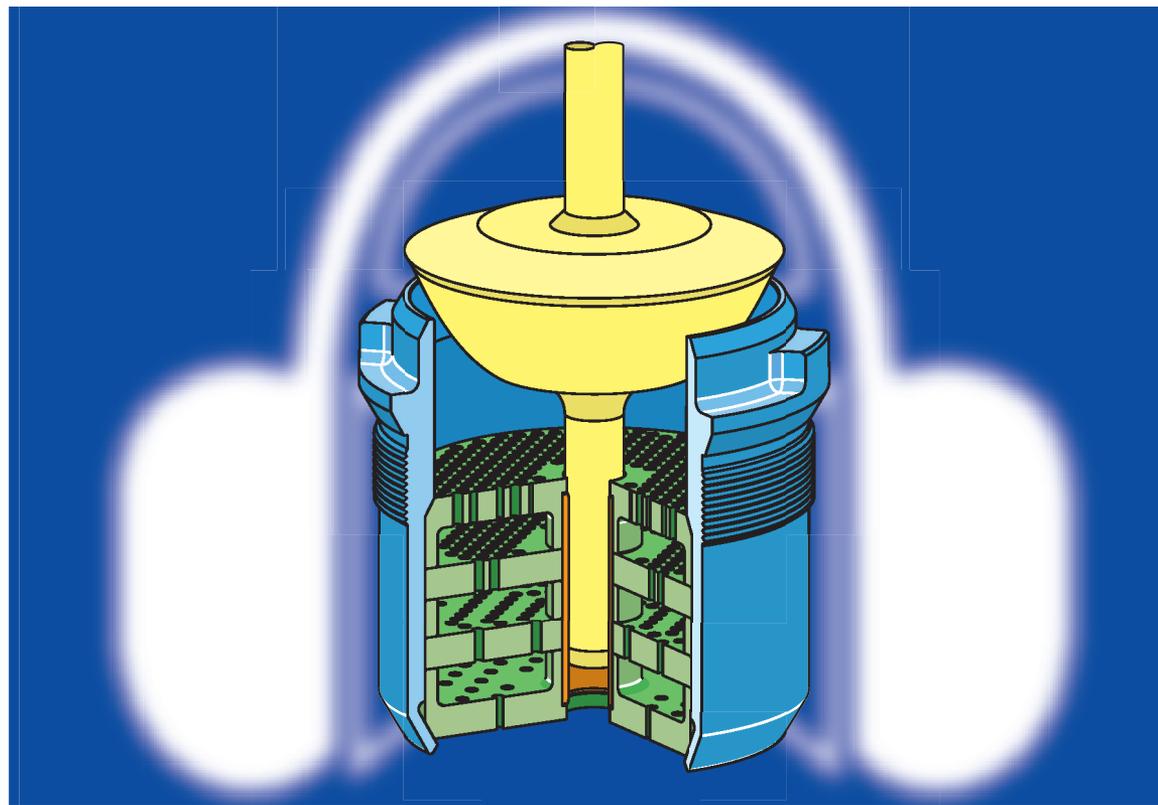


Control valves for critical applications



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Know the causes of cavitation and flashing and how to prevent them

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In refineries, the process media flowing through valves are primarily liquids. With liquids, critical operating conditions caused by cavitation or flashing may occur. Symptoms are, for instance, increased noise emission, valve and pipe component erosion or low-frequency mechanical vibration in the valve and the connected pipeline. Under these conditions, in particular, neglecting details can result in negative influences on plant performance and costs of ownership. Unfortunately, common practice today is to select control valves in a “quick and dirty” fashion, because the phases of planning, bidding and order processing are connected with significant pressures of cost and time.

This article presents the basic principles underlying these problems and shows how to eliminate them based on practical examples from refineries. Moreover, a new throttling element is introduced, that is especially suited to reducing noise emission produced by cavitation. This new throttling element is being implemented in refineries with increasing success.

Typical refinery process conditions. Fig. 1a illustrates normal refinery process data. A typical centrifugal pump with constant speed and normal pressure losses in the hydraulic system is characterized by the following:

Valve inlet pressure, p_1 , decreases as mass flow, W , increases, while valve outlet pressure, p_2 , increases. Vapor pressure, p_v , depends on process medium temperature.

Flow conditions known as cavitation and flashing originate from vaporization occurring in the throttling area. Cavitation occurs when the differential pressure ratio,

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

(according to IEC 60534-8-4^{1,2}), is higher than the valve’s coefficient of incipient cavitation, x_{Fz} . Vapor bubbles are formed in the valve where they soon implode again because $p_2 > p_v$. The cavitation index, σ_I , for incipient cavitation in citation 3 is equivalent to the reciprocal value of x_{Fz} . If p_2 is lower than p_v , flashing occurs and the vaporization continues into the pipeline.

Fig. 1b shows in accordance with Fig. 1a that x_F

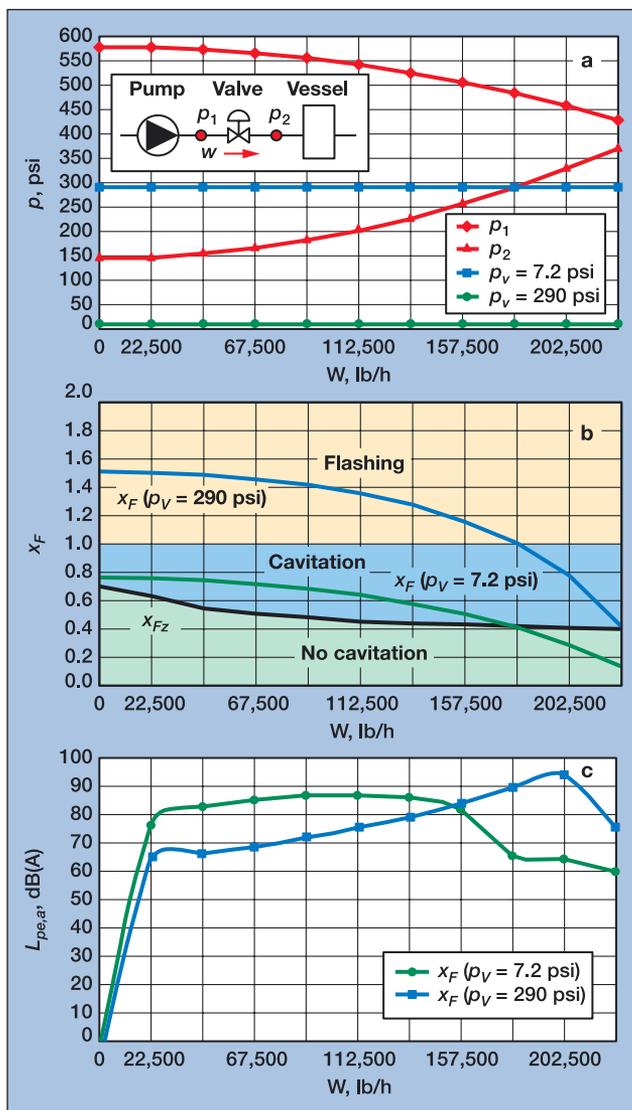


Fig. 1. Typical refinery conditions, process data (a), differential pressure ratio and cavitation/flashing (b) and sound level (c).

generally decreases as W increases. The three different areas of flow status in the valve are marked by the following colors:

- Purely turbulent flow for $x_F < x_{Fz}$ (green area in Fig. 1b)
- Cavitation for $x_F \geq x_{Fz}$ and $x_F < 1$ (light blue area in Fig. 1b)
- Flashing for $x_F \geq 1$ (yellow area in Fig. 1b).

Vapor pressure, p_v , in combination with operating pressures, p_1 and p_2 , as well as the valve coef-

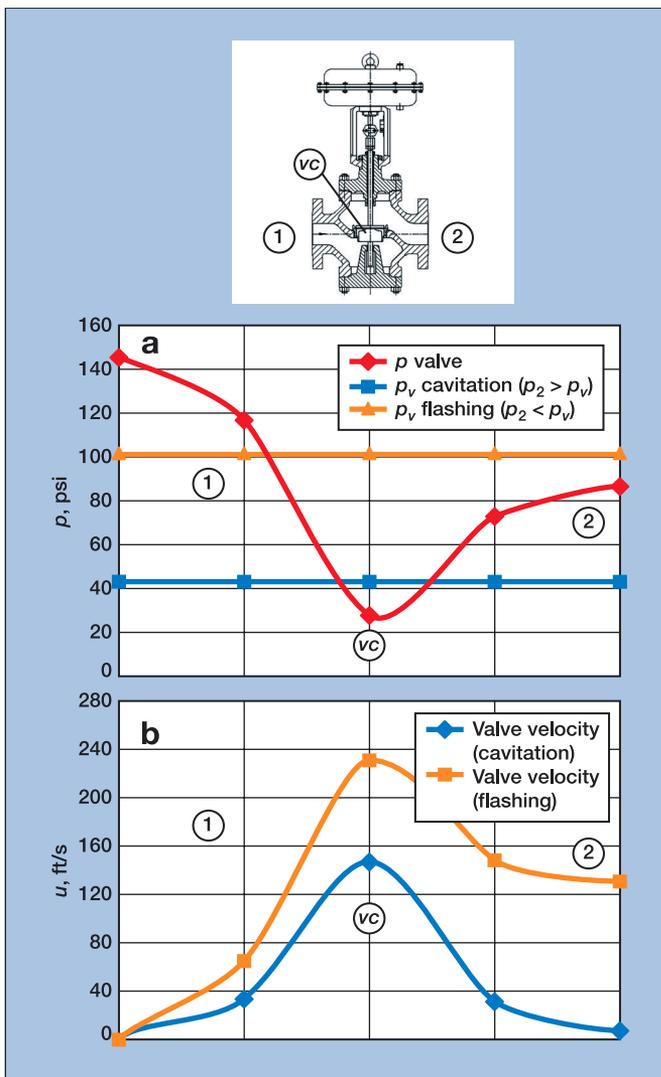


Fig. 2. Pressure and velocity during cavitation and flashing (1: valve inlet, vc: vena contracta and 2: valve outlet).

ficient, x_{Fz} , determine if and when cavitation or flashing occurs. If the vapor pressure is relatively low, cavitation and noise emission will occur in the control valve at small-to-medium flowrates. For the lower vapor pressure value, $p_v = 7.2$ psi, cavitation does not occur between 180,000 and 225,000 lb/h; it does occur, though, when the flowrate is smaller than 180,000 lb/h (Fig. 1b, green x_F curve).

Consequently, flowrates between 22,500 and 145,000 lb/h induce a relatively high sound pressure level, $L_{pe,a}$, i.e., between 80 and 88 dB(A) (Fig. 1c, green $L_{pe,a}$ curve). For the higher vapor pressure value, $p_v = 290$ psi, cavitation occurs when flowrate is in the upper range (Fig. 1b, light blue x_F curve). Flashing occurs below 180,000 lb/h. Since noise emission induced by cavitation is clearly higher than the noise emission induced by flashing,¹ the valve is noisy at flowrates between 155,000 and 225,000 lb/h, reaching decibel levels from 85 to 93 dB(A) (Fig. 1c, light blue $L_{pe,a}$ curve).

To select a control valve suited to a particular application, and to evaluate flow conditions in as much detail as possible, knowledge of the exact process data is an absolute must. This is especially true for the actual pressure conditions downstream of the valve. Extent of



Fig. 3. Visualization of cavitation (a) and flashing (b) with flow coming from below.

the pressure losses in pipes and elbows between the valve and, for example, a vessel at constant pressure, influences valve outlet pressure and, hence, flow status. Either flashing or cavitation can exist when the operating point is near the sharp transition from cavitation to flashing at $x_F = 1$.

Cavitation and flashing. All examination of these phenomena is based on the variation of pressure and flow velocity in the main flow stream of the medium through the control valve (Figs. 2a and b¹).

Cavitation. Since pressure difference at the control valve increases with p_1 held constant, minimum pressure at the throttling area, generally referred to as *vena contracta*, vc , drops to the liquid vapor pressure, p_v ; for $x_F > x_{Fz}$. Because $p_2 > p_v$, vapor bubbles in the area at and around the throttling element collapse in the valve space downstream of the vc . Fig. 3a illustrates cavitation of the flow acting on the valve plug downstream of the vc .

The following effects occur, depending on cavitation intensity:

- ▶ Noise emission due to cavitation with maximum values of 100 dB(A) (Fig. 1c¹)
- ▶ Damage to valve components or pipeline due to

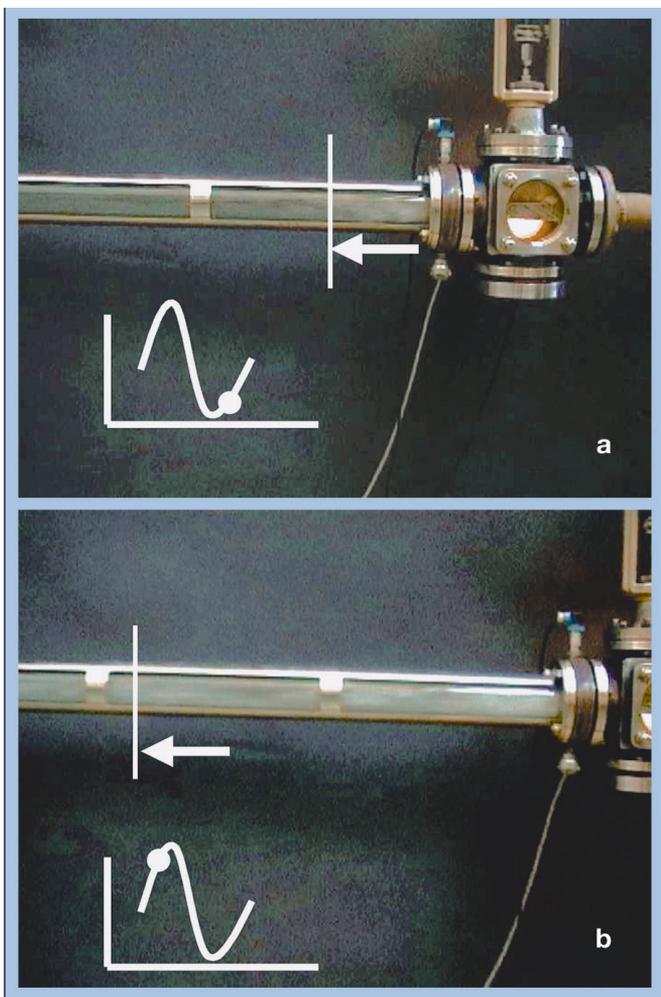


Fig. 4. Propagating shock waves during flashing result in pipeline excitation (a: beginning shock wave, b: ending shock wave).

cavitation erosion (Fig. 5¹)

► Vibrational excitation of the throttling plug and entire valve (Fig. 8)

► Choked flow which must be considered when calculating the C_v value.⁴

Flashing. Cavitation ends and flashing starts when the pressure difference, $p_2 - p_v$, becomes ≤ 0 . Vapor bubbles in the two-phase flow that exit the valve do not implode, but remain intact (Figs. 3b and 4a and b).

The result is a significantly reduced liquid/vapor mixture density on the downstream side of the valve compared to the valve inlet side (which is liquid phase only). This is linked to an increase in average flow velocity, which, in the case of flashing, is a multiple of the valve inlet velocity. In Fig. 2b, for instance, outlet velocity is approx. 130 fps, which is clearly higher than the inlet velocity of 6.6 fps. The higher the average flow velocity at the valve outlet, the higher the imbalance between the liquid and vapor phases. Figs. 4a and b clearly show the propagating shock waves produced by the collapsing bubbles, whereby the entire pipeline system can be excited to mechanical or low-frequency vibration (< 10 Hz).

In process applications with an inlet pressure that is just slightly higher than the vapor pressure, a two-phase liquid and vapor mixture is already present at the valve inlet. As a result, the described effects are even more intensified. Correspondingly, flashing, i.e., true two-

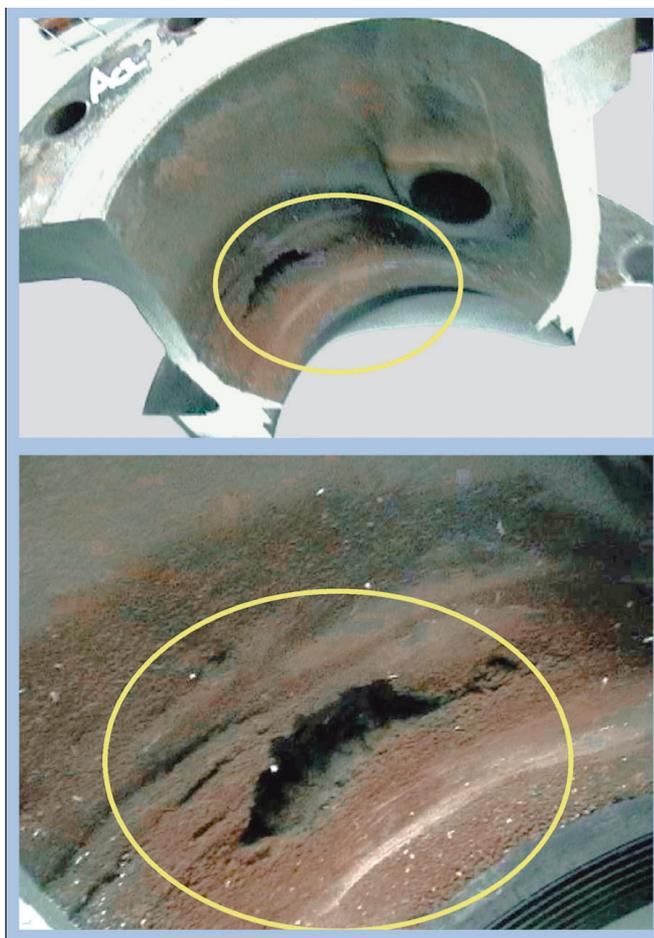


Fig. 5. Erosion in rotary plug valve body induced by cavitation.

phase flow, may also cause great problems such as:

- Noise excitation, but at distinctively lower levels than those caused by intense cavitation (Fig. 1c)
- Damage to valve components and pipelines caused by erosion due to liquid phase droplet impact, which can be further intensified by corrosive medium constituents
- Vibration excitation of the entire pipeline due to shock waves propagating from the valve outlet, caused by the imbalance between the liquid and vapor phases (Figs. 4 a and b)
- Choked flow, and, in addition, substantially reduced flowrate in cases with two-phase flow at the valve inlet.⁵

Measures against cavitation erosion. Erosion due to cavitation (Fig. 5) is most likely to occur when flow is choked due to a high vapor bubble formation rate¹ and when kinetic energy of the vapor bubbles swept away by the cavitating flow (Fig. 3a) through the vc is high enough. The decisive parameters in the case of cavitation are the cavitation index, K_c , value^{6,7} ($=1/\sigma_c$ in citation 3), which corresponds to the differential pressure ratio⁶ at incipient choked flow, and the pressure differential, $p_1 - p_2$, which is directly linked to kinetic energy of the cavitating flow.

In general, outlet velocity should be below 7 fps and low-vibration valve plug designs should be used. Depending on valve trim design and material, a certain critical

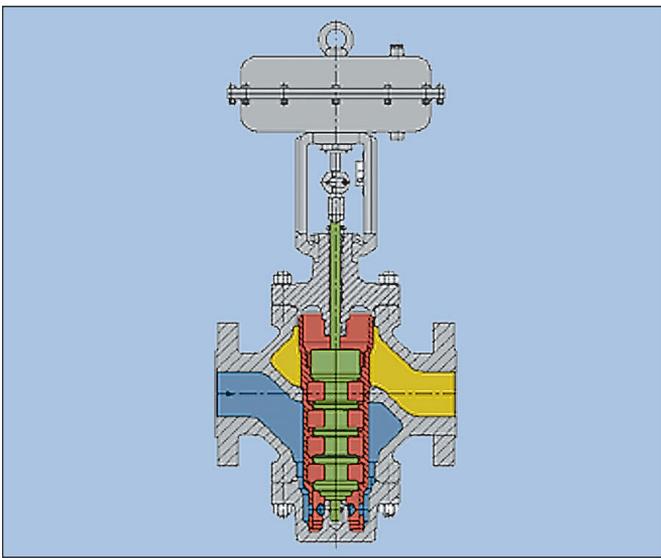


Fig. 6. Multistage axial plug prevents cavitation erosion.

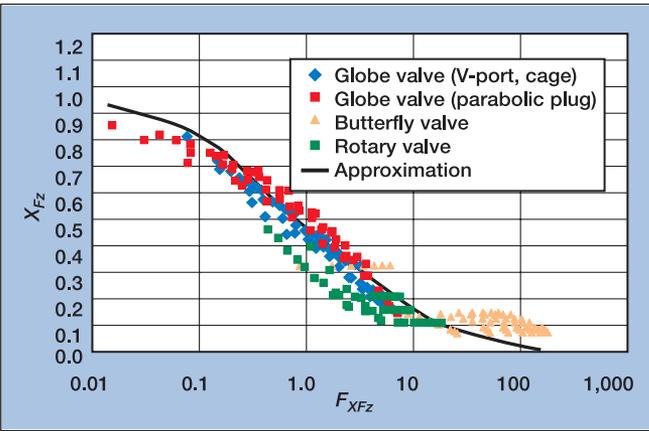


Fig. 7. X_{Fz} as a function of $F_{XFz} = 0.93 \cdot \sqrt{1/F_L C_v} \cdot F_d$

differential pressure value for significant cavitation damage, $p_{crit,cav}$, is allowable. Based on this, the following empirically gained values (“rules-of-thumb”) can be stated to prevent difficulties caused by cavitation erosion (assuming no corrosive medium ingredients !):

- ▶ $p_1 - p_2 < K_C \cdot (p_1 - p_v)$: no problems
- ▶ $p_1 - p_2 \geq K_C \cdot (p_1 - p_v)$ and $p_1 - p_2 < p_{crit,cav}$ (see Table 1): no problems
- ▶ $p_1 - p_2 \geq K_C \cdot (p_1 - p_v)$ and $p_1 - p_2 > p_{crit,cav}$ (see Table 1): problems
- ▶ $p_1 - p_2 \geq K_C \cdot (p_1 - p_v)$ and $p_1 - p_2 > 360$ psi: multistage globe valve design recommended (e.g., axial plug, Fig. 6).

Table 2. Noise behavior of various single-stage globe valve plugs

Globe valve type	X_{Fz} for 75% of C_v	X_{Fz} for < 75% of C_v	Resistance to contamination	Vibration behavior
Parabolic plug	0.25 to 0.35	Clearly up to 0.8	High	Poor for single-guided plugs
Piston-balanced plug with cage	0.25 to 0.35	Up to 0.5	Low	Good
V-port plug	0.25 to 0.35	Up to 0.5	High	Excellent
Perforated plug	0.35	Very low	Low	Good

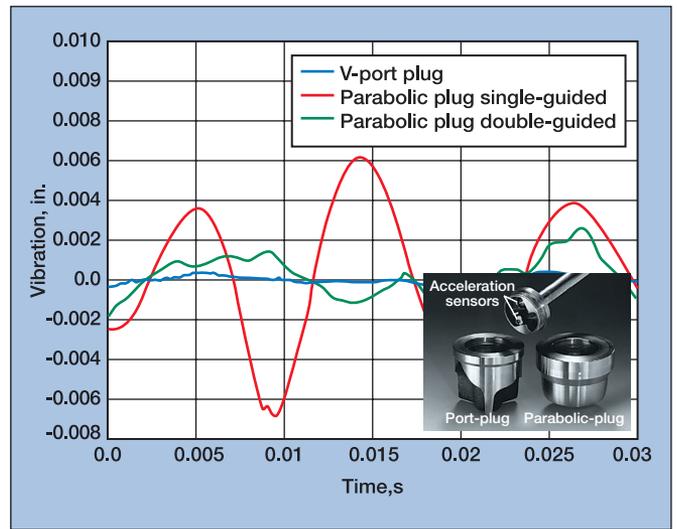


Fig. 8. Differences in vibration behavior (horizontal to the plug stem movement direction) of V-port plug vs. parabolic plug.

Table 1. Limit values for preventing cavitation erosion

Valve design	K_C	$p_{crit,cav}$, psi
Single-stage globe valves with stellite hardened valve plug and seat	0.7	360
Single-stage globe valves with standard materials	0.7	220
Rotary plug valves (with eccentric spherical disk)	0.4	140
Butterfly and ball valves	0.2–0.3	70

Measures against excessive noise emission caused by cavitation.

If noise emission reduction requirements are very strict, the control valve must be selected so that x_{Fz} values (which depend on the valve stem position) are sufficiently high to fulfill existing process conditions, i.e., $x_{Fz} > x_F$.

The K_C value, as a measure for incipient choked flow, is essentially determined by valve design, i.e., globe or butterfly valve. Additionally, x_{Fz} values are influenced by the special design and geometry of the throttling element. Typical x_{Fz} values versus valve stem position are listed in citation 1. A variety of measurements showed that the valve’s Reynolds number plays an important role. It is accounted for in the variable F_{XFz} , as depicted in Fig. 7, which is used to approximate x_{Fz} values for different valve types and sizes with one curve. The black curve represents a mean approximation which will also be part of the currently being revised IEC 534-8-4 standard² at the suggestion of the author. From this, two fundamental tendencies can be derived as follows:

- As the C_v value rises, the x_{Fz} value decreases
- As the valve style factor, F_d^4 , decreases, the x_{Fz} value rises.

Throttling element design details are expressed by the F_d value. The smaller the hydraulic diameter, the

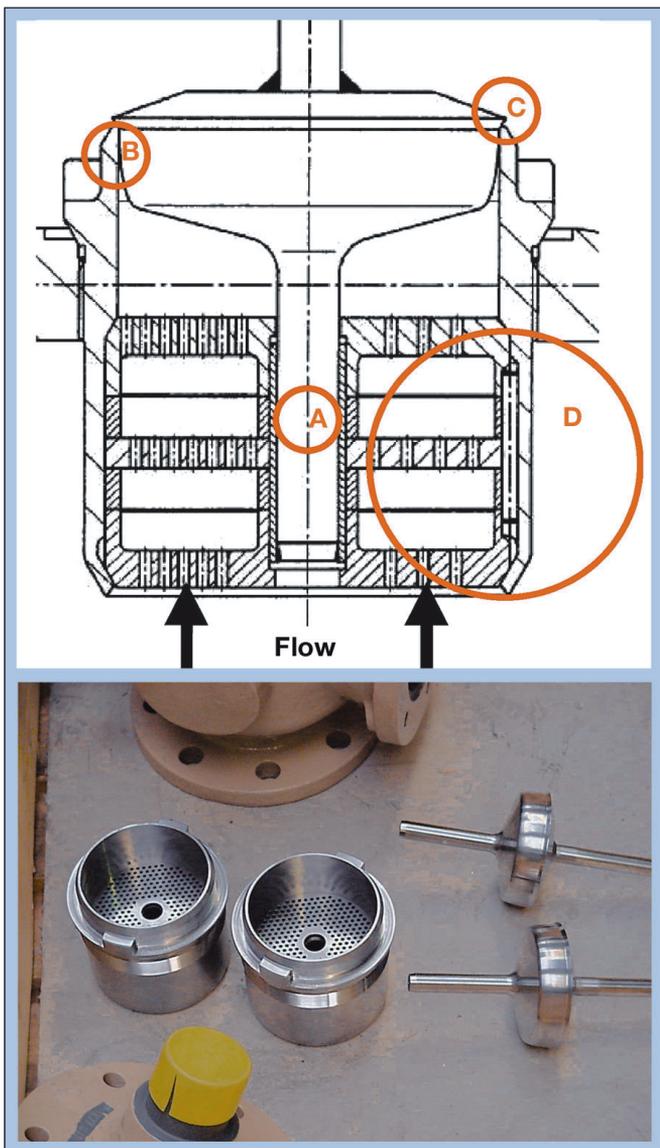


Fig. 9. New type of throttling system for standard valve bodies to minimize noise caused by cavitation.

smaller this variable becomes. The smallest hydraulic diameters in the vc can be achieved with small annular gaps, while surface area of the throttling element remains the same compared to other vc geometries. Such geometries are found with parabolic plugs that have correspondingly a reduced rated C_v value and maximum possible seat diameter in the valve body.

Globe valves with a nominal size > 2 in. with relative travels of 75% normally have x_{F_z} values between 0.2 and 0.35. Depending on the type of throttling element used, the x_{F_z} value increases more or less as relative travel percentage decreases. This is directly linked to a more or less decreasing F_d value.

Table 2 clearly shows that all plug types, except the parabolic, in typical refinery applications such as in Figs. 1 and 2, can develop high-noise emission levels produced by cavitation at differential operating pressure ratios $x_F > 0.5$ to 0.8. Hence, parabolic plugs have a distinct advantage due to their higher x_{F_z} values when valve relative travel position is below 75%. The disadvantage that goes with it, however, is that occurrence of cavitation may cause strong mechani-

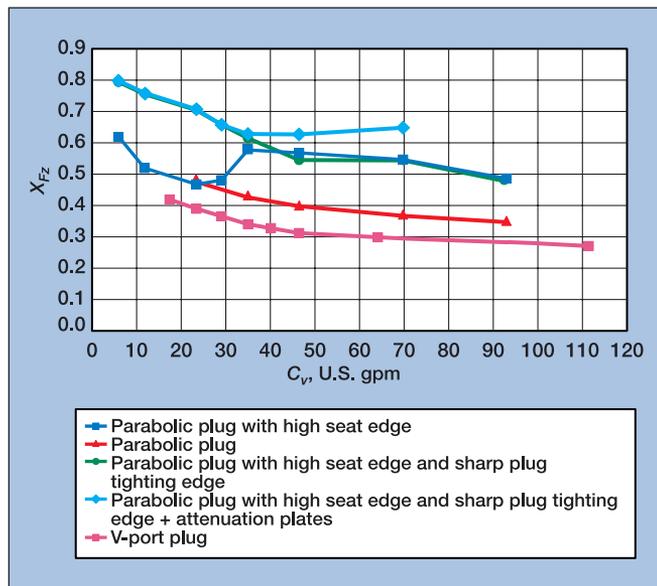


Fig. 10. x_{F_z} values for 80 mm seat diameter with throttling system as shown in Fig. 9 and V-port plug as shown in Figs. 8 and 14.

cal vibration that could result in a torn-off plug when the plug stem is only single-guided in the top of the valve body. V-port plugs with asymmetrical orifice geometry show the least susceptibility to vibration (Fig. 8). Similarly, cage-style valves and perforated plugs have a smaller tendency to vibrate. However, they have the disadvantage of being highly susceptible to contamination from particulate matter, resulting in fretting (holes or channels formed by erosion) and subsequent jamming problems.

Can the advantages of the individual plug types be combined so that noise levels produced by cavitation can be minimized?

Fig. 9 shows a new kind of throttling system that can be integrated in standard globe control valve bodies. This system has been optimized, using computational fluid dynamics (CFD) and performing a series of measurements. The system has the following properties:

- ▶ Seat size is not reduced to attain the thinnest annulus between the seat and the plug. The result is a small hydraulic diameter and, therefore, small F_d factor, which has a positive effect on the x_{F_z} value. In addition, the longer “throttling channel” causes pressure to decrease continuously (Fig. 11a).

- ▶ To prevent mechanical vibration, the plug is double-guided in the body, and optimally in the seat, near the location where vibration is produced (Fig. 9, circled letter A).

- ▶ The seat has a “high seat joint” (seat/plug sealing surface Fig. 9, circled letter B), which increases the x_{F_z} value for large valve plug openings because this has a positive effect on the vortex structure in the area below the plug (Fig. 11d).

- ▶ The seat joint is cut back at the plug (Fig. 9, circled letter C). This design increases the x_{F_z} value when the valve relative travel position is below 75%, because it has a positive effect on the vortex structure in the area above the plug (Fig. 11c).

- ▶ Attenuation plates can be integrated in the seat below the plug. They increase the x_{F_z} value additionally when the valve-open position is over 75% (Fig. 9, circled

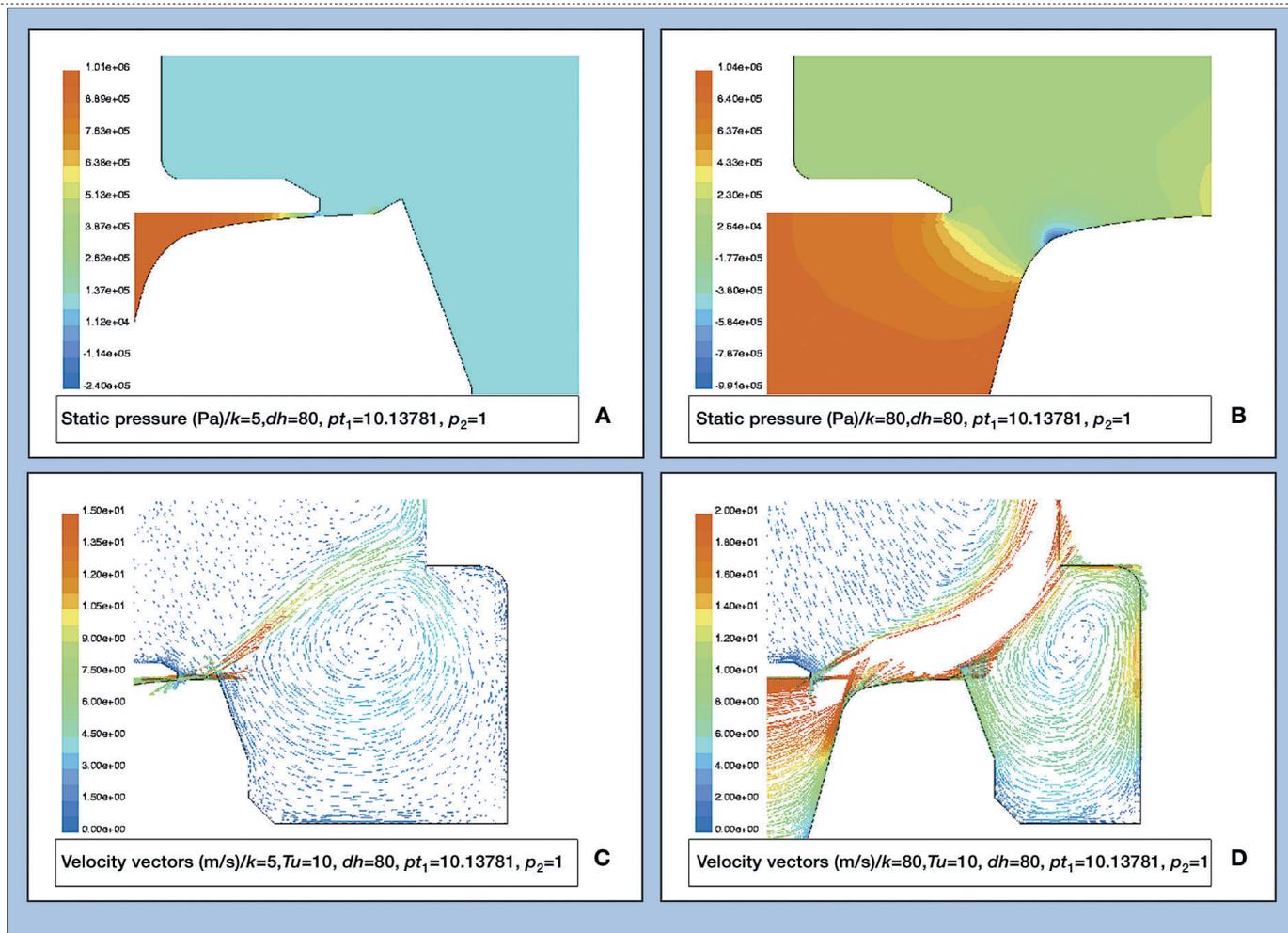


Fig. 11. CFD simulations for two plug positions ($C_v = 5.9$: a, c and $C_v = 94$: b, d): isobars and velocity vectors.

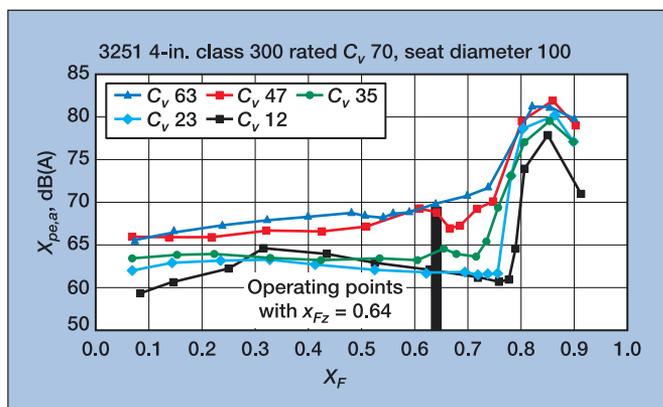


Fig. 12. Practical example from a refinery with high noise reduction requirements ($p_1 = 160$ psi, $p_2 = 58$ psi, $Q = 54,905 \dots 263,000$ lb/h oil, $X_F = 0.64$ and external sound pressure level, $L_{p,e,a} = 70$ dB (A)).

letter D). Additionally they further reduce noise level.

► The plug can also be designed as a pressure-balanced version—without a cage (instead with a “hollow” skirt-shaped plug bored with small holes and sealed internally against an assembly fixed to the underside of the bonnet).

► For pressure differentials > 145 psi with severe cavitation ($x_F > K_C$), the plug or seat should be stellite or, better yet, precipitation-hardened carbide or martensitic steels.

Based on the measured x_{Fz} value plots, Fig. 10 clearly shows that such a system emanates noise produced by cavitation only at differential pressure ratios that are much higher than for V-port or piston-balanced cage-style plugs.

This new type of throttling element was used, for example, in a typical refinery application to adhere to the required sound level < 70 dB(A) for the given operating points (Fig. 12). This application in practice confirmed measurements made on the test bench.

In a different refinery, a rotary plug valve caused great noise and erosion problems at maximum valve inlet pressures $p_1 = 220$ psi and $x_F = 0.5 - 0.9$. After $1\frac{1}{2}$ yr of operation, the valve body was eroded by cavitation (Fig. 5, critical cavitation, see Table 1) and was replaced by the same rotary valve type. Valve body thickness of the new valve was checked by thermographic visualization equipment in regular intervals by an independent consulting institute. After a further $1\frac{1}{2}$ yr and multiple valve investments, the rotary valve was replaced with a globe valve designed as described previously. This new valve successfully eliminated the problems in this case as well.

Measures against flashing. In case of flashing or two-phase flow, it is imperative that a sufficiently large nominal valve size be selected to limit average flow velocity on the downstream side of the valve,

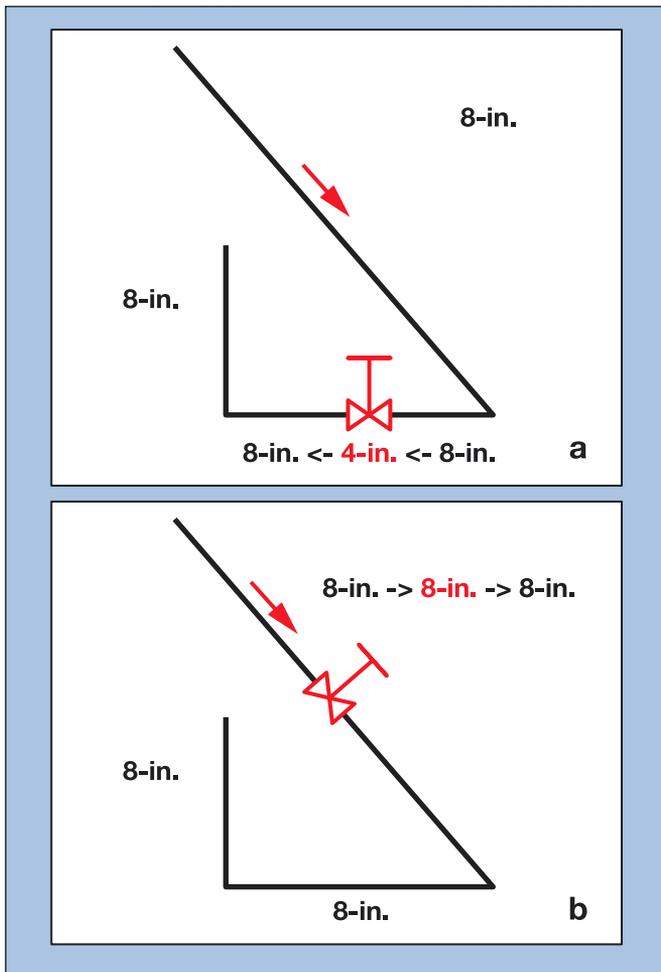


Fig. 13. Changes in valve and pipe isometry in a refinery system (a: before, b: after).

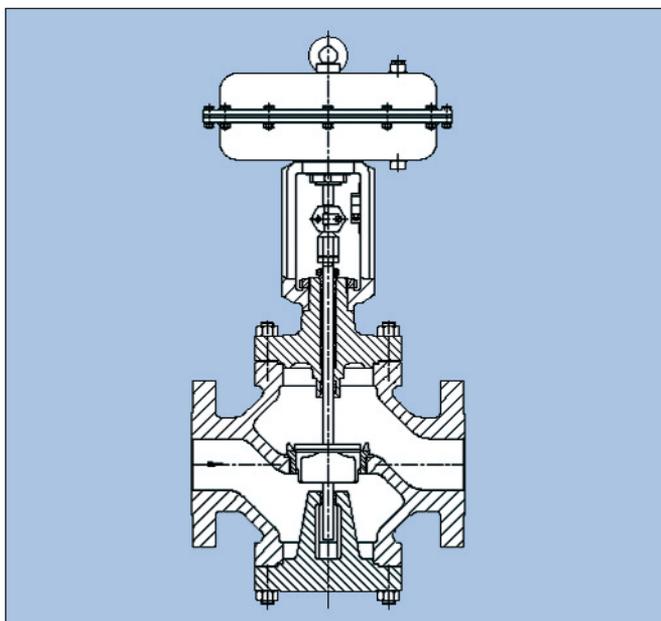


Fig. 14. Globe valve with low-vibration V-port plug.

thus minimizing the problems described in the section on cavitation and flashing. Experience shows that a maximum value of 200 fps should not be exceeded.⁵ For applications with active corrosive media, proper material selection additionally plays a

significant role. Also, seat size should be reduced by at least one step to instantly provide as much space as possible for flow on the p_2 side.

In a refinery, a 4-in. valve with the operating data described below caused significant problems: In the valve and in elbows upstream and downstream of the valve (Fig. 13a), erosion in combination with corrosion occurred. Evaluating existing operating data—including enthalpy data—showed that average outlet velocity at standard flowrate and approximately 13% vaporization was approx. 820 fps. Therefore, wear caused by erosion due to droplet impact was practically inevitable. An additional disadvantage was installing the valve between two elbows within a short distance.

The problems were eliminated when an 8-in. valve featuring a low-vibration V-port plug with C_v 120 linear trim (Hastelloy trim chosen for corrosion resistance) was installed. The valve was centrally installed in the long pipeline upstream of the first elbow. Here costs of ownership were ten times higher than the first investment cost of the implemented valve.

At a different location in the same refinery, massive vibration problems resulted in cracking the pipeline supports. The originally-installed rotary plug valve was much too small (nominal size 12 in.). Resulting excessive flow velocities produced the flow conditions shown in Figs. 4a and b, thus causing a strong vibrational pipeline excitation that was further intensified by the high-pressure recovery intrinsic to rotary plug valves. Installing a globe valve with a larger nominal size (20 in., Fig. 14) and a V-port plug solved this problem. Similar to the examples described previously, costs of ownership significantly added to the original investment cost, until a suitable valve solution was finally selected and installed. ■

ACKNOWLEDGMENT

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