends upon the valve style. For instance, a butterfly valve can endure only minimal cavitation levels before suffering extensive damage. Therefore, σ_{mr} of a high pressure-recovery ball valve approaches to σ_i, i.e., the cavitation inception point.

In contrast, a globe valve with trim consisting of multiple small flow passages and operated flow-to-close is designed and applied such that vapor bubbles produced in the trim collapse in the throat of the valve, away from critical trim surfaces. This enables the valve to tolerate much higher cavitation levels without appreciable damage. The acceptable cavitation level for valves with multistage tortuous-path anticavitation trim can approach σ_{ch}, the point of

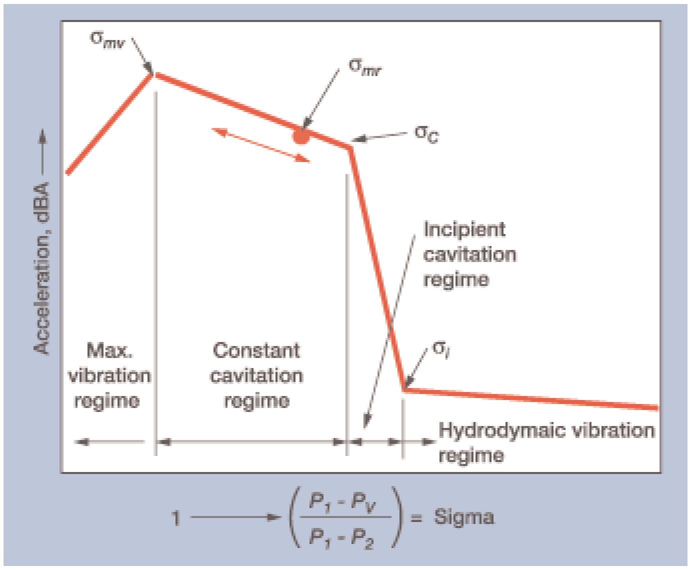

Fig. 8, σ_{mr} typically lies between σ_C and σ_{mv}

choked flow! Again, laboratory testing establishes acceptable cavitation level for each valve and trim combination. In practice, an applications engineer evaluates valve suitability for an application by calculating σ for the application conditions and comparing it to the σ values of the valve. If the application-condition σ lies above the valve (i.e., scaled) σ_{mr}, the valve will provide good anticavitation performance in the application.

The sigma methodology improves markedly on prior methodologies to predict cavitation performance, such as F_L, greatly reducing the sagacity and expertise required to safely apply a valve in a potentially cavitating service. Its formulation effectively combines the basic flow physics with empirical test results to provide a more reliable indication of control valve cavitation that takes into account the distinct flow and pressure fields of each different valve style.

In distinct contrast to the physics-based sigma method, the *KE* criteria fail to address the most fundamental aspect of cavitation: that cavitation only occurs if the pressure in the liquid falls below the liquid vapor pressure. The *KE* criteria do not yield any information about whether the flow will cavitate, let alone the damage potential of the process condition for a given valve. Moreover, the *KE* criteria also neglect most other key factors in successful anticavitation valve applications: valve and flow topology, and material and fluid properties. In sum, the *KE* criteria will not necessarily avoid cavitation damage or may lead to excessively conservative and expensive valve solutions where they are not required. We strongly advocate using the ISA sigma method to evaluate and prevent control valve cavitation damage.

Low-frequency pulsation. Generally, excessive control valve noise or vibration occur at the valve or valve trim outlet, or downstream of the valve. In rare instances, extremely damaging vibration can develop upstream of the valve and cause dramatic and destructive vibration of the upstream piping. Schafbuch, et al.,[26] first noted this phenomenon, known as low-frequency upstream pulsation, in piping upstream of a globe-style valve with a drilled-hole cage installed flow-to-open (flow-under-the-plug).

At lower lift points, it exhibited acceptable vibrations. As the lift increased, upstream vibration levels rose precipitously, particularly when the lift exceeded 70%, and forced valve closure. Surprisingly, the dominant vibration intensities occurred at very low frequencies, 5.9 and 18.1 Hz, far removed from the peak frequency of any conceivable aerodynamic noise source. Ominously, peak vibration frequencies coincided with structural and acoustic modes of the upstream piping. After further study of the low-frequency vibration in the laboratory, it was concluded that the problem arose from an interaction between the flow entering the valve throat and upstream piping.

Specifically, steady-state computational fluid dynamics (CFD) simulations demonstrated that flow entering the valve throat switched from a “closed” (more stable) separation to an “open” (less stable) separation structure at higher lift points. They posit that instability of these structures leads to large, unsteady separated flow structures. These create pressure pulsations that occur at frequencies close enough to the upstream natural frequencies such that they mutually reinforce each other and ultimately coincide at the upstream piping natural frequencies.

Kiuchi[27] uncovered other instances where valves with higher stroke length-to-plug diameter aspect ratios operated flow-to-open developed unacceptable upstream vibration levels. Fluid flow simulations carried out by the first author of this article in a flow-to-open globe valve with a drilled-hole cage trim reinforced the key role that the stroke length-to-plug diameter aspect ratio plays in creating conditions conducive to low-frequency upstream pulsation. He found that a valve at full lift, with a stroke-plug diameter ratio of ≈ 0.375, formed a steady-state closed separation flow structure. This suggests minimal potential for low-frequency pressure pulsations. Indeed, field tests by Kiuchi[27] of the same valve confirmed the absence of large-scale upstream vibration. Simulations for an over-stroked valve (stroke-plug diameter ratio ≈ 0.5) showed a radically different flow field, including a large “open separation” that suggested a distinct potential for low-frequency pressure pulsations.

Short of modifying the upstream piping structure, there exist two reasonable ways of preventing low-frequency pulsation problems in flow-to-open valves. First, flow into the valve throat can be conditioned (straightened) to eliminate open flow separations using a proboscis on the plug tip[26] or a standard drilled-hole diffuser.[27] Second, the engineer can select a valve that will operate at a “small enough” stroke length-to-plug diameter aspect ratio that precludes developing the “open” separated flow structures. In most globe valves, this translates into an aspect ratio of less than 0.45 to 0.7, depending on the valve style.

Needless to say, the *KE* criteria do not address upstream low-frequency pulsation issues.

Not a single standard or recommended practice authored by major technical societies serving the valve industry and their customers (American Society of Mechanical Engineers, International Electrotechnical Commission, Instrument Society of America, Valve

Manufacturer's Association of America or VDMA, i.e., the VMA of Germany) contains *KE* limitations. Instead, we advocate use of the physics-based models to accurately model the flow phenomena that cause control valve problems and to develop reliable, cost-effective control valve solutions.

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